Instrument and Control Manual
for Operating Engineers

Instrument and Control Manual for Operating Engineers

by Eugene W. F. Feller

Operating Engineer; Formerly
Assistant Chief Operator, Safe Harbor Water
Power Corp.; Associate Editor, Power; Member: National
Association of Power Engineers, National Association of
Corrosion Engineers, The Instrument Society of
America; Associate: The American Society
of Refrigerating Engineers

McGRAW-HILL BOOK COMPANY, INC.
NEW YORK AND LONDON
1947

INSTRUMENT AND CONTROL MANUAL FOR OPERATING ENGINEERS

PRINTED IN THE UNITED STATES OF AMERICA

To My Father

PREFACE

Today, the size of industrial plants, value of materials in process, and the number of units to be controlled in a single plant all tend to render hand control impractical if not impossible. In many instances, dependable and accurate control of pressures, temperatures, liquid level, humidity, and speed is essential to efficient operation. Instruments take the guesswork out of power-plant and process operations, and controllers increase efficiency by making uniform and continuous adjustment that cannot be approached by hand manipulation.

This book is a practical manual for the man in the plant who needs fingertip information for the quick and sound solution of specific metering and control problems. It explains the basic principles of control and describes the construction and operation of liquid-level, pressure, temperature, speed, and humidity indicators and controllers. After the individual units that go to make up a complete control system are described, they are connected together to show how the operation of one unit affects that of another.

Considerable space is devoted to explaining the operating fundamentals of each primary element as the text progresses, to give the reader a clear understanding of its physical characteristics. The material is presented in a practical way so that the reader will become familiar with all available types of instruments and controllers, will know how they operate, and will learn what points to consider when new equipment is to be selected.

The author is indebted to *Power, Chemical Engineering, Mechanical Engineering, Valve World,* and *Instrumentation* for material secured from these sources, and to his wife Claire for her help in reading proof. Acknowledgment is made throughout the text for data generously contributed by manufacturers and for technical information published by other engineers.

PORT WASHINGTON, N.Y.
September, 1947

EUGENE W. F. FELLER

CONTENTS

CHAPTER IX

CHAPTER X

CHAPTER XI

CHAPTER XII

CHAPTER XIII

CHAPTER XIV

CHAPTER XV

CHAPTER XVI

CHAPTER I

SIMPLE LIQUID-LEVEL INSTRUMENTS

Perhaps the simplest method of measuring liquid depth is the chain used by sailors to take soundings. Although simple, it is not convenient to use in factory or power-plant tanks that are many shapes and sizes and are located in many out-of-the-way places. For such tanks, other means are required to indicate a true level of the contained liquid. Among these are *staff gages, floats, gage cocks, gage glasses, pressure gages, controlled air pressure* in a pipe submerged in the liquid, and *pressure differential* between two liquid columns.

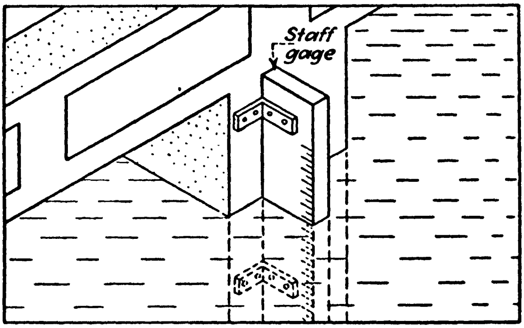


Fig. 1-1.—Wood or metal staff gage serves for an easily accessible tank or reservoir.

A staff gage (Fig. 1-1), the least expensive and easiest to make, is commonly used on reservoirs and open tanks. It is nothing more than a piece of channel iron or wood planking with divisions representing quantity or feet of elevation above reservoir bottom. The common measuring stick is a portable staff gage for insertion through small openings in enclosed tanks.

Floats made of buoyant material or metal rise and fall with the liquid to operate indicators or controllers. Organic-material floats soon lose buoyancy unless given a liquidproof coating. Boiling in paraffin or beeswax protects against water; shellac prevents oil absorption. Periodic renewal of the coating extends service life considerably; because, once the surface becomes cracked and the float soaked, it cannot be dried satisfactorily unless the entire coating is removed. Troubles from liquid absorption are eliminated with metallic airtight floats (Fig. 1–2). Although these are slightly higher in first cost, their replacement frequency and cost are low. Operating shafts pass through packing glands, sealing bellows, or flexible diaphragms (Fig. 1–3).

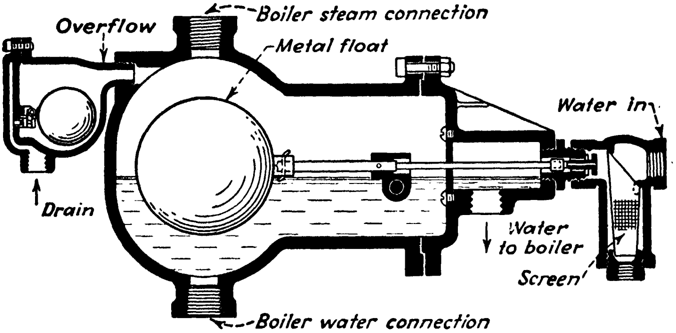


Fig. 1–2.—Metallic airtight floats eliminate trouble from liquid absorption.

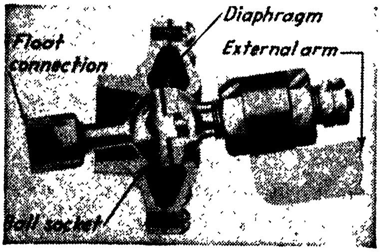


Fig. 1–3.—Flexible diaphragm seals float chamber. (*Courtesy of Davis Regulator Co.*)

The net buoyancy of a hollow float completely submerged is obtained by subtracting the weight of the float from the gross buoyancy as determined from tables or by the formulas $1/6\,\pi d^3 =$ volume and volume (cu. in.) $\times\ 0.0361 =$ pounds gross buoyancy in 70 F water. For steam-pressure service, calculations of gross buoyancy should include a factor for the weight of water at the maximum operating pressure and temperature. The specific gravity of other liquids should be included in the calculations.

Seamless-copper floats serve in a wide variety of liquid-level control and indication applications for hot and cold water and steam service where operating pressures range up to 150 psi.

Stainless-steel floats are adaptable to high-temperature (800 to 1600 F) high-pressure (1,500 psi) service and where high corrosion resistance is necessary. Conventional steel floats are used for open-tank cold-water service and for ammonia, oil, and alkaline solutions up to 1,500 psi. They are available with protective coatings of zinc, nickel, chromium, or cadmium.

Floats made of aluminum can be used in low-specific-gravity liquids where heavier floats do not have sufficient buoyancy. Although attacked by solutions of salt, sulphur, and hydrochloric acid, they can be used in alcohol, turpentine, nitric acid, dilute sulphuric acid, formaldehyde, and many other liquids. They are not suited for high-temperature service.

Pure nickel is highly resistant to corrosion in natural sea water and in water containing hydrogen sulphide or free carbon dioxide. It resists hydrochloric acid and acid salts but is not especially resistant to sulphuric acid. Nickel also resists corrosion by neutral or alkaline-salt solutions and is useful in food products and refrigerant brines.

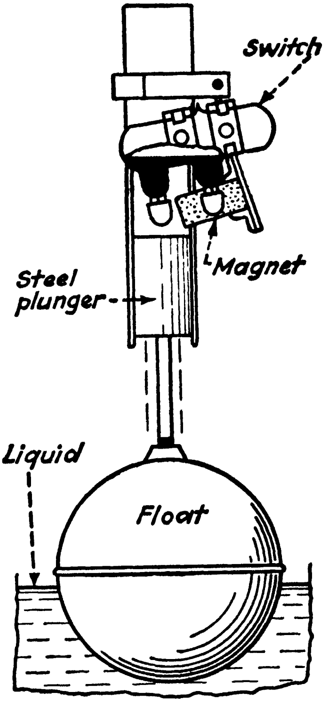


FIG. 1-4.—Metal float operates switch with permanent magnet.

Monel possesses a useful degree of resistance to more corrosive liquids than perhaps any other malleable metal. It is highly resistant to refrigerant brines and is widely used in laundries and textile dyeing. It is one of the few metals that can be used in hydrochloric acid. Monel and nickel floats can be used under pressures up to 1,000 psi and temperatures up to 800 F.

Floats are widely used to record and control the liquid level in traps, boilers, reservoirs, and elevated tanks with practically every kind of control device. The unit in Fig. 1–4 moves a steel plunger up and down inside a sleeve. A powerful magnet outside is attracted as the plunger moves upward. As the magnet swings on its pivot, it opens or closes the mercury-switch contacts to operate a signal or control valve.

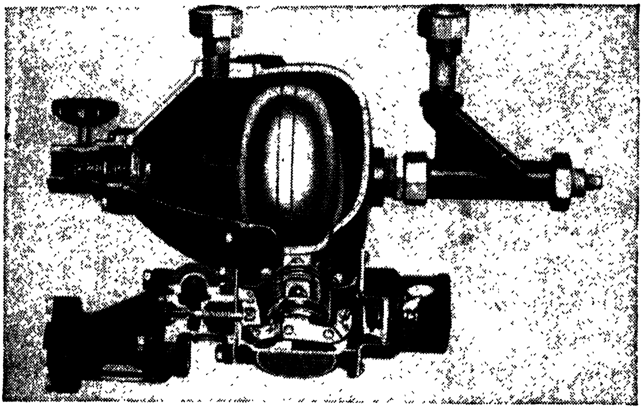


FIG. 1–5.—Float-operated boiler feeder also protects against low water. (*Courtesy of McDonnell & Miller.*)

The boiler feeder and combination cutoff float controller (Fig. 1–5) regulate feed-water flow to a boiler to maintain correct water level; but, if low water occurs, the automatic cutoff feature shuts off the fuel supply. The deep float chamber is provided with a blowoff valve to remove sediment. The valve can also be opened to lower the water level and test the feeder operation.

The float mechanism in Fig. 1–6 moves a pilot to admit pressure to or bleed pressure from the main diaphragm chamber. As pressure acting on top of the diaphragm bleeds away pressure under the valve, the disk raises it from its seat. On reverse operation, the pilot admits pressure on top of the diaphragm, which has a greater area than the valve disk. Thus the diaphragm can force the disk to its seat.

The system in Fig. 1–7 controls liquid level in the tank by regulating the pressure applied to the diaphragm-operated supply valve. Supply pressure in the pilot line passes through a pressure-reducing valve and adjustable orifice on its way to the diaphragm and float-operated bleed valve. Since the reduced pressure is constant, the flow through the orifice and therefore the pressure acting on the diaphragm valve are determined by the position of the float-operated bleeder valve.

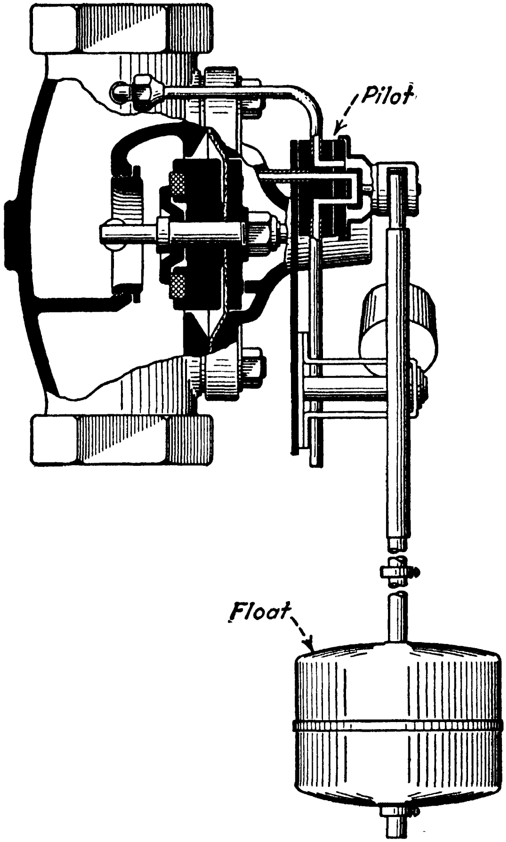


FIG. 1–6.—Float-controlled pilot for diaphragm valve.

High-pressure floats must be light and yet so strong that they will not collapse. Filling them with air or gas under high pressure is one possible solution, but this requires a thick-walled or extremely heavy float. The unit in Fig. 1–8, suitable for pressures up to 5,000 psi, contains a float ball that is vented to tank pressure by the hollow stem *A* leading from the bottom of the float to chamber *B*. Once the float and stem are filled with air or gas, no great amount of liquid can enter even if the unit becomes submerged.

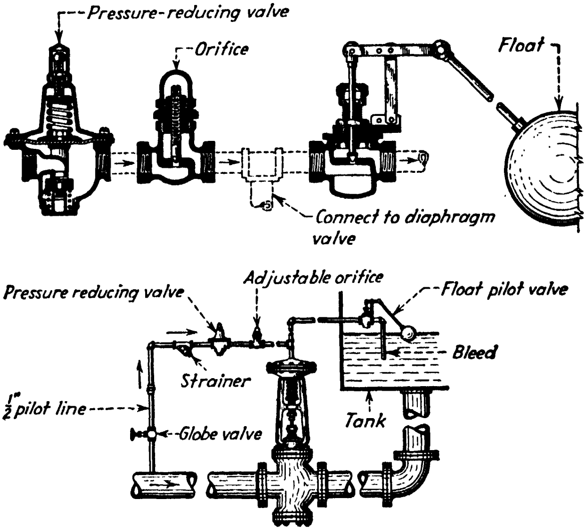


FIG. 1–7.—Throttling control with float. (*Courtesy of Kieley & Mueller, Inc.*)

The float can fill with liquid only if steam or other vapor condenses inside.

Assume a 4-in. 33.5 cu in. spherical float to operate in a vessel under 5,000 psi abs. The buoyancy of the float when loaded and totally submerged is 8 oz. Thus 8 oz of liquid must enter to make it sink. Suppose the tank contains water that weighs 0.578 oz per cu in. Then $8 \div 0.578 = 13.83$ cu in. of water needed to sink the float. Thus, if this quantity of water enters, the volume of air in the float must be compressed into $33.5 - 13.83$, or 19.67, cu in. The pressure required to compress the air into this space is $\frac{5{,}000 \times 33.5}{19.67} =$ 8,520 psi. Thus the vessel pressure must almost double before the float will sink. Enclosing chamber *B* also traps air or gas, which must be compressed in the float before water can enter the stem.

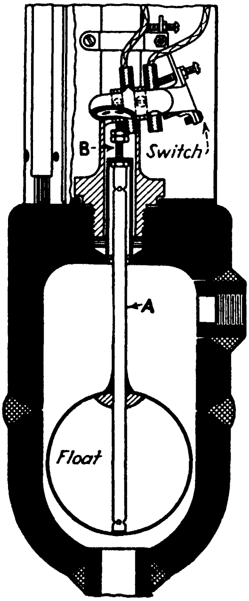


FIG. 1-8.—Hollow float for high-pressure service. (*Courtesy of Fred H. Schaub Engineering Co.*)

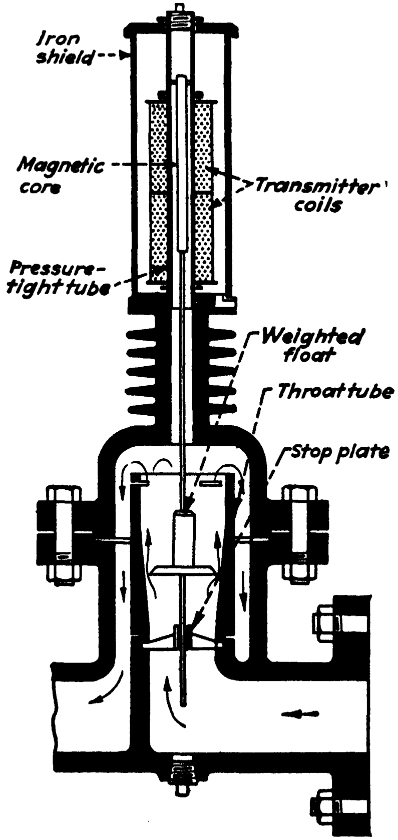


FIG. 1-9.—Float movement unbalances an impedance-coil circuit.

The manufacturer must be given full particulars of the installation requirements when this device is ordered.

Another design places the float directly in the flow stream. In one design, the float carries a permanent magnet that moves a follower. In another (Fig. 1-9), the float moves an iron core inside a pair of impedance coils to unbalance a twin pair of coils in the indicator or controller. This method controls the liquid level in a vessel by interconnecting two transmitting elements to a master unit. Here one float element is piped to the vessel inlet and the other to the discharge. Unbalanced flow operates a master controller to correct the level. These mechanisms are suitable for heavy oils and tars that must be heated to flow freely because instrument parts remain in contact with the heated fluid. Their accuracy is affected if temperature variations change the density of the fluid.

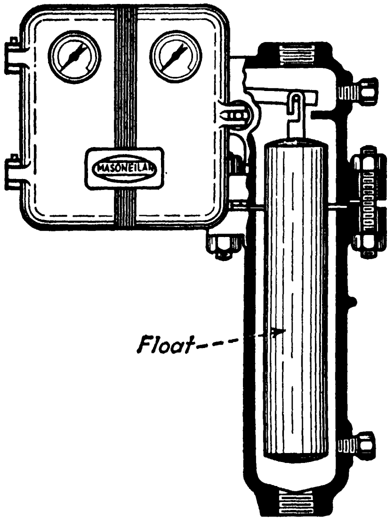


FIG. 1-10.—Weighted displacer acts like a float. (*Courtesy of Mason-Neilan Regulator Co.*)

The weighted displacer (Fig. 1-10) operates like a float in that the surrounding liquid tends to make the object weigh less as felt by its connected linkage, and the displacer therefore rises slightly in proportion to the liquid-level increase. Change in weight is directly proportional to the amount the displacer is submerged; it is heavy enough to sink in any liquid encountered, and the buoyancy acting is equal to the weight of the liquid displaced.

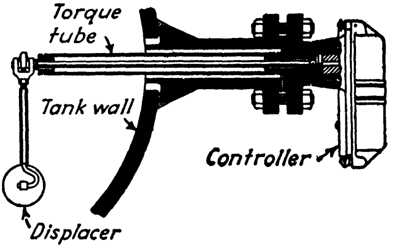


FIG. 1-11.—Torque tube seals displacer chamber. (*Courtesy of Fisher Governor Co.*)

Since the displacer actually moves a relatively small distance for a wide level change, this principle of measurement gives a greater range of control than that provided by conventional floats. The longer control ranges are obtained with a long displacer of small cross-sectional area. Weighted displacers are especially suitable for interface service (indication and control of the interface level of two liquids having different specific gravities, *e.g.*, to indicate or control the water level in an oil-and-water separator).

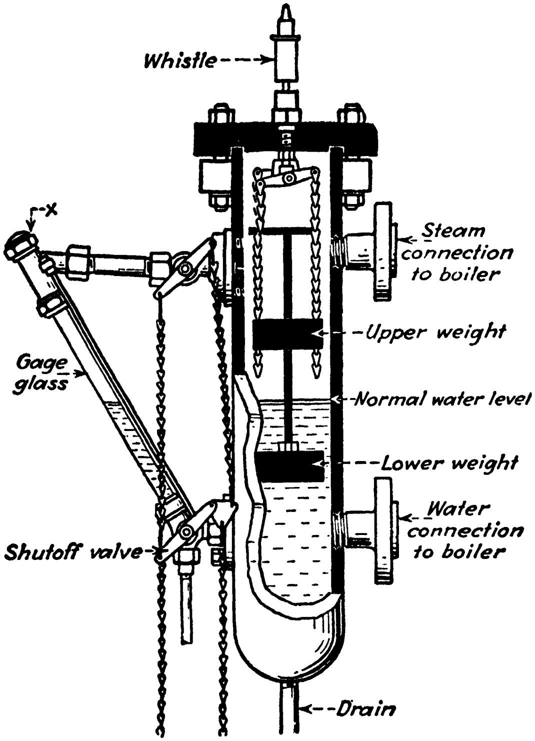


FIG. 1–12.—Two counterbalanced weights indicate boiler-water level. (*Courtesy of Yarnall-Waring Co.*)

Because weighted-displacer units are suitable for high-pressure and high-temperature service, a torque-tube seal instead of ordinary packing is used in the operating linkage. Displacer movement oscillates the shaft that passes through the tube (Fig. 1–11). The tube and shaft are welded together at the displacer end so that, when movement occurs, the torque tube twists about its axis. Two solid metal weights balanced against one another in the same vessel

(Fig. 1–12) also register liquid level by sounding an alarm if the level gets too high or too low.

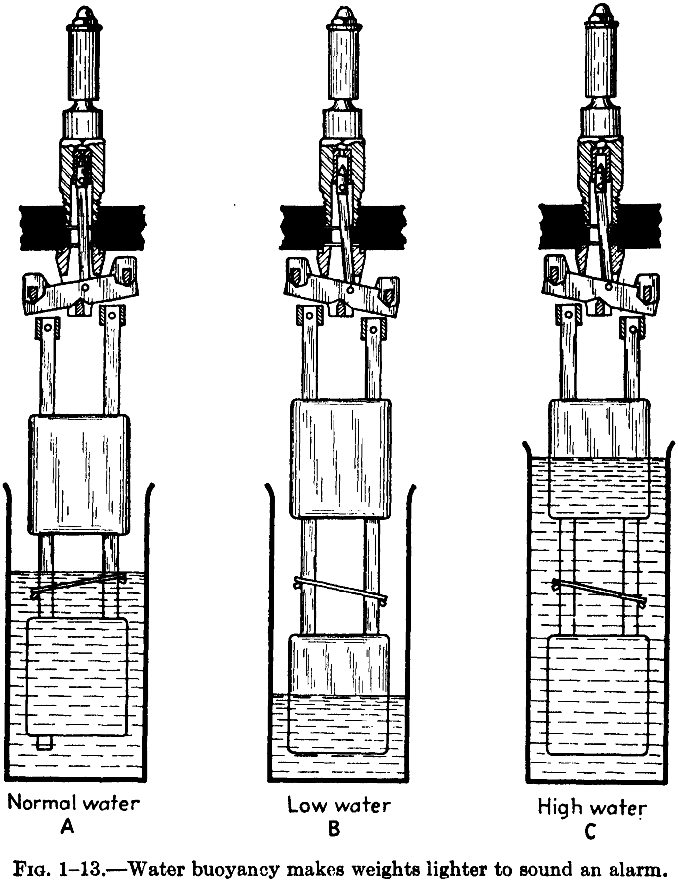


FIG. 1–13.—Water buoyancy makes weights lighter to sound an alarm.

The mechanism in Fig. 1–13 (similar to Fig. 1–12) operates on the principle that objects submerged or partly submerged in water are buoyed up by a force equal to the weight of water they displace. This loss of weight is utilized to open and close a needle valve through which steam is admitted to the alarm whistle. The two solid weights *H* (upper) and *L* (lower), mounted one above the other, are suspended from opposite ends of a lever that pivots on a knife-edge fulcrum attached to the cover flange.

The weights *H* and *L* are of equal volume and so balanced that, when *L* is submerged and *H* is in the steam space (normal water level), the alarm valve is closed, *A*. When weight *L* is only half submerged (low water), equilibrium is destroyed and the whistle blows, *B*. If the upper weight H becomes half submerged (high water), the whistle again blows, *C*.

For active liquids, a float must have rigid guide or be restricted by still wells to prevent its floating around over the surface and giving a false reading. For tanks below floor level, the float is fastened to a rod and target that moves vertically along a staff gage; on overhead tanks, a flexible cable moves the target up and down a staff gage mounted vertically on the outer tank wall.

With only a single pulley, the float moves the target toward the foot of the staff gage as the tank fills. A triple pulley and counterweight arrangement reverses the target movement so that its position corresponds to that of the liquid. Self-contained float chambers mounted on the side of a tank or boiler at the normal liquid level (Figs. 1–2 and 1–5) indicate within a narrow range when the supply exceeds or falls below a given amount.

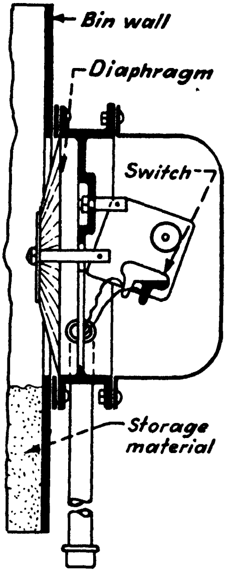


FIG. 1-14.—Diaphragm operates mercury switch to show level of bulk material. (*Courtesy of The Bin-dicator Co.*)

The unit in Fig. 1–14 is a diaphragm-operated mercury switch for use on bulk materials, semiliquids, and liquids. It serves as an indicator or automatic control for conveyors, elevators, feeders, pumps, and weighing devices. Mounted on the cover of a screw conveyor, the unit gives an alarm on choking, overfeeding, or underfeeding. Several units mounted on a vertical tank indicate the internal level in steps.

Gage cocks, although sometimes used alone, are often installed as an emergency check against other level indicators such as gage glasses. One (Fig. 1–15*A*) uses the threaded stem and the other (Fig. 1–15*B*) the lever-operated push-button method of opening.

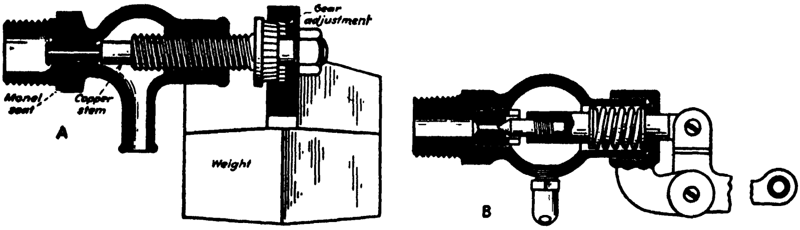


FIG. 1–15.—Gage cocks can be used to determine water level.

The threaded-stem cock has the advantage of turning action to clean the seat as it closes, whereas the push-button valve must be ground to an accurate seat to prevent leakage. Reseating is also much easier on the threaded-stem valve, because it does not require final grinding after being refaced in a lathe.

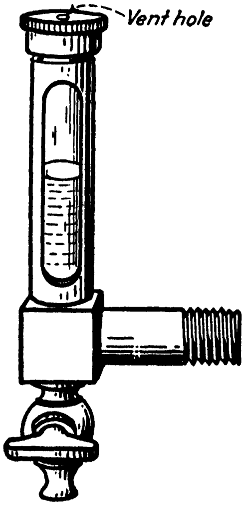


FIG. 1–16.—Single-connection gage glass serves on bearing oil reservoirs.

Gage glasses constitute one leg of a U tube, the other leg being the tank or vessel itself. The liquid in the two sides of a U tube assume a common level as long as both ends are acted upon by the same temperature and pressure. Higher pressure on one leg forces the liquid down until the height in the opposite leg equals and balances this higher pressure. A temperature difference or a change in specific gravity (steam bubbles) of the liquid in the two legs will also unbalance the levels. This accounts for a water elevation in a boiler drum slightly higher than that shown by the gage glass if it is some distance away.

Gage glasses are commonly used on storage tanks, stills, and boilers. To meet special requirements and prevent frosting from differences in temperature between the atmosphere and the contained liquid, two tubular glasses are arranged one inside the other and their ends fused together.

This forms an annular chamber between the two tubes which is evacuated to permit clear and accurate reading of the liquid level at all times. Gage glasses can have both ends connected (Fig. 1–12) or, as in the bearing-oil-level indicator (Fig. 1–16), only one end connected. When the single connection is used on a vessel vented to the atmosphere, the free end of the gage must also be vented, or a false reading results. A motor-bearing oil-level well having its upper end closed by a hinged metal flap depends on uneven contact of the metal cap for venting. If the cover is flapped closed continually, it hammers the metal to an airtight seal against atmospheric pressure, and the oil then rises and seeps out under the cover. This may be corrected by drilling a vent hole in the cap or filing a notch in the top edge of the well.

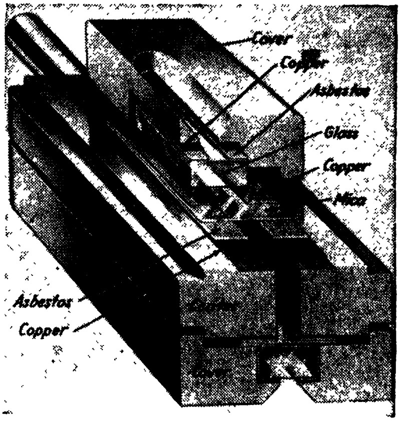


FIG. 1–17.—Mica inserts protect flat glass against etching.

Tubular glasses must withstand the pressure acting on the cross-sectional area of the tube end as well as the internal surface pres-

TABLE 1–1.—RECOMMENDED MAXIMUM WORKING PRESSURES FOR TUBULAR GAGE GLASS

Length, in.	Corning standard, psi	Pyrex broad red line, red line, and high pressure, psi	Pyrex heavy wall, psi
Up to 10	200	500	600
11–20	200	420	600
21–30	200	340	600
31–40	180	290	
41–50	160	220	
51–60	130	180	
61–70	100	140	

sure. A short glass is quite stiff and not appreciably bent by this loading, and its resistance to internal pressure is practically unaffected. A long tube, however, tends to bend, stressing the tube wall and adding to the stress caused by internal pressure.

Although there are no hard and fast rules covering gage glasses for different pressures, recommended practice has placed certain limitations on the kind and length of glass (Table 1–1). This is supplemented by a recommendation that flat glass be used on power boilers for all pressures up to 350 psi and that flat glass with mica inserts (Fig. 1–17) or mica alone be used for higher pressures.

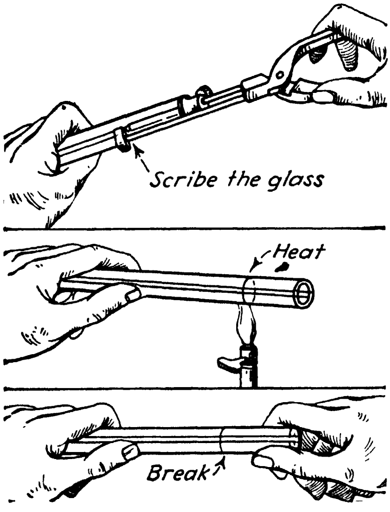


FIG. 1–18.—First scribe the glass, heat it uniformly, and break as shown.

Tubular glasses should be of the best material available. They should preferably be bought already cut to correct length because the manufacturer is equipped to produce a good fused end. Most operators prefer the inside cutter (Fig. 1–18) because it prevents flaking of the inside edge. After cutting, the ends are fused to a smooth surface. Sufficient length should always be allowed to reach past the upper packing recess when the glass rests on its seat in the bottom gage fitting. Letting the glass extend too far into the upper fitting gives opportunity for etching to take place. Any extensive sign of etching is a warning that the glass should be replaced. Metal objects nick the glass and should not be used for cleaning purposes.

The rate of attack on glass by condensing steam varies with temperature and water pH. The higher the pH value of the water the more rapid the attack. Every drop of condensate that trickles down a gage glass shortens its life. A windy or relatively cool location hastens condensation. Condensate from inlet steam piping should drain back to the water column instead of down through the glass. If this cannot be avoided, a tapered nipple inserted in the upper end of a vertical glass causes droplets to fall free.

The composition of water in the gage glass constantly changes because condensed distilled water continually enters, but quiescent conditions prolong glass life. Rapid fluctuation of the water level washes away the protective film of corrosion that forms on the surface, thus accelerating etching.

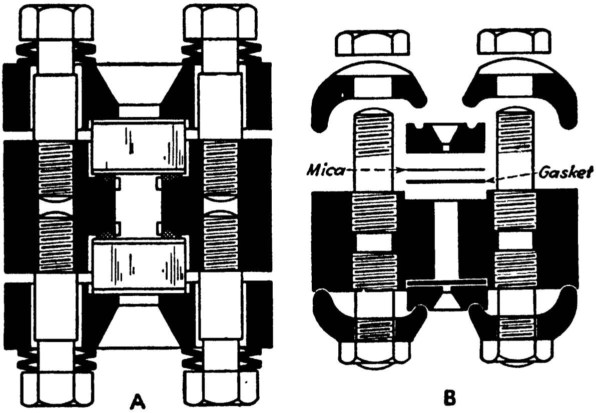


Fig. 1–19.—Clear flat glass and mica form the windows of these gage glasses.

To install a new tubular glass, remove the cap or plug X (Fig. 1–12) (when provided) from the upper fitting, and insert the glass at this point. Turn a plugless upper fitting slightly to one side, insert the upper end of the glass, and then turn the fitting back so that the lower end can slide into the lower fitting. Glasses should be of the correct diameter to slip easily into the gland—if they are too tight, expansion against the metal breaks the glass.

After the glass is in place, push the packing and the packing-follower washers into the glands. The packing should enter easily without being forced. Never use a wrench to turn down the gland nut; thumbtight is sufficient and allows for normal expansion. Test by turning the glass by hand. If a wrench is needed to stop leakage, insert new packing instead of applying extra pressure to the old. Slight initial leaks cease when full operating temperature is reached.

Poor-quality packing washers lose their resiliency quickly and soon permit steam to escape; this rapidly increases until the entire washer blows out or sufficient steam escapes to set up violent vibration and unequal expansion. Gage-glass washers for low and medium pressures are made from a high-grade vulcanized composition; for higher temperature and pressures, asbestos or metallic packing is necessary. Asbestos is not recommended for circular glasses, since it grips the glass too tightly and prevents longitudinal expansion.

Flat gage glass is available in two designs—one with clear and the other with reflex glass. The clear or plain glass (Fig. 1–19*A*) forms two sides of a case containing the column of water. With proper illumination behind the glass, the water level is plainly visible. A reflex glass has longitudinal ridges on the back surfaces that act as prisms to reflect light above the water level, giving that portion of the glass a bright silvery appearance that contrasts sharply with the dark area below the water level.

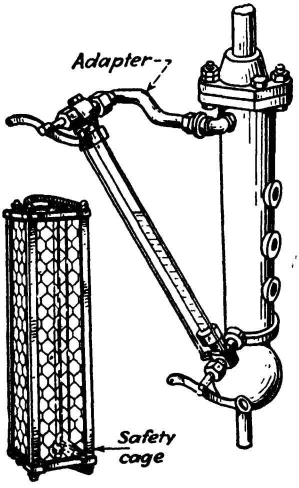


FIG. 1–20.—The safety cage catches flying glass. Adapter inclines the gage for better seeing.

Mica inserts are used with flat glass because they resist etching and lengthen glass life considerably. When used alone (Fig. 1–19*B*), a mica thickness of 0.025 in. is suitable for pressure up to 450 psi; for higher pressures, mica should be at least 0.035 in. thick. Discolored mica should be resplit and the good sheets saved for building up other inserts.

Flat glasses, which have a large area clamped by metal parts, need metal frames to support the glass uniformly around its rim. Uniform tension is assured by spring washers like those in Fig. 1–19*A*. When installing a new glass, draw the clamping nuts up thumbtight, and then tighten them in pairs to give uniform clamping sufficient to effect a positive seal.

Gage-glass protectors made of wire-embedded polished plate gage glasses, set at an angle to each other, utilize the difference in and steam. The guard consists of a metal frame supporting the four-sided glass enclosure (Fig. 1–20) suitable for mounting around

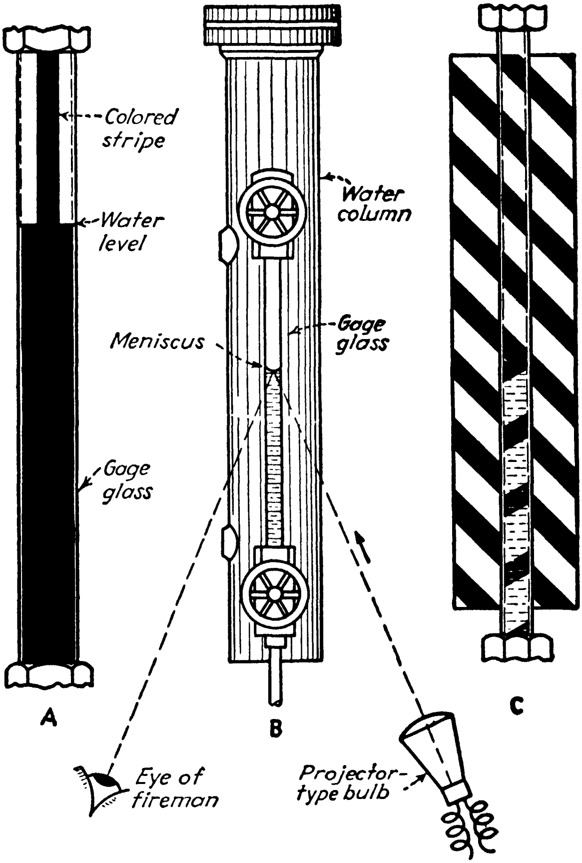


FIG. 1–21.—Several methods of improving gage-glass indications.

the tubular gage glass. The plates withstand the shock of a bursting glass and will not shatter or fly.

For high overhead gages, an adapter (Fig. 1–20) transforms a vertical glass into an inclined one. This permits easy level reading from the operating floor. Adequate lighting or a beam directed from a spotlight mounted at an angle below the glass (Fig. 1–21*B*) also helps visibility. A painted plate with barber-pole stripes placed behind the gage glass (Fig. 1–21*C*) makes the water level stand out with greater clarity. Another method (Fig. 1–21*A*) is to place a vertical stripe on the back of the tubular glass. A reflected light and mirror (Fig. 1–22) brings the gage image to the operating floor.

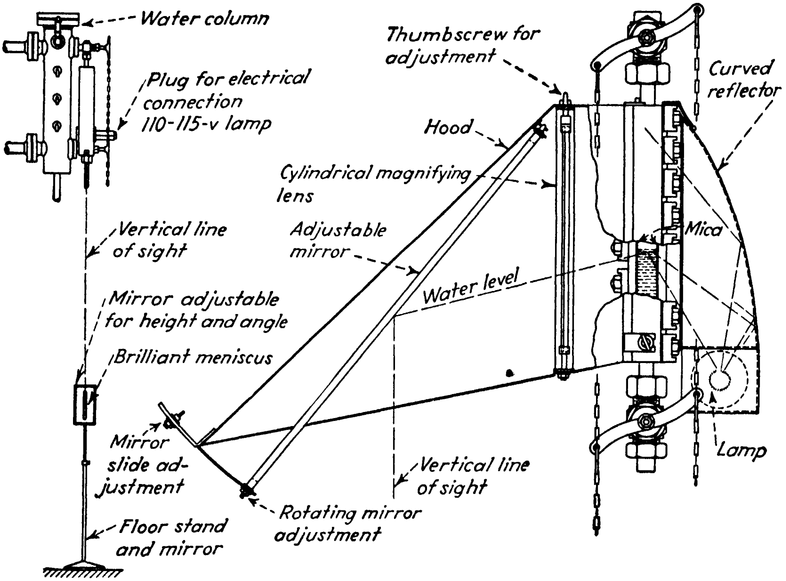
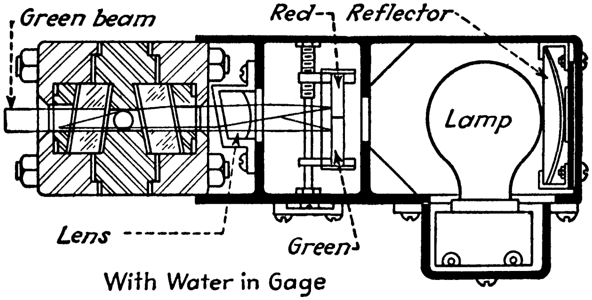


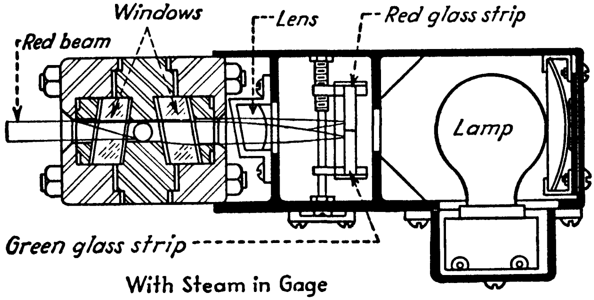
Fig. 1–22.—Light and reflector project the image to a remote mirror. (*Courtesy of Reliance Gauge Column Co.*)

A two-color reflecting-mirror device operates on the principle that refraction of a light beam differs as it passes obliquely through different substances. When the beam passes through air or steam, the amount it bends is not the same as when it passes through a column of water.

Parallel panes of red and green glass stand between the light source and a lens adjacent to the gage glass (Fig. 1–23). The gage glasses, set at an angle to each other, utilize the difference in refraction between water and steam to best advantage. Beams of red and green light project through the strip lens and strike the gage window at different angles. In the steam area, the green beam bends out of vision; but, in the water area, it remains in view. On the other hand, the red beam stays in view in the steam area and bends out of vision in the water area. The water level is indicated where red and green colors meet. Mirrors project the image to the operating floor.



With Water in Gage



With Steam in Gage

FIG. 1–23.—A reflecting gage glass uses color to distinguish water level.

When installing gage-glass fittings on a circular surface, use extreme care to ensure that the fittings are in proper alignment and that no strain is placed on the glass itself. To drill curved surfaces, hold the drill level and point it at the center of the tank. A template and drill guide or floor square (Fig. 1–24) can be used on any tank to hold the drill in the correct position. If you use the floor square, make allowances for any out-of-level condition between tank and floor.

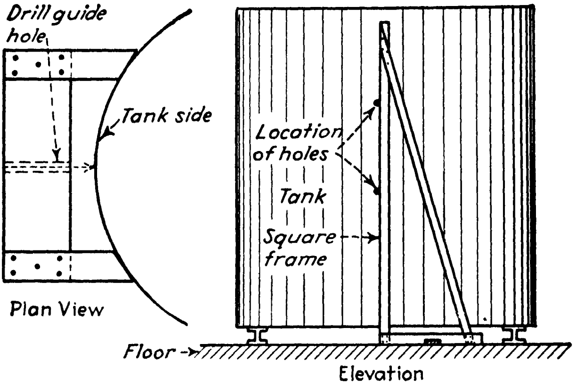


FIG. 1-24.—Prevent damage to gage glasses by installing the fittings properly.

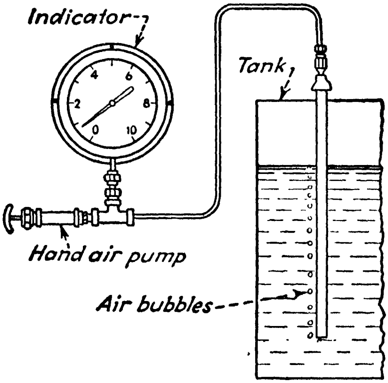


FIG. 1-25.—Pressure gage and hand air pump indicate liquid level.

After the hole positions are marked, move the square out a drill's length away from the tank, keeping its edge in direct line with the hole mark, then move it to the right a distance equal to half the drill diameter. When drilling, keep the drill level, with its side just touching the square edge. Tap the holes accurately so that the gage fittings protrude the same distance and are in perfect alignment with each other. Tipping a fitting to the right, left, up, or down will bring the glass in contact with metal parts and cause breakage within a short time.

Pressure gages indicate liquid level when they are fitted with the correct scale. When adjusted to take care of pressure drop, a gage can be installed on the pipe line at some distance from the tank. The operating element, consisting of a bourdon tube, metallic bellows, flexible diaphragm, or mercury chamber, can be actuated by the liquid pressure or by air pressure sufficient to balance the liquid head. Figure 1–25 shows a simple indicator consisting of a low-reading pressure gage, a hand-operated air pump or bulb, and the necessary piping. The gage dial can be calibrated in feet or gal-

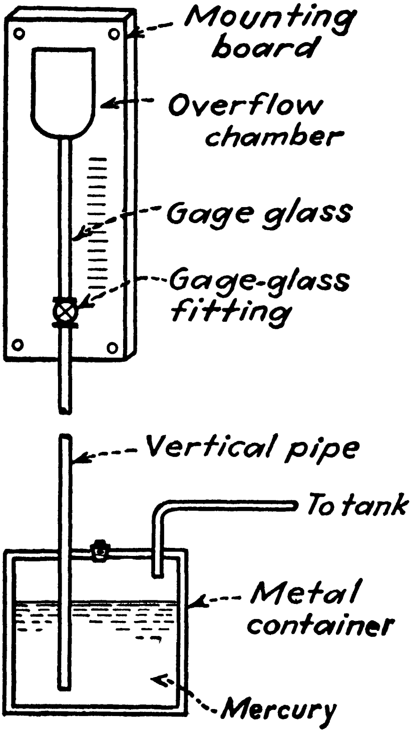


FIG. 1–26.—Mercury gage converts feet of water into inches to make a small gage.

lons. The pump fills the pipe with air of sufficient pressure to overcome liquid head, and excess air leaks out at the lower end of submerged pipe. The pressure acting on the gage is equal and corresponds to the pressure from the liquid surrounding the pipe.

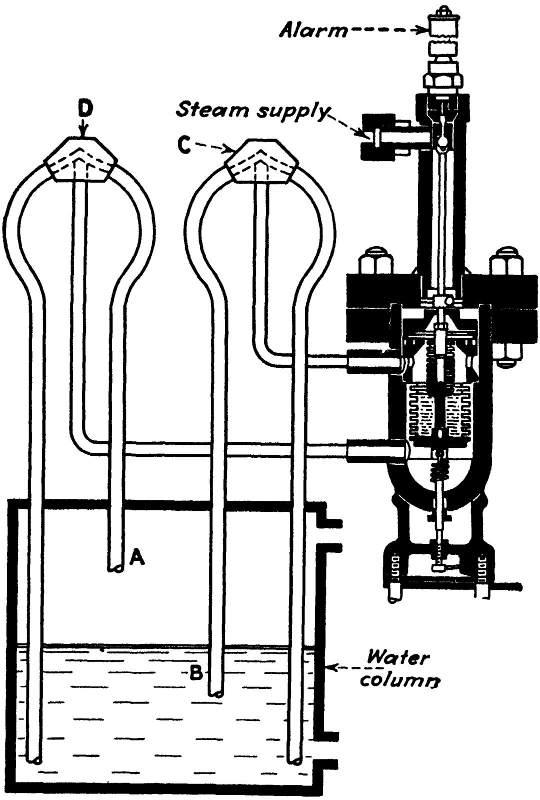


FIG. 1-27.—Two pipe loops in water column apply differential pressure to the alarm. (*Courtesy of Diamond Power Specialty Corp.*)

In a mercury-gage indicator (Fig. 1-26), liquid head is converted to mercury head and provides accurate indications for high overhead tanks. Although mercury cost is high, the device itself can be constructed and installed at small cost. With a specific gravity of 13.58, 0.88 in. of mercury is equivalent to 1 ft of water, and a short mercury gage serves a storage tank of considerable height.

The boiler water alarm (Fig. 1–27) uses two pipe loops connected to the water column. One end of each loop terminates near the bottom of the water column. Ends *A* and *B* terminate at the high- and low-water level, respectively. The actuating alarm is divided into three chambers by two bellows. The top chamber contains a steam supply to operate the alarm, the middle chamber receives pressure from the top of loop *C*, and the bottom chamber connects to loop *D*. With both pipe ends *A* and *B* under water or out of water, pressures in the middle and lower chambers are equalized and spring tension opens the steam valve to the signaling device.

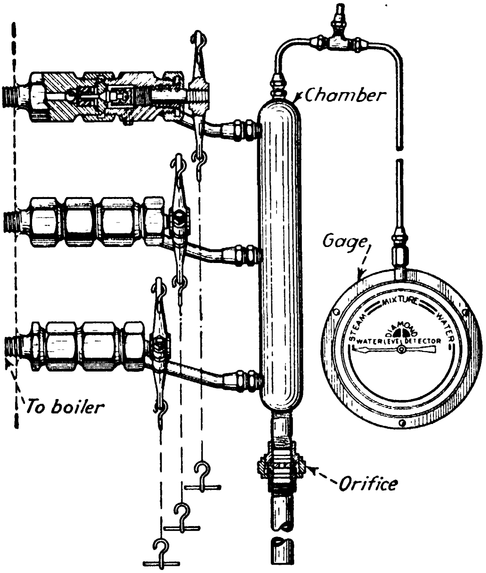


Fig. 1–28.—A pressure chamber and discharge orifice show whether water or steam emerges from the vessel. (*Courtesy of Diamond Power Specialty Corp.*)

The boiler water-level detector (Fig. 1–28) consists of three quick-opening water-level valves (instead of conventional gage cocks) connected to an expansion chamber fitted with an orifice to control discharge to the atmosphere. The indicator gage hand indicates whether the mixture in the chamber is steam or water or a mixture of the two. The amount of water entering the chamber determines how much the gage pointer deflects. The principle of operation is similar to that of the varying-pressure feed-water regulator in Chap. XIII, Fig. 13–8.

CHAPTER II

PRESSURE ELEMENTS

Measuring pressure is one of the most necessary functions in power and industrial plants. A pressure, or vacuum, gage is likely to need special attention in many processes because its vital measuring element is usually exposed to the fluid whose pressure is being measured. For instance, the fluid may seriously corrode the measuring element, solidify at ambient temperature, or deposit heavy compounds that plug the line. Moreover, violent pressure pulsations may destroy accuracy and cause serious wear in linkages and delicate gears as well as fatigue failure of the elastic member.

The many instruments that meet varied plant needs are classified as having pressure that (1) acts on a known area, as in liquid-column manometers, bell gages, and spring- or weight-loaded piston gages; (2) produces deformation in an elastic member, as with bourdon tubes, diaphragms, and bellows; or (3) causes change in a body's physical property, as with hot-wire and piezoelectric gages.

The simplest and oldest way to measure pressure is to balance it against a resisting force whose magnitude can be measured. For example, water weighs 62.4 lb per cu ft; and, since pressure is usually measured in pounds per square inch, you divide 62.4 by 144 to find that a 1 sq in. column of water 1 ft high weighs 0.434 lb. This is the pressure it exerts at its base. If you insert a glass tube in a pail of water and suck on the open end until water rises to a height of 1 ft in the tube, it indicates that atmospheric pressure acting on the water surface in the pail is 0.434 psi greater than that acting on the water surface in the tube.

Most pressure gages register the difference between two pressures (atmospheric and the one being measured) and are, therefore, differential gages, although this term usually applies only to instruments operating on the U-tube principle—those measuring the difference between two pressures, both separate and distinct from atmospheric (Fig. 2-1).

The liquid column (Fig. 2-2) is the simplest method of measuring pressure and serves as a standard to check the accuracy of other pressure-measuring instruments. It consists of a tube filled with a liquid that exerts a pressure at the tube base that varies directly with the liquid-column height *H*. Mercury serves for vacuum and high pressures or pressure differentials; and kerosene, glycerin, water, alcohol, or other liquid that is not too volatile is used for low pressures.

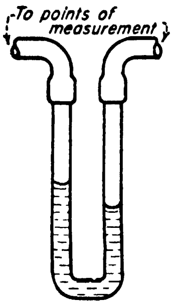


FIG. 2-1.—Simple U tube that measures differential pressure between two points.

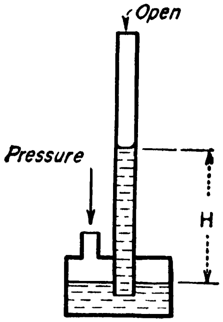


FIG. 2-2.—The liquid column is a standard to check other measuring instruments.

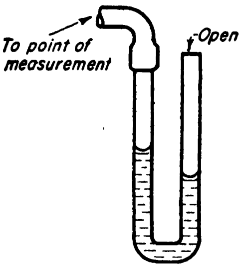


FIG. 2-3.—U tube containing water serves to measure draft or check draft gages.

The lowest pressure that can be measured by a liquid column is that of the vapor pressure of the liquid, the upper limit being set only by the practical working height of the column. For high pressures where extreme height becomes inconvenient, several short columns can be connected in series. Here a column of less dense liquid (water) transmits pressure from the top of one mercury column to the bottom of the next. The total pressure reading equals the sum of mercury-column heights minus the sum of water-column heights.

In an open U tube (Fig. 2-3), the difference in height of the two liquid legs is a direct measure of the pressure (in comparison with that of the atmosphere) acting at the point of attachment to the tank, pipe line, or other vessel.

The well gage (Fig. 2–2) differs from the U tube only in that one leg is replaced by a chamber that is made large so that the liquid in it remains at practically a constant level regardless of how high it stands in the tube. For precision work, consideration must be given to the change in chamber level and curvature (meniscus) caused by the surface tension of the liquid in both the tube and the well (Fig. 2–4).

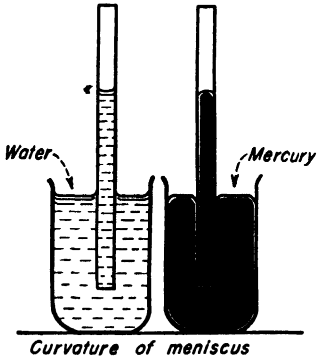


FIG. 2–4.—Shape of meniscus depends on whether or not tube is wetted by the liquid used as the measuring fluid.

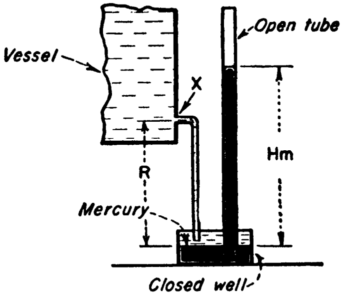


FIG. 2–5.—Liquid head *Hm* measures pressure acting at point *X* when densities of fluids in legs *R* and *Hm* are known.

Determining vessel pressure with a liquid column is easy when the density of the manometer fluid and the fluid being measured is known, together with the tube level and the distance between the point of measurement and the well level. Using Fig. 2–5, the equation is

$$(Hm \times pm) - (R \times pa) = P$$

where P = pressure at X, psi.

Hm = height of mercury column, in.

R = height between mercury surface, and point of pressure measurement, in.

pm = density of mercury, lb per cu in.

pa = density of fluid being measured, lb per cu in.

When the pressure of air or other gas having negligible weight is being measured, the *Rpa* factor can be disregarded except for precision work. For a liquid or a gas that requires a sealing liquid, however, the complete equation must be followed.

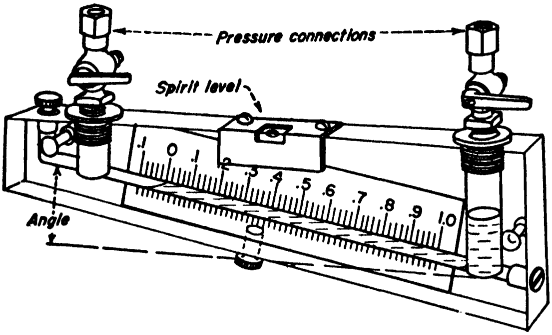


FIG. 2–6.—The inclined gage filled with a light liquid measures static pressure in air-conditioning systems or draft in furnaces. (*Courtesy of F. W. Dwyer Mfg. Co.*)

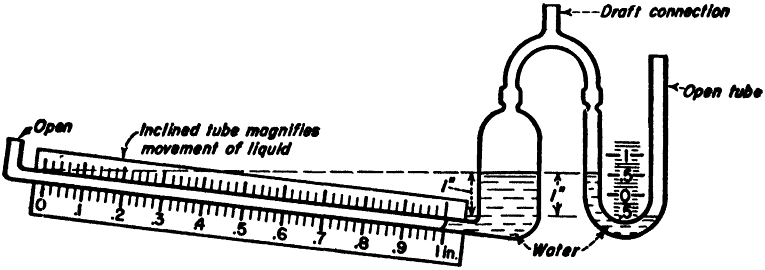


FIG. 2–7.—The inclined tube magnifies vertical level movement so that small increments of pressure can be measured.

To obtain the required precision when measuring small pressures, the reading can be magnified by (1) using a fluid of different density, (2) substituting an inclined U tube, (3) providing one leg with a reservoir of much larger bore than the tubing, or (4) using a two-fluid U tube.

For measuring boiler draft and static pressure in air-conditioning system, the inclined tube (Fig. 2–6) serves to advantage because an increase or decrease in vertical liquid height (Fig. 2–7) causes the level to move in the inclined tube a distance equal to (vertical liquid height) × (cosecant of angle sloping tube makes with horizontal position). Thus, if the angle's cosecant is 10, a change in vertical liquid height of 1 in. shows up as 10 in. in the inclined tube. The gage can be graduated to 0.01 in. For low-pressure service, an oil considerably lighter than water is used as the filling agent. This permits lengthening the scale still farther. These instruments are provided with a spirit level to assure correct positioning of the tube. A connection at either end facilitates using the tube to measure pressure above or below atmospheric or a pressure difference between two points. This device is not readily adaptable to recording duty, however.

A liquid-sealed bell (Fig. 2–8) can be used in place of the inclined U tube to indicate, record, or control pressure. In a single-bell unit, a spring, weight, or float balances the bell, and the pressure to be measured is applied to its underside. The bell movement then positions a pointer, pen, or controller. The double-bell device in Fig. 2–9 measures differential pressure, one of two pressures being applied under each bell. The instrument becomes a recording flowmeter when connected across a boiler pass or between the furnace and the last pass, the scale being calibrated accordingly. Frequently a gas-temperature thermometer and steam-flow indicator are mounted inside the same case so that the chart records steam flow, air flow, and uptake temperature.

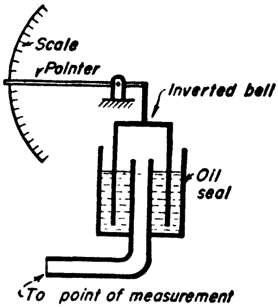


FIG. 2–8.—The single inverted bell measures low pressures if properly sealed with liquid.

A closed column (Fig. 2–10) measures absolute pressure directly, provided that the space in the closed end is substantially a perfect vacuum. A typical example is the barometer, an instrument that measures absolute pressure of the atmosphere in terms of mercury-column height. Normal barometric pressure equals 29.92 in. (760 mm) of mercury at 32 F. For accuracy, the reading must be corrected for temperature—because of thermal expansion of the mercury and the glass or metal tube, altitude above sea level, and instrument calibration. This is particularly important when measuring low absolute pressures with vacuum indicators (Fig. 2–11) on steam-condenser tests. A mercury gage (Fig. 2–12) has a center zero for measuring pressures both above and below atmospheric pressure.

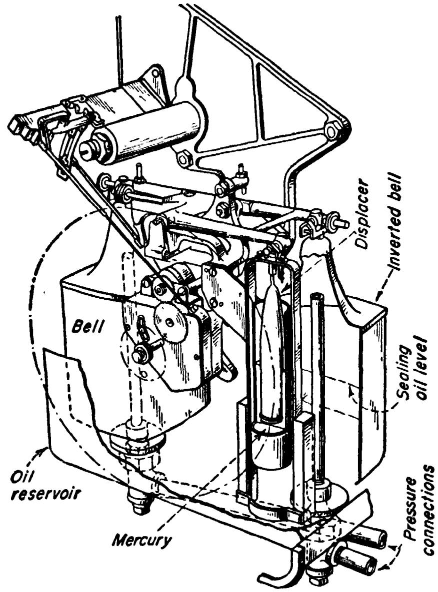


FIG. 2–9.—A balanced set of inverted metal bells measures differential pressure. (*Courtesy of Bailey Meter Co.*)

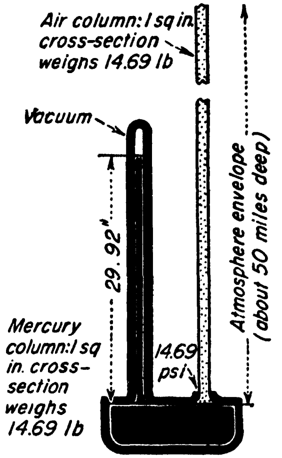


FIG. 2–10.—A 1 sq in. column of mercury 29.9 in. high is balanced by a column of atmosphere having the same cross-sectional area.

The differential U tube (Fig. 2–13) and inverted tube (Fig. 2–14) measure the pressure difference between taps *A* and *B*. The inverted tube, usually filled with gas, serves to measure liquid-pressure differentials when open columns would be too high or where the liquid cannot be exposed to atmosphere. Valve *X* permits varying the quantity of fluid in the gage.

The two-fluid manometer (Fig. 2–15) is more sensitive than a simple U tube for small gas pressures, because it indicates measurements over a broader scale for the same pressure differential. Best results are obtained when the specific gravity of one liquid is only slightly greater than that of the other. Figure 2–16 shows the difference in mercury elevation when two tubes, one containing water and the other air, are subjected to the same pressure differential.

Liquid-column manometers are subject to an error in reading if the internal diameter of the tubing is too small, because the meniscus is quite pronounced. This is especially true if the areas of the two surfaces involved in the measurement are widely different. Bore variations also introduce errors. For these reasons, tubing below ¼ in. ID (inside diameter) should not be used.

The piston gage (Fig. 2–17) serves for pressure ranges from 5 psi up and consists of a rotatable close-fitting piston riding in a vertical cylinder and mounting a weight platform on its upper end. The piston must be rotated when the gage is used, to eliminate friction errors. At low pressures, the inertia of the light piston is so small that it is difficult to maintain rotation. On higher pressures, piston diameter must be kept small in order to avoid excessively heavy weights. The latter difficulty is avoided in the device shown in Fig. 2–18 by the use of a two-diameter piston.

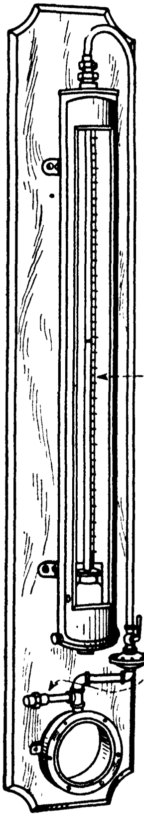


FIG. 2–11.—Mercury column shows steam condenser pressure in terms of vacuum.

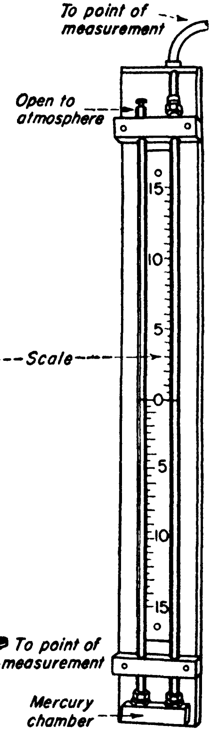


FIG. 2–12.—Center zero gage measures pressures above or below atmospheric.

The piston principle is used in (1) pressure-gage testers, (2) engine indicators, and (3) transmission of pressure readings. In the tester (Fig. 2–19), it produces a known pressure for checking bourdon-tube gage accuracy. Here the piston carrying a known weight produces a known pressure in the cylinder communicating with the pressure gage. The screw-operated piston *X* is used to force liquid into the cylinder to raise the pressure piston. To avoid friction effects, the weight table is rotated slowly during the test.

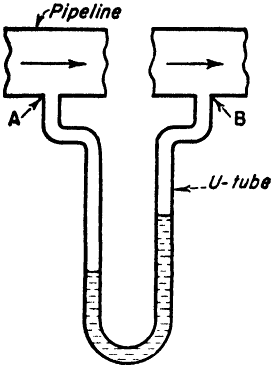


Fig. 2-13.—Liquid-filled U tube measures pressure difference between points *A* and *B*.

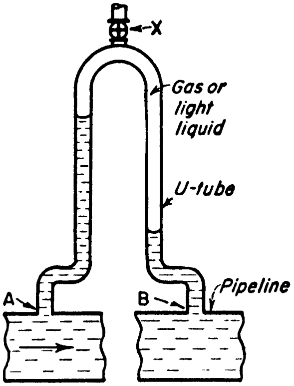


Fig. 2-14.—Gas-filled inverted U tube measures the pressure drop in liquid-filled line.

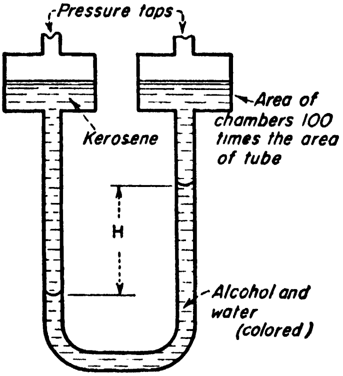


Fig. 2-15.—Using two fluids of slightly different specific gravity gives a greater level change for a given pressure value.

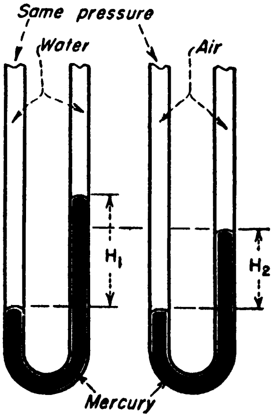


Fig. 2-16.—Action of a two-fluid tube that uses water and air separately with mercury.

In an engine indicator (Fig. 2–20), the spring-loaded piston is acted upon by pressure inside the engine cylinder. The piston moves a pen that writes on a paper attached to the movable drum. A mechanical connection between the engine piston rod and the instrument drum through a reducing motion gives an oscillating

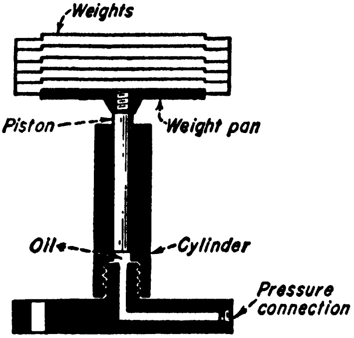


Fig. 2–17.—Simple piston gage with weights that balance actuating pressure.

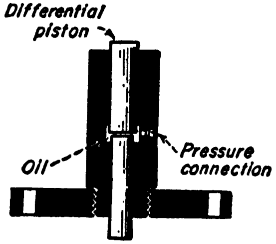


Fig. 2–18.—When measured pressures are high, a differential piston can be used to reduce the size of weights.

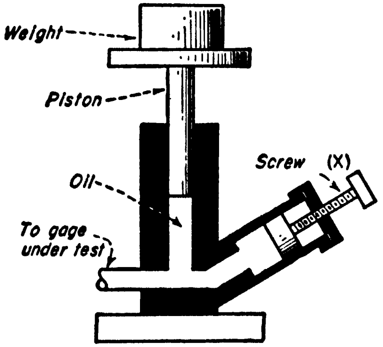


Fig. 2–19.—Adaptation of piston gage for checking conventional bourdon-tube pressure gage.

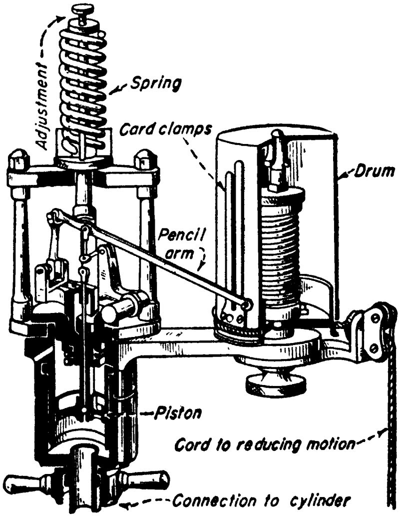


Fig. 2–20.—Engine indicator (spring-loaded) uses piston-gage method.

movement to the drum as the engine operates. Thus the pen and drum movement produces a visible record of varying cylinder pressure throughout the stroke.

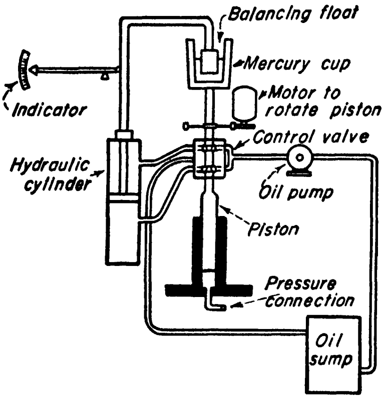


FIG. 2-21.—Piston-gage transmitter with motor-rotated piston weight.

When this principle is used to transmit pressure readings, the piston is rotated by an electric motor. A change in pressure moves the piston to operate a pilot valve that controls the oil flow to a piston in another cylinder (Fig. 2-21). This cylinder positions a float in a mercury cup carried by the pressure piston. Moving the float varies the load on the pressure piston to return it to its original position. The float height is telemetered to a distant point and recorded as pressure.

In another automatically balanced piston gage (Fig. 2-22), the pressure on the piston is opposed by a scale beam. Its movement closes high or low contacts to operate a reversing motor that balances the beam by sliding the poise. Pressure-piston instruments can be adjusted to indicate measurements over a narrow range—on a 750 psi boiler, it can be arranged to indicate from 725 to 775 psi.

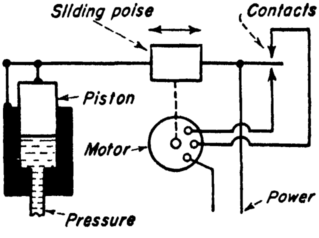


FIG 2-22.—Piston moves contact arm and motor rebalances poise.

A bourdon-tube gage (Fig. 2-23*A*) operates by deflection of an oval tube closed at one end and bent in the shape of a circle. The tubes are made of various metals for different pressures. When used to measure the pressure of hot or corrosive fluids, the substance must be kept out of the tube by a suitable seal (Fig. 2-28).

Tube movement operates a pointer through a toothed sector and pinion (Fig. 2-23*A*) or a helical groove shaft (Fig. 2-23*B*). It is customary to select gages that operate near the middle of their scale, because continuous operation near full-scale reading or at overpressure may upset the tube metal so that it becomes inaccurate at lower readings. The helical element (Fig. 2–24) and spiral element (Fig. 2–25) are similar to the single tube in Fig. 2–23 except that their greater length gives broader pointer deflection for a given pressure change.

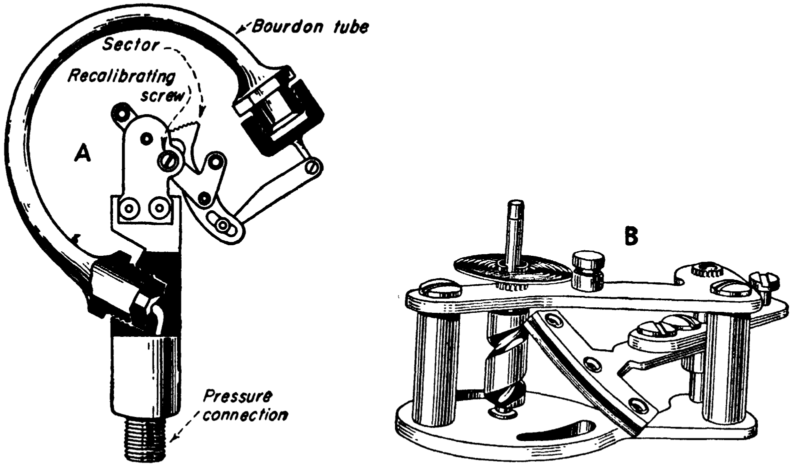


FIG. 2–23.—Conventional half-turn bourdon tube for pressure gage. Tube end moves gear sector and pinion or helical shaft. (*Courtesy of J. P. Marsh Corp. and American Chain & Cable Co.*)

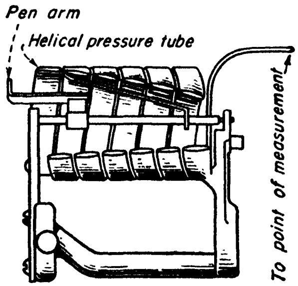


FIG. 2–24.—Winding the tube in the form of a helix also offers the advantage of greater pen deflection.

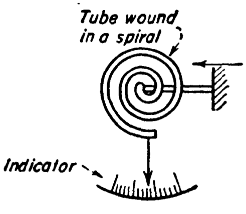


FIG. 2–25.—Spirally wound bourdon tube produces a greater pointer deflection for a given pressure change.

Figure 2–26 explains why the tube tends to change its curvature when it is subjected to internal pressure. When pressure is applied to an elastic oval tube *A*, it tends to become circular, *B*. Hence, when the cross section tends to change from oval to circular, the outer circumference of the ring increases from C_1 to C_2 and puts the metal under tension. The inner circumference also decreases from D_1 to D_2 and puts the metal under compression. This action on the ring is such as to move it toward a straight shape *E* and position the pointer.

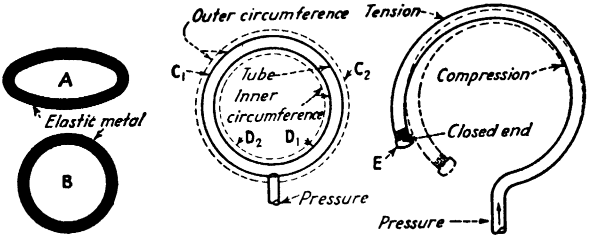


FIG. 2–26.—An oval-shaped tube *A* tends to assume the shape of a circle *B* when pressure is applied to it internally. If the tube is bent in the shape of a hook with one end anchored, pressure moves free end in a direction to straighten out the hook *E*.

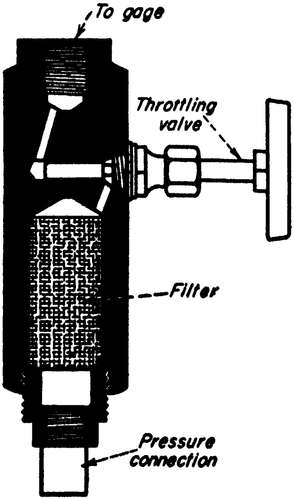


FIG. 2–27.—Snubbers throttle pulsating pressures and minimize damage to gages.

Bourdon-tube gages are available in these designs: (1) compound pressure and vacuum, (2) gage with suppressed zero, (3) magnified graduations over part of the scale, (4) double-tube gages, and (5) multiple-tube instruments.

Pulsating pressures that soon destroy the accuracy of the sector-and-pinion movement by wearing the teeth and bearings can be minimized by throttling the shutoff valve or installing a pressure snubber (Fig. 2–27). Other designs use (1) a moving piston whose inertia prevents rapid pulsation, (2) a large number of small tortuous passages to smooth out pulsations, or (3) a separating diaphragm and a gage-filling liquid that must pass through a felt obstruction.

Diaphragm gages serve the same purpose as bourdon-tube elements, although they are not employed for high pressures. Distortion of the diaphragm, which is clamped inside a housing, moves a pointer or other indicator through a linkage connected to the diaphragm center. Corrugating a metal diaphragm gives it about four times as much deflection as a flat one for the same pressure change. In general, diaphragm gages provide larger force than bourdon tubes for actuating indicating and recording devices (Fig. 2-29).

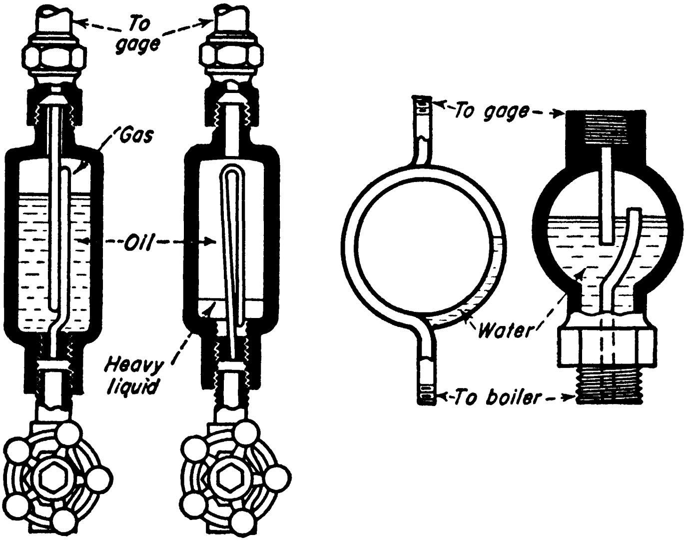


FIG. 2-28.—Liquid sealing chambers filled with a suitable fluid keep corrosive or hot substances out of the pressure gage. Be careful when mounting to see that they retain the filling agent.

Limp- (flexible) diaphragm instruments use soft elastic material such as specially selected leather, treated cloth, or rubber. The diaphragms are spring-loaded. The unit in Fig. 2-30 can be connected to read pressure above or below atmospheric or pressure differential. The diaphragm is carried by the drive linkage and is opposed by a leaf spring. Turning the adjusting screw moves the bell crank about its pivot and shifts the calibrated spring linkage to adjust the instrument pointer to zero. Stops are provided to prevent jamming of the pointer by overdeflection.

For measuring differential pressure, the higher pressure is applied to the chamber containing the spring and the lower pressure is applied to the left side of the diaphragm. Connecting the gage across the furnace and the last pass of the boiler makes it an indicating flowmeter for low static pressure.

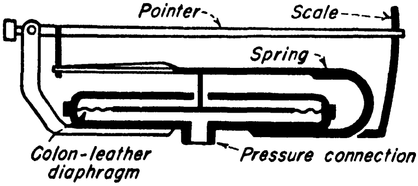


FIG. 2-29.—A limp diaphragm made of flexible leather or rubber operates a pointer to measure positive or negative pressures.

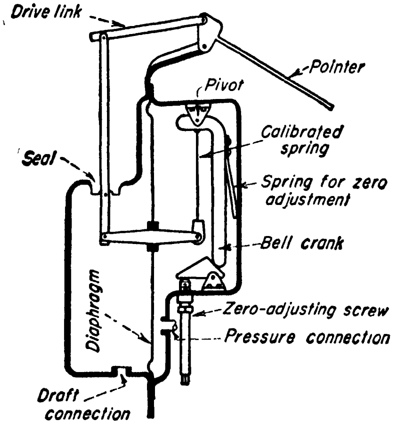


FIG. 2-30.—Limp-diaphragm gage adaptable for measuring positive or negative pressure or furnace pressure differential.

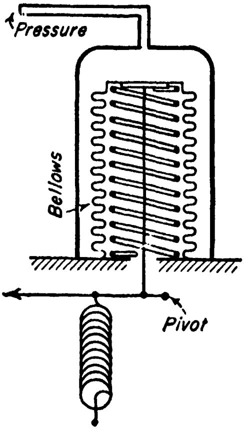


FIG. 2-31.—Spring-loaded bellows serves for measuring pressure.

The metallic bellows (Fig. 2-31) provides an element sensitive to low pressure, able to withstand fairly high pressure, and yet powerful enough to operate recording and indicating mechanisms.

Bellows are formed from (1) tubing by rolling, spinning, or hydraulic forming; (2) corrugated disks soldered or welded together; and (3) solid rods machined to shape. In practice, the bellows is spring-loaded, and deflection characteristics are the resultant of those of the spring and bellows. Metals used for fabrication are (1) brass, (2) phosphor bronze, (3) beryllium copper, (4) copper-nickel alloys, (5) mild steel, and (6) monel metal.

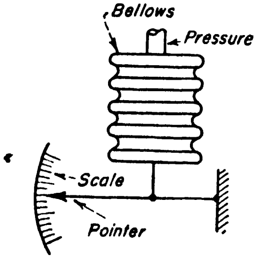


Fig. 2-32.—Long bellows may buckle when pressure acts inside.

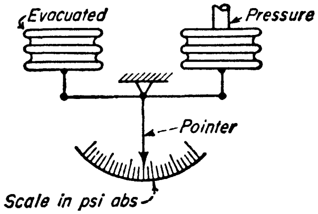


Fig. 2-33.—Double bellows (one unit evacuated) reads absolute values when pressure acts inside the remaining unit.

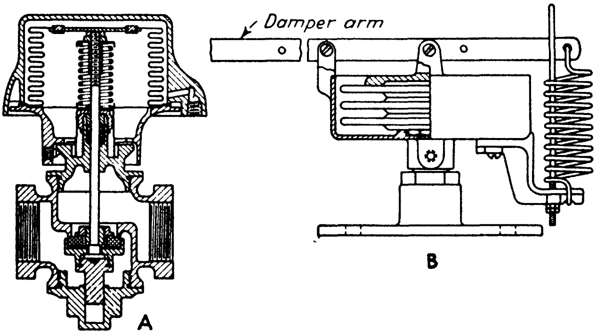


Fig. 2-34.—Bellows arranged to operate a valve and ventilating damper. (*Courtesy of Johnson Service Co. and The Powers Regulator Co.*)

A bellows that is long in proportion to its diameter tends to buckle when pressure is applied internally (Fig. 2-32) and a load applied at one end, since under this condition each convolution tends to get longer. Best design applies the pressure externally

(Fig. 2–31) so that each convolution gets shorter and pulls on adjacent ones to keep the bellows straight. The double-bellows unit (Fig. 2–33), with one side evacuated to as near a perfect vacuum as practicable and the other connected to the pressure to be measured, registers absolute pressure with reasonable accuracy. Atmospheric pressure affects both bellows alike and therefore cancels itself out. Figure 2–34 shows a bellows arranged to operate a valve *A* and ventilating damper *B*.

The hot-wire pressure gage uses an electric-bridge circuit with two resistance elements heated by an electric current. One element is exposed to the atmosphere and the other to the air space being measured. A change in the density of the atmosphere surrounding the exploring element changes the heat conductivity and therefore the temperature of this element. Thus its resistance changes and upsets the bridge balance.

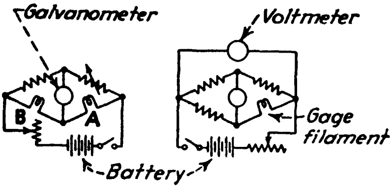


FIG. 2–35.—Pirani gage uses Wheatstone bridge.

The Pirani gage (hot-wire instrument) operates on the physical principle that the thermal conductivity of a gas is related to its pressure. The instrument measures the thermal conductivity of the gas indirectly by ascertaining the change in resistance of a hot filament placed in the pressure area. As the pressure changes, the speed at which heat is conducted away from the filament varies; this in turn changes the filament temperature and hence its resistance. These variations are recorded by a Wheatstone-bridge circuit (Fig. 2–35).

One method of operation keeps the heating current constant and measures the change in resistance as the pressure is varied; another keeps the temperature and therefore the resistance constant and measures the change in voltage as the pressure is altered. The balancing resistance, in the bridge arm opposite to the gage filament *A*, is an identical filament *B* sealed off in a bulb at the lowest possible pressure. It compensates for variations outside the measuring circuit.

CHAPTER III

HEAD METERS FOR LIQUID LEVEL, PRESSURE, AND FLOW

In order to transmit indications and control the level of a liquid a primary element is required that actuates a recorder through a telemetering circuit or an automatic valve through a telecontrol system.

The primary element, a pickup that actually transfers the level to a transmitting device, consists of a *float, series of electrodes, capacitor, pressure dome, diaphragm box, bubbler,* or *differential-head meter.* These can be either (1) mechanically or directly connected or (2) arranged for remote transmission—either electrical, hydraulic, or pneumatic. Direct-connected units have the indicating and actuating mechanism integral with the primary device, and the instrument must therefore be placed immediately adjacent to the vessel. A remote-transmission system consists of a separate primary element adjacent to the vessel and connected to a distant valve or recorder through a suitable control system. Float actuators are described in Chap. I.

These instruments fit applications where the liquid is in open tanks and, with few exceptions, in closed pressure vessels. Every kind of liquid can be indicated or controlled, whether it be flowing, turbulent, quiescent, corrosive, carrying suspended matter, or covered with ice.

Electrode units depend upon current conduction through the liquid for their operation. Hence the device is limited to fluids that are electrically conductive. In general, the construction consists of a current-responsive electric relay connected in the circuit of one or more insulated electrodes extending into the liquid chamber. Current from the electrodes flows through the liquid to the metal vessel and back to the relay. The electrode voltage ranges from 24 to 100 volts; current seldom exceeds 0.10 amp. The single-electrode unit in Fig. 3–1 serves as a low-water cutout.

The immersion-electrode device (Fig. 3–2) consists of an *A*-shaped relay with a laminated core. Primary coil *X* is wound on the upper bar of the core and a secondary coil *Y* on the lower bar. An armature lying below the core legs connects to an insulated arm carrying two contact bridges. When the armature is raised, these bridges close or open the motor and electrode circuits, depending upon whether the contacts are normally open or closed. The holding-circuit switch *A* closes the circuit between the two electrodes, causing the longer electrode to become an extension of the shorter. Switch *B* closes the circuit to an indicator, alarm, or control device.

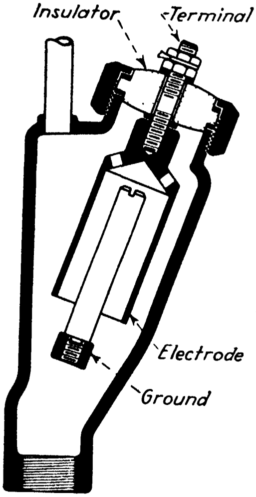


FIG. 3–1.—Electrode arrangement for low-water cutout.

When alternating current is connected to terminals 1 and 2, the primary coil sets up a magnetic flux that, following the lines of least resistance, circulates through the shortest path or through the bar of the lower coil. This magnetic flux induces a voltage in the secondary, or electrode-circuit, coil, but no current flows unless the circuit is closed.

Closing of the secondary circuit, which occurs when liquid reaches the upper electrode, sets up a bucking action in the lower bar of the core because current then flows in this circuit. This bucking action diverts lines of magnetic flux to the core legs to raise the armature. Immersion-electrode devices are available that have a series of electrodes.

The capacitor transmitter (Fig. 3–3) operates on the radio principle and consists essentially of a vacuum-tube and relay assembly and capacity pickup. The vacuum tube, maintained in an oscillating condition, generates high-frequency alternating current. This current is conducted to the gage glass or immersion pickup unit by a coaxial cable consisting of a circular metal braid filled with fish-spine insulators that carry a central conductor. Any object entering the pickup field detunes the oscillator, and the resulting change in current through the relay operates its contact.

The electrical properties of a liquid can be either conductive or dielectric as compared with the capacities of two electric condensers, one filled with liquid and the other empty. A pickup condenser so constructed changes its capacity upon the approach of a liquid. The liquid level itself may form one plate of the condenser, whose capacity

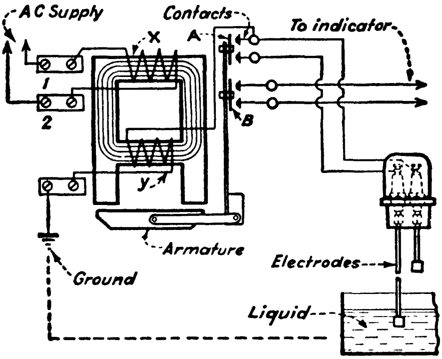


FIG. 3–2.—Liquid contacting the upper electrode short circuits coil *Y*. (*Courtesy of The Johnson Corp.*)

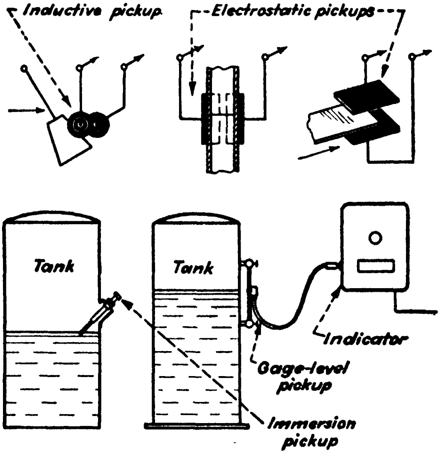


FIG. 3–3.—Inductive or condenser capacity varies with liquid level. (*Courtesy of Wheelco Instrument Co.*)

changes as a result of varying the distance between the plates or the mutually exposed surfaces of the plate. In another arrangement, the capacity changes because of liquid entering the field of a two-plate condenser. A level-sensitive condenser or pickup forms part of a high-frequency resonator that connects to a transformer, electronic tube, and relay.

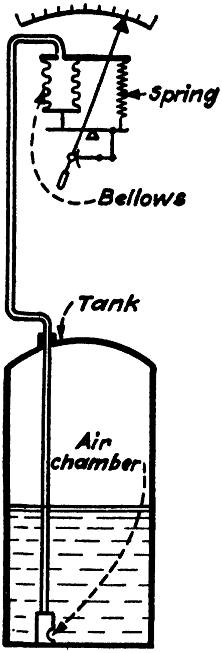


FIG. 3-4.—Air trapped in the lower chamber exerts pressure inside the bellows equal to the liquid height. *(Courtesy of The Liquidometer Corp.)*

The gage-glass pickup consists of a bakelite body, approximately 2 in. square, containing the resonator, and two separable cylindrical plates —a ground plate and a sensitive plate. These plates clamp to the gage glass with a thumbscrew. The immersion pickup unit consists of a tubular body, constructed like the coaxial cable so that it serves to extend the cable; a tubular resonator; and a sensitive tip. The unit is never used bare, but always inside a suitable dielectric well or protecting tube.

Since the liquid in an open vessel or tank exerts a pressure on all points within the container, devices that operate on pressure can be used to indicate liquid level. Allowing for the specific gravity of the liquid, a direct relation exists between the pressure and the depth, or static head, at the point of liquid level. Although measuring the height of the liquid is quite simple with this method, volumetric measurement (gallons or cubic feet) requires special calibration for spherical and horizontal cylindrical tanks. For square and vertical cylindrical tanks, each inch of increase in liquid height equals the same volume of liquid.

The simplest method of measuring height by pressure is to connect an ordinary pressure gage directly to the vessel. In this case, the center line of the gage must be on the same level as the minimum liquid level so that the gage will register zero when the tank supply is at minimum. The gage must have a range equivalent to the static head when the tank is full.

Other devices that utilize a pressure gage are the *pressure dome, diaphragm box,* and *bubbler,* or *purging system.* In the pressure-dome unit (Fig. 3-4), air trapped under the dome and in the tubing and bellows is compressed as the liquid level rises until its pressure equals that of the liquid head. Because the liquid absorbs air, it must be replenished occasionally with a hand pump or from a compressed-air source.

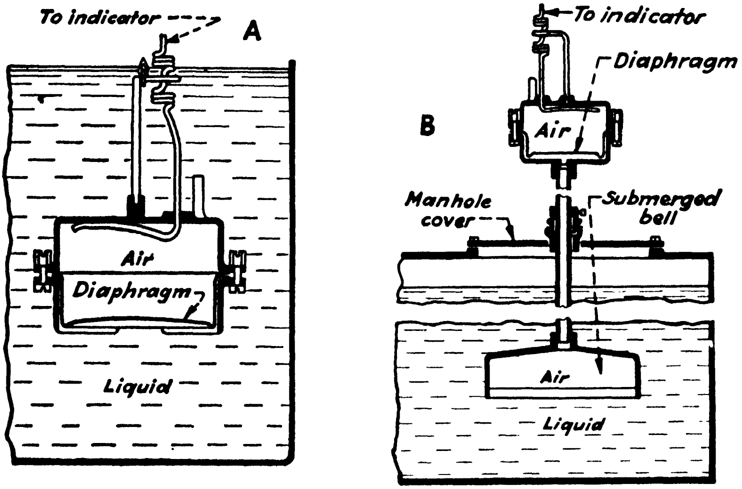


Fig. 3-5.—Airtight diaphragm compresses the air in connected tubing and instrument as level increases. (*Courtesy of Foxboro Co.*)

Diaphragm-box units (Fig. 3-5*A*) consist essentially of an airtight diaphragm connected by an air-filled tube to a pressure-measuring instrument that may be a recorder, indicator, or controller. Pressure exerted by the liquid at the diaphragm compresses the entrapped air in the system and transmits pressure to the instrument, where it is measured in terms of liquid height. Since air is the actuating medium, the pressure gage (but not the diaphragm box) can be mounted at any elevation in relation to that of the tank. For quick response the tubing should not be more than 250 ft long.

When such a unit is installed in liquids containing suspended solids, a flushing line is connected to the center of the perforated bottom of the diaphragm box (Fig. 3–6*A*). This facilitates flushing solid particles from under the diaphragm. In the closed-box system (Fig. 3–6*B*), flushing is accomplished by placing a valve and tee in the connecting line between the vessel and box. Thus, by closing the valve, unplugging the tee, and connecting the flushing hose to the box vent hole (normally plugged), the sediment is forced out the tee discharge.

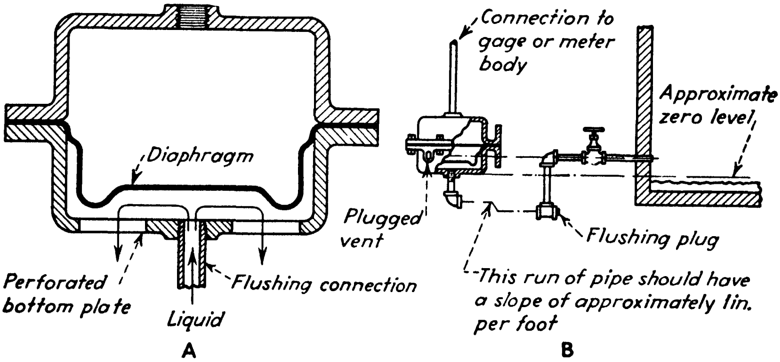


FIG. 3–6.—Flushing connections for open- and closed-diaphragm boxes.

Diaphragm boxes are usually of acid-resisting bronze tinned on the inside of the upper section. As additional protection, the bronze can be lead-wiped. Special cast-iron or stainless-steel boxes serve for acid corrosive to bronze. For chlorine, the boxes are coated inside and out with silver. If the liquid to be measured will injure the diaphragm, an extension (Fig. 3–5*B*) mounts it above liquid level. This makes a combination pressure-dome and pressure-diaphragm instrument, and the dome air must be replenished occasionally. To work satisfactorily, the diaphragm box, tubing, and instrument must be leakproof.

The bubbler gage (purging system) is particularly well suited for liquids carrying solid matter. A small amount of air flowing into a pipe submerged in the liquid displaces liquid in the pipe. A gage measures the air pressure, which is directly proportional to the liquid head above the pipe terminal, and records it in terms of liquid level (Fig. 3–7). Compressed air held at constant pressure by the regulator flows through the sight-feed glass on its way to the tank. The height of the water above the measuring pipe *A* determines the rate of bubbling, which in turn controls air pressure in the gage line. This pressure acting on the gage indicates the liquid level. The consumption is about 5 cu ft of free air per hr.

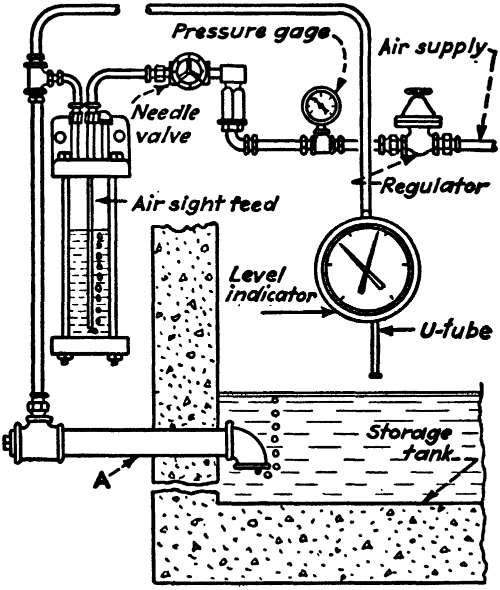


FIG. 3–7.—Compressed air flows through a sight glass on its way to measuring pipe. Height of liquid determines the resulting pressure. (*Courtesy of Bailey Meter Co.*)

Assume that the tank liquid is at minimum level and just sealing off the measuring-pipe opening. The air supply is then adjusted at the regulator to a pressure slightly greater than the maximum range of the instrument. The needle valve is set to pass 15 to 20 air bubbles a minute through the sight feed. With this setting, the pressure gage should read zero because the bleeding air is passing through a negligible liquid head. Once the air pressure and feed are set and the pressure gage is adjusted for zero at minimum liquid level, the adjustments should be left alone.

This system serves to measure almost all liquids, because the air or other gas bleeding from the measuring pipe acts as a purging medium to keep the line open. It is adaptable to corrosive liquids or to those containing suspended solids. Also, the pressure-gage location is not critical; it can be mounted above or below the tank.

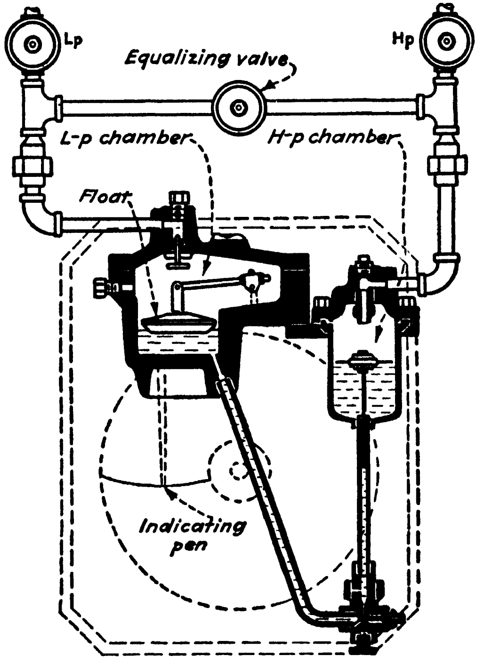


FIG. 3–8.—Mercury level changes with the differential pressure to operate a float attached to the meter mechanism. (*Courtesy of Taylor Instrument Cos.*)

Differential-head manometers operate on the U-tube principle described in Chap. II and can be used to measure liquid level and flow as well as pressure. Figure 3–8 shows the construction of a commercial instrument complete with a float mechanism and mercury as the measuring fluid. Manometer design varies to some extent according to whether the float is located in the high- or in the low-pressure chamber. To see how the unit operates, assume that the high-pressure leg connects to the tank, that the low-pressure one is open to atmosphere, and that the instrument registers zero liquid level in the tank. Then, as the liquid level rises, pressure across the U tube becomes unbalanced, and mercury is forced over into the low-pressure chamber, where it raises the float to a new position to indicate the new level. The design is such that, for a change from zero to full level, the metal float moves its pen or controller arm through the full scale.

The unit in Fig. 3–9 eliminates the float and levers by permitting the mercury to rise and make contact with a helically arranged series of resistance electrodes to vary the metering current to a receiving instrument.

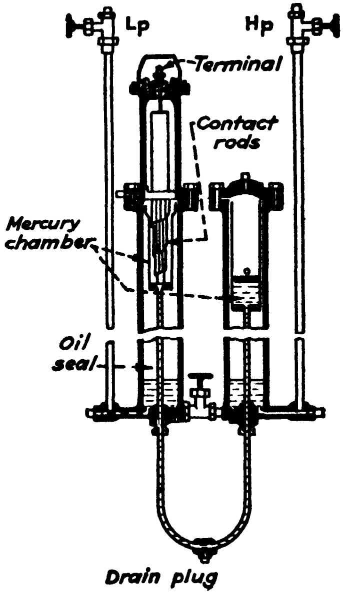


FIG. 3–9.—Instead of moving a float, the mercury moves over contact rods to vary the electrical resistance of the metering circuit. (*Courtesy of Republic Flow Meters Co.*)

When the manometer is mounted at some distance below the minimum water level in the tank, additional mercury must be put in the leg open to atmosphere to balance the liquid head from tank to manometer, or the device will not register zero at minimum liquid level. This method is more convenient than installing a riser on the atmosphere leg and filling it with the tank liquid to a height equal to that in the measuring leg.

Also, on high water towers, where only the level variation in the tank is to be indicated, some means must be provided to suppress the instrument zero so that the static head of liquid in the supply pipe (from ground level to the tank bottom) will not be registered on the indicator. Since the suppressed head in this case is considerably greater than the measured head, it is necessary to apply external pressure to the manometer when adding the mercury (Fig. 3–10).

To accomplish this, first fill the manometer with the standard amount of mercury; then add water, or sealing liquid if the instrument is to be used on corrosive fluids, to the high-pressure connecting piping. Apply air pressure to this leg until the instrument reads full scale. Then pour additional mercury into the low-pressure leg until the instrument returns to zero. Repeat the operation until the amount of mercury specified for the required suppression has been added and the instrument reads zero. After the adjustment is correct, close the high-pressure valve, disconnect the air supply, and connect the manometer to the tank.

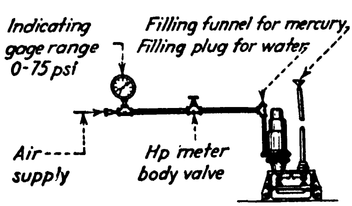


FIG. 3–10.—To operate under a suppressed head, the manometer must be filled under pressure. (*Courtesy of Brown Instrument Co.*)

To guard against loss of mercury on suppressed-zero manometers, install a mercury trap in the pipe going to the tank. Then, if the tank is drained or some other event causes loss of static head between the tank and the manometer, mercury will be caught in the trap instead of flowing over into the distribution system (Fig. 3–11).

A device made especially for installation on water towers and other overhead tanks is the counterpoise gage (Fig. 3–12). The dial can be calibrated for tank depth or for the working range of the liquid level. The instrument consists of a U tube of considerable length and can be connected at any convenient point near the tower feet. The pressure effect of water in the pipe riser is balanced and eliminated by the mercury counterpoise.

Differential-head meters also serve to measure the liquid level in pressure vessels. When used in this way, both manometer legs connect to the pressure vessel in such a way that, as the tank liquid level moves from the full position, the differential pressure acting across the manometer body is increased.

Indications are obtained by transmitting the variable liquid head to the low-pressure side and maintaining a fixed liquid head on the high-pressure side by a special piping arrangement. Static pressure in the two legs cancels out. Figure 3–13 shows the installation on a boiler drum. Devices of this type employ mercury or a proprietary fluid and a diaphragm or bellows measuring element.

Because these manometers operate under differential head, a difference of specific gravity in the liquids of the two legs causes inaccurate indications or control that must be compensated for in some manner. The piping scheme in Figs. 3–13 and 3–14 serves to minimize or eliminate the effect of specific-gravity variations between the two legs caused by temperature differences. Here the temperature of the high-pressure leg is kept about the same as that of the low-pressure one by extending one tube up inside the other. To make the system fully effective, the outer chamber must be completely insulated.

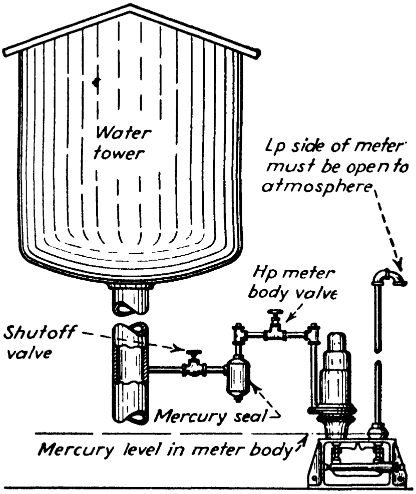


FIG. 3–11.—A sealing chamber prevents loss of mercury if head pressure fails.

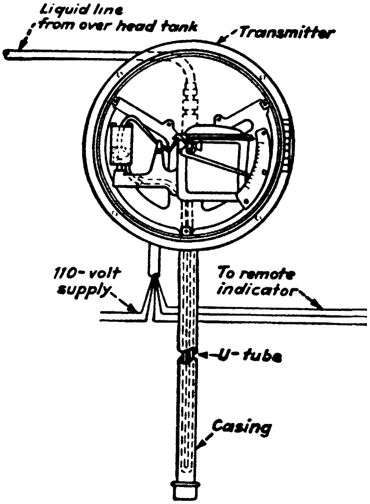


FIG. 3–12.—The counterpoise gage serves for overhead tanks requiring a suppressed zero. (*Courtesy of The Bristol Co.*)

To illustrate the effect of different densities in the two legs, assume that the main vessel contains oil having a specific gravity of 0.89 at 0 F and 0.72 at 450 F. The vessel temperature is 450 F and the outside-air temperature is 0 F. The tank is fully insu-

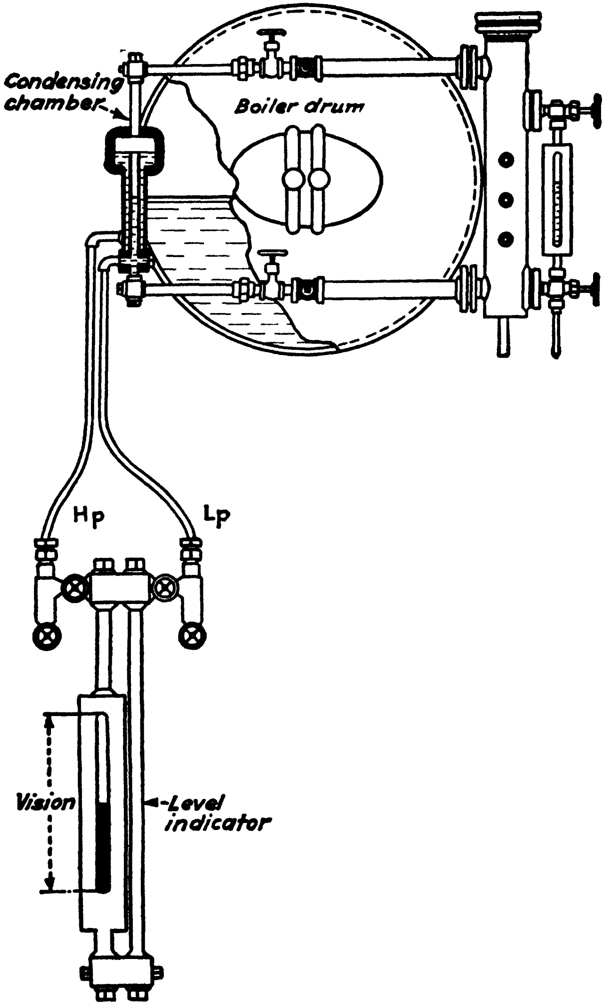


Fig. 3–13.—U tube with proprietary fluid registers boiler drum water level. (*Courtesy of Reliance Gauge Column Co.*)

lated, but the outer leg is exposed to the weather. No sealing liquids are used, and the material in the outer leg is the same as that in the vessel. Since the distance between taps is 100 in. the outer leg has a weight equal to 0.89 × 100 = 89 in., and the weight of the inner leg is 0.72 × 100 = 72 in. The problem is to compensate for this difference in weight. Filling both legs with a suitable sealing liquid keeps them both at the same density, but correction must still be made for any difference between the specific gravity of the filling liquid and that of the liquid in the vessel.

Under this condition, the two legs of a mercury manometer can be balanced by adding more mercury to the low-pressure leg; but this method does not permit zeroing and checking the meter in the customary manner, because additional mercury has been added. Some manufacturers provide a sensitivity slider on the float arm to cancel the effects of difference in density mechanically by sliding the connecting lever in a slot until the arm travel is multiplied in exact ratio to the difference in densities.

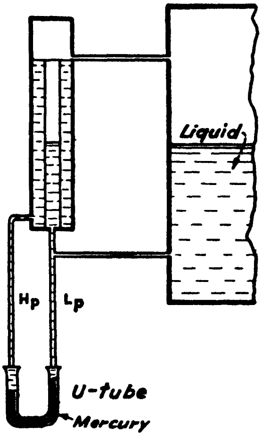


FIG. 3-14.—For accuracy, liquid in the two legs must be kept as near the same temperature as possible.

The U-tube unit in Fig. 3-13 contains a colored proprietary liquid and is piped to a special chamber connected to the water column. As the water level rises or falls, a difference in hydrostatic head pressure occurs in the two legs. This pressure, counterbalanced by the special fluid, shows up as a change in water level.

A modified differential-pressure instrument (Fig. 3-15) serves when the contained vapor will not condense in ordinary chambers or when the liquid in the piping is volatile or sluggish at atmospheric temperature. A small flow of air or gas at *A* from an outside source purges the system to prevent vapor from entering the top connection and liquid from entering the bottom connection. The pressure in the upper line equals the vessel pressure, and the pressure acting on the lower line is the vessel pressure plus the liquid head. The difference is indicated on the instrument in terms of liquid level. Frequently volatile or corrosive liquids or those containing suspended matter clog or choke the meter piping. Here the liquid-purge system that injects cleaning liquid into the piping helps. The liquid flow is restricted to a low rate that does not produce a measurable velocity drop yet is sufficiently rapid to wash out foreign matter as fast as it enters the piping.

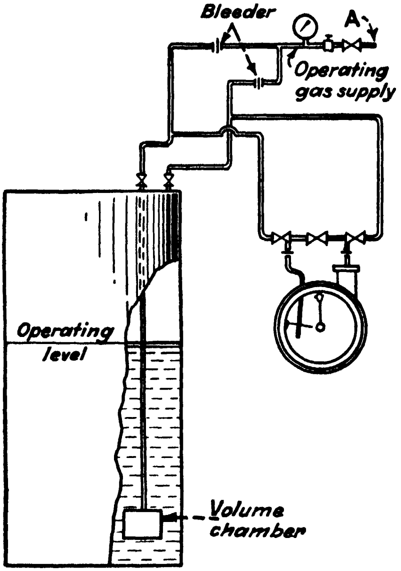


FIG. 3-15.—Differential-head instrument modified for use when liquid vapors do not condense. (*Courtesy of Foxboro Co.*)

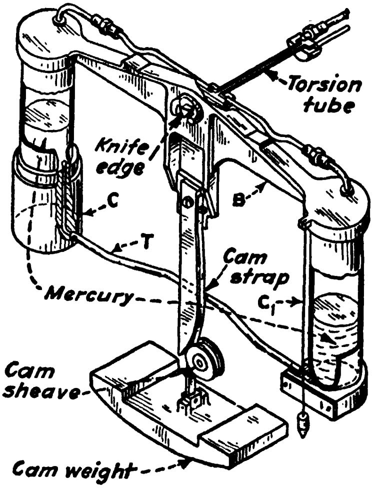


FIG. 3-16.—Unbalance of mercury in the two legs positions the beam. (*Courtesy of Cochrane Corp.*)

Beam-balance-scale or tilting-mercury-cup instruments operate on the unbalanced weight of mercury in the two ends. In Fig. 3-16, the mercury wells are two cyclinders C and C_1, connected at their bottom ends by tube T. At their top ends, the cylinders connect to a beam B supported at its center on a knife-edge. Cylinder C_1 connects to the high-pressure and C to the low-pressure tap on the vessel. Changes in liquid level move mercury from one chamber to another and tilt the beam B, which operates an indicator or controller through the torque tube.

The unit in Fig. 3–17 consists of a large cast-iron U tube partly filled with mercury and delicately balanced on a fulcrum. Connections are made to the vessel through torsion tubes. The position of the sliding weight determines the liquid level that the regulator will maintain. When the liquid level changes, mercury flow unbalances the chamber so that it tilts on a fulcrum and moves a controller through the weigh beam.

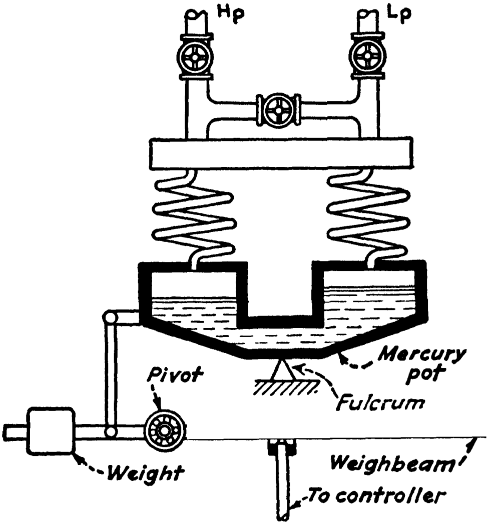


FIG. 3–17.—Tilting mercury cup consists of a pot delicately balanced on a fulcrum. (*Courtesy of Republic Flow Meters Co.*)

The vessel liquid is admitted to the weigh tube (Fig. 3–18) through the lower connection. Air, gas, or vapor in the upper part of the tube is expelled to the vessel through the upper flexible connection as the liquid rises. The increased weight of tube *A* caused by entering liquid is balanced by air pressure built up on reaction diaphragm *B*.

Although constructed of light material, diaphragm elements are suitable for high-static-pressure vessels because the diaphragm is subjected only to the differential pressure. In Fig. 3–19, unbalanced force on the diaphragm rotates the seal assembly on its pivot against the adjustable loading spring to move the regulator valve.

The element (Fig. 3–20) utilizes a neoprene diaphragm to actuate the indicator. A beryllium spring connected to the diaphragm carries a specially designed horseshoe magnet that straddles a thin-walled tubular-well sealing chamber. Inside the tube, a spirally formed strip armature mounted on jeweled bearings carries the pointer. Magnet movement along the well axis causes the armature to rotate.

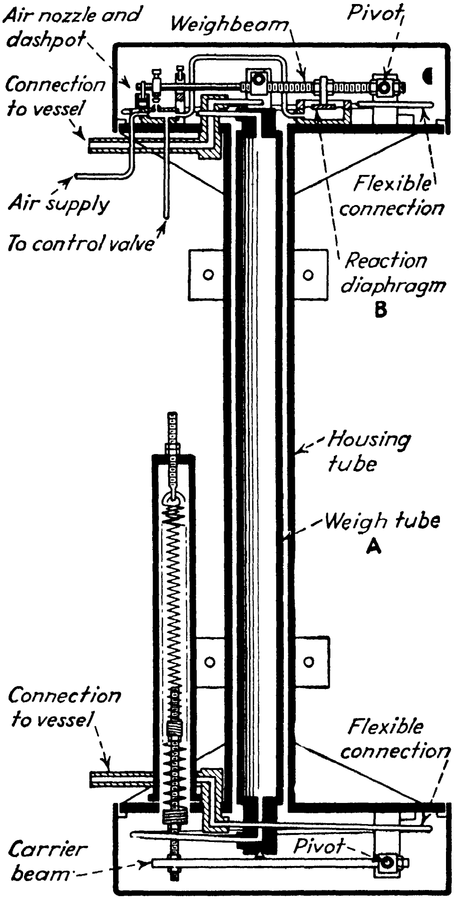


Fig. 3–18.—The weigh tube controls air pressure to its diaphragm *B* according to the mercury level. (*Courtesy of Republic Flow Meters Co.*)

The diaphragm device in Fig. 3–21*A* and *B* operates like a weigh scale. Differential pressure applied to diaphragm *A* produces a force that is transmitted through the push rod and balance beam *B* to reaction diaphragm *C*. A tip mounted on the beam end throttles compressed-air leakage from nozzle *D* to regulate the air pressure acting beneath the reaction diaphragm and applied to the control mechanism. Thus the differential-pressure force multiplied by its lever length is balanced by the reaction-diaphragm force multiplied by its lever length to bring the system to equilibrium.

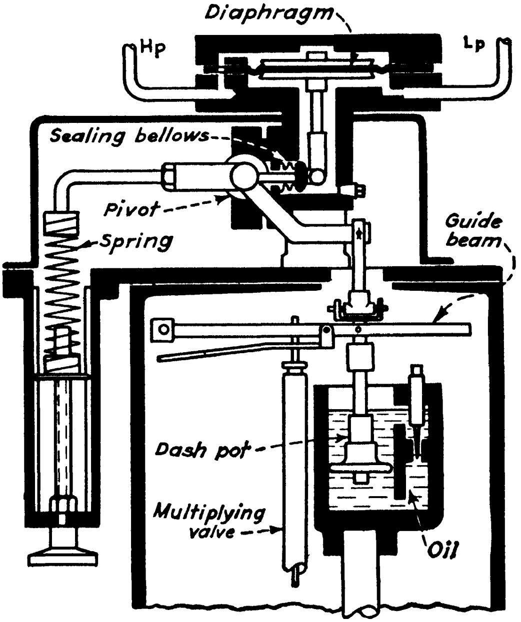


Fig. 3–19.—Unbalanced force, caused by differential-head pressure on the diaphragm, moves control valve. (*Courtesy of Republic Flow Meters Co.*)

A bellows manometer positions the pointer or controller according to the pressure differential acting between the two vessel connections, one tap connecting inside the bellows and the other outside. The device in Fig. 3–22 has a controller pilot valve connected directly to the indicator pointer.

In Fig. 3–23, the high-pressure tap connects to the outside of the bellows, and the low-pressure tap goes to the inside. A yoke attached to the bellows connects to a lever through flexible strips. The other ends of the levers clamp to the free end of a three-element torque tube that seals the manometer chamber. The movement of the bellows is transmitted through the torque-tube center shaft to a pointer on the other end (Fig. 3–24). When differential pressure increases, the free end of bellows *A* moves to the left, rotates lever *B*, and rotates torque tube *C* (Fig. 3–23). The actuating mechanism illustrated in Fig. 3–25 consists of a pressure

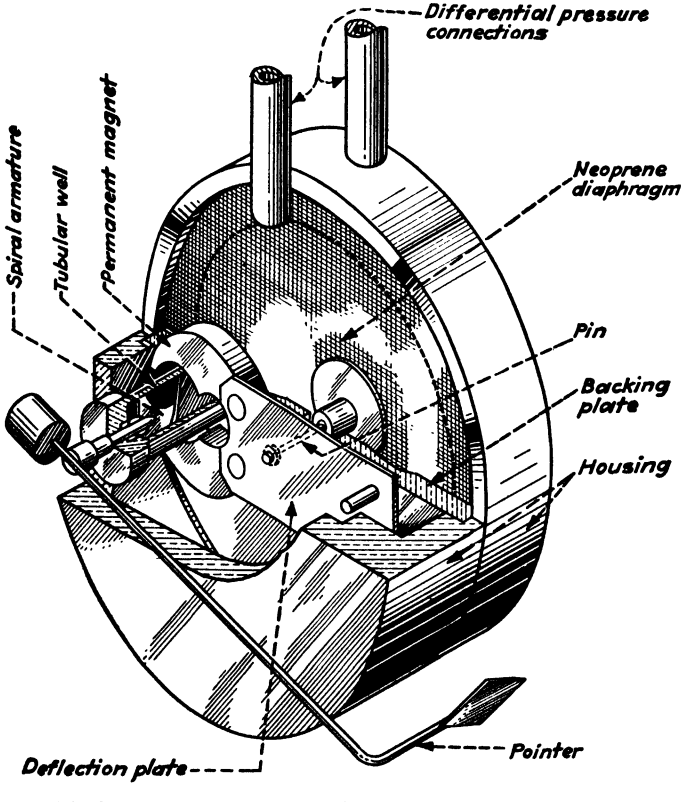


FIG. 3–20.—Diaphragm moves a magnet along a tubular well to rotate a spiral-strip armature inside. (*Courtesy of Yarnall-Waring Co.*)

chamber sealed at one end by a bellows. Static pressure admitted to the chamber acts on the outside of the bellows and is opposed by a coiled spring inside the bellows. Thus this device measures static pressure instead of pressure differential and can be used only on open tanks. The motion of the bellows is transmitted through the push rod to a wiper arm that moves across a commutator. The commutator consists of alternate disks of copper and insulating material with each copper disk connecting to a fixed electrical resistance. If the pressure or liquid level changes, the wiper arm moves across the commutator to vary the electrical resistance in the indicator circuit.

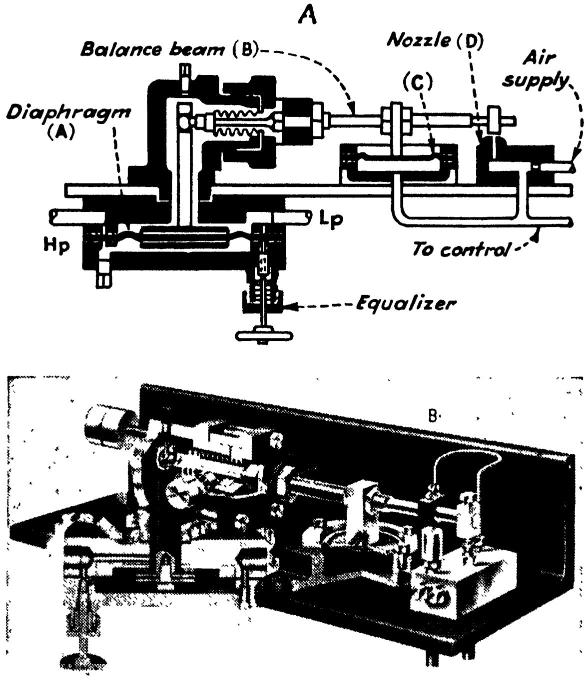


FIG. 3-21.—Main diaphragm *A* moves to throttle output from air nozzle *D*.

Differential-head manometers can also be used to measure and control flow if a restriction of some sort is placed in the pipe line to create a differential pressure that can be applied across the manometer (Fig. 3–26). For closed pipes, the restricting (primary) element may be a thin-plate orifice, flow nozzle, pitot tube, venturi tube, or a modification of one of them.

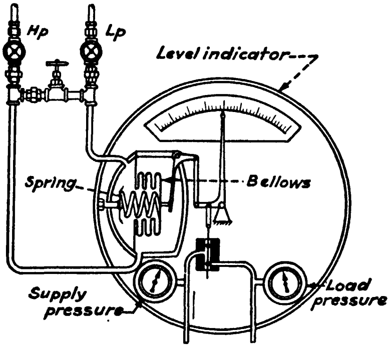


FIG. 3–22.—Differential-head instrument with bellows-operated control valve and pointer. (*Courtesy of Bailey Meter Co.*)

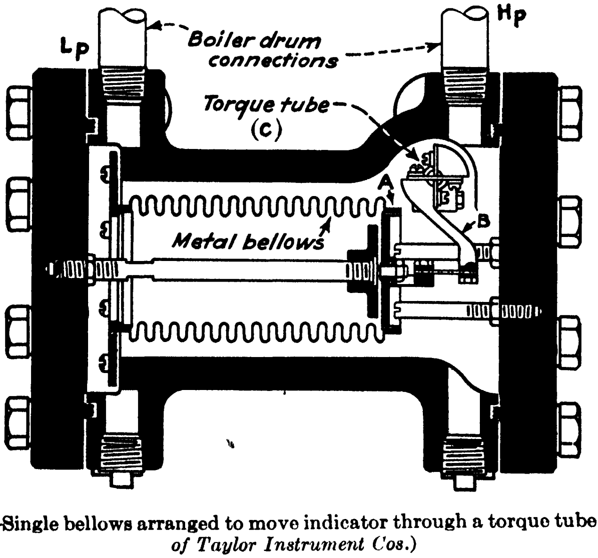


FIG. 3–23.—Single bellows arranged to move indicator through a torque tube. (*Courtesy of Taylor Instrument Cos.*)

If a disk orifice is used and pressure measurements are made when fluid is flowing through the pipe, the pressure will vary like that shown by the water columns in Fig. 3–26. This occurs be-

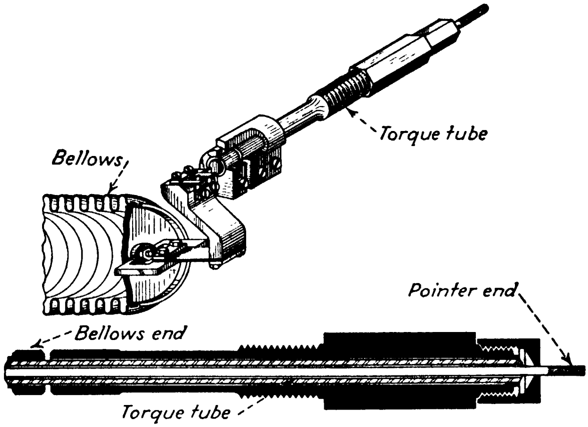


FIG. 3-24.—Torque tube uses a shaft and two sleeves to provide a pressuretight seal. (*Courtesy of Taylor Instrument Cos.*)

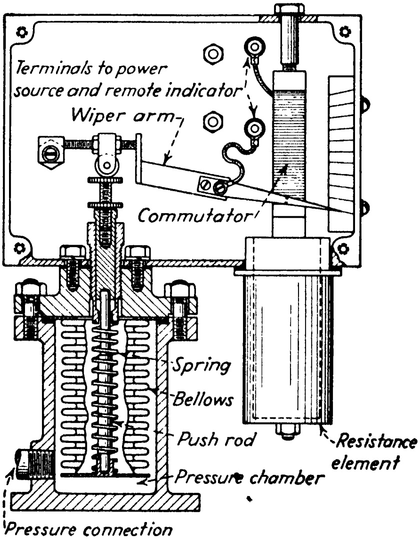


FIG. 3-25.—Single bellows unit slides an arm over a commutator to vary the resistance of the metering circuit.

cause the velocity of the fluid changes as it passes through the orifice. When fluid flows through the orifice, it accelerates, and part of the static pressure turns into velocity head that produces a pressure drop. A short distance from the orifice, however, the fluid returns to its original velocity, and a large part of the static-pressure drop is regained within five or six pipe diameters downstream from the orifice.

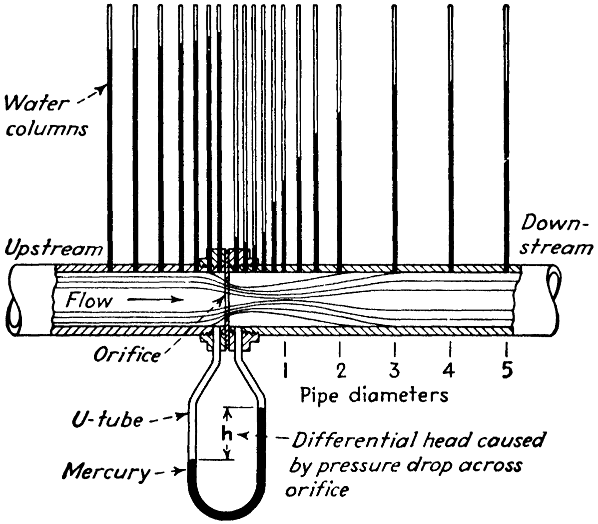


FIG. 3–26.—Water columns show how the pressure varies ahead of and beyond an orifice.

On the upstream side, because of the abrupt change in flow area (caused by the orifice), a slight pressure rise occurs at the orifice plate. On the downstream side, minimum pressure occurs not at the orifice but at some distance away. It is at this point that the jet of fluid attains its greatest velocity and the static pressure its lowest value. The point where minimum pressure occurs is called the *vena contracta*. How far this occurs downstream from the orifice depends on fluid viscosity, orifice size, static pressure ahead of the orifice, pipe size, and the specific gravity of the fluid. Examine Fig. 3–27 to see why the three following tap locations came into use.

Vena contracta taps, commonest on steam and water lines, are made directly into the pipe wall and do not require special flanges to hold the primary element. They measure the differential pressure at nearly its maximum value, because, as the name suggests, the downstream tap is located at the minimum diameter of the jet and hence at the minimum downstream static pressure. The center line of the upstream tap is one pipe diameter from the upstream face of the orifice. The downstream-tap center-line position (in pipe diameters) measured from the same *upstream* face varies with the d/D orifice-to-pipe-diameter ratio shown in Table 3–1.

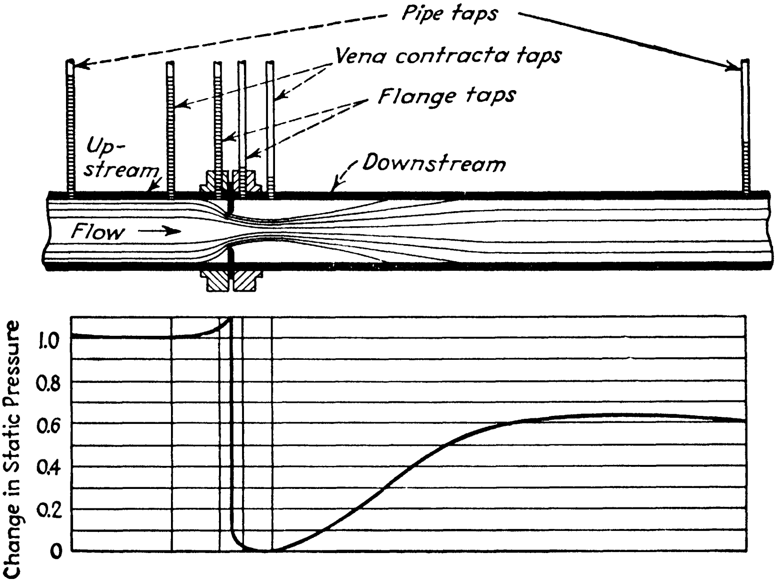


FIG. 3–27.—Change in static pressure downstream from the orifice and how it affects pressure-tap location.

Flange taps are incorporated in special orifice flanges and have their center line a distance of 1 in., respectively, from the upstream and downstream faces of the orifice plate. Like vena contracta taps, they measure the differential pressure at near its maximum value, because, even though the downstream tap is not at the point of minimum downstream static pressure, the upstream tap is closer to the inlet side of the orifice where a slightly elevated static pressure exists.

With pipe-line taps, the upstream tap is placed 2½ pipe diameters from the upstream face of the orifice plate, and the downstream tap is placed seven pipe diameters from the *upstream* face. These distances are extensively used in gas measurement. The downstream tap is at or near the point of maximum downstream static pressure (minimum differential), and hence these connections measure the same volume of flow as the other taps but with a considerably smaller differential-range meter. Pipe taps require 10 pipe diameters of additional straight pipe in comparison with the necessary lengths for flange or vena contracta taps. On gas measurement, where long straight lengths of pipe are usually available, these taps find greatest use; but, for the typical power- or process-plant application to steam or water lines, this extra straight-length requirement is enough to disqualify them in most cases. Moreover, greater accuracy can be obtained by vena contracta or flange taps.

TABLE 3–1. DOWNSTREAM TAP POSITION

Ratio d/D	Distance in pipe diameters		
	Min	Mean	Max
0.20	0.37	0.86	1.30
0.30	0.44	0.80	1.15
0.40	0.47	0.73	1.00
0.50	0.47	0.66	0.84
0.60	0.42	0.57	0.70
0.70	0.34	0.45	0.55

If one of the differential-head manometers or a mercury U tube is connected across the orifice in Fig. 3–26, mercury in the upstream leg will fall, and that in the downstream leg will rise. The difference in levels h represents the difference in static pressure between the two points where the U tube connects. This pressure differential varies as the square of the fluid velocity and can be used as an accurate index of the flow rate.

For example, if at 25 per cent flow the differential pressure is 2 in. of mercury, it will be $(50^2 \times 2) \div 25^2 = (50 \times 50 \times 2) \div (25 \times 25) = 8$ in. of mercury at 50 per cent flow. In like manner, at 75 per cent flow the differential pressure is 18 in., and at 100 per cent it is 32 in. Any meter that indicates or records directly from differential pressure will have an unevenly divided scale because of this square-of-the-velocity factor. Divisions get wider up scale.

To get a uniformly divided scale, some means must be provided in the manometer for taking the square root of the differential pressure. Taking the square root of the pressures previously computed gives $\sqrt{2} = 1.4$, $\sqrt{8} = 2.8$, $\sqrt{18} = 4.2$, and $\sqrt{32} = 5.7$. Here the square root of the differential head at 50 per cent flow is twice that at 25 per cent flow, at 75 per cent it is three times, and at 100 per cent flow it is four times as much.

We cannot extract the square root of the differential pressure in a manometer similar to those described for measuring liquid level, because the mercury wells have uniform cross sections (Figs. 3–8 and 3–9). In these units, the difference in mercury level in the two wells at 50 per cent flow will be four times that at 25 per cent flow, nine times at 75 per cent flow, and sixteen times at 100 per cent.

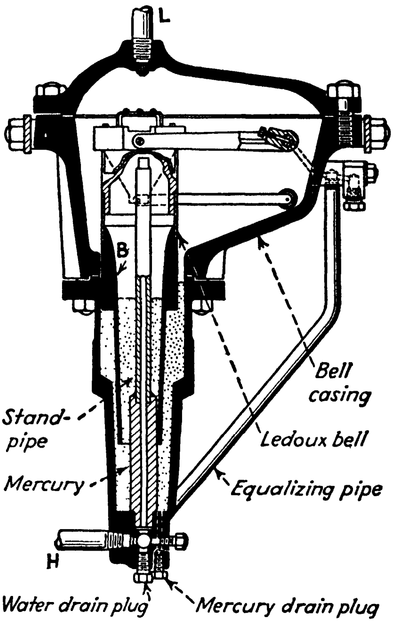


FIG. 3–28.—Bell-shaped float, because of its flaring wall, gives an evenly divided chart scale. (*Courtesy Bailey Meter Co.*)

A manometer that automatically extracts the square root of the differential pressure to give a uniformly divided scale is shown in Fig. 3–28. It consists of a bell *B*, known as a *Ledoux bell*, sealed in mercury. A high-pressure tap from the pipe line connects at *H* and acts inside the bell. A low-pressure tap connects at *L* and acts on the outside of bell. At no flow, the bell is at its lower position floating in the mercury, and the indicator is at zero.

As flow increases, differential pressure acting across the bell increases, and it rises from the mercury until the reduction in buoyancy just balances the differential pressure. The sloping contour of the bell wall makes the instrument pen move in direct proportion to change in flow: therefore scale graduations are uniform. The manometer in Fig. 3–29 operates in a similar manner. Here the trumpet-shaped range tube *B* takes the place of the Ledoux bell. For every inch the mercury level in *B* falls, a greater quantity flows over into float chamber *F*. The cam in Fig. 3–16 serves the same purpose.

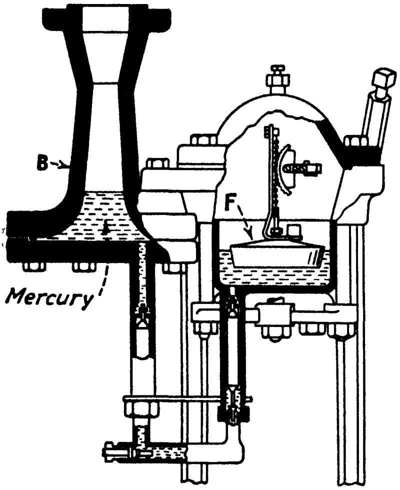


FIG. 3–29.—In this manometer, the trumpet-shaped mercury chamber gives an evenly divided chart scale.

A high degree of accuracy is required in making the orifice, because it must be calibrated to give the required pressure differential for specific conditions of flow. Thin-plate orifices are of three general designs: *concentric*, with the orifice in the center of the plate (Fig. 3–30*A*); *eccentric*, with it off center (Fig. 3–30*B*); and *segmental*, Fig. 3–30*C* and *D*).

Concentric-orifice plates are used in most installations and for steam or gas service are provided with a small hole at the bottom, flush with the pipe to permit the condensate to drain through. On horizontal liquid lines, this hole is at the top to permit air or gas to pass. For fluids carrying suspended matter that may build up back of the plate, the eccentric or segmental design is recommended. With the opening installed flush with the bottom of the pipe, the foreign material does not pile up. Orifice plate *D* is riveted at a convenient point in large pipe lines, obviating the necessity of installing a special flange.

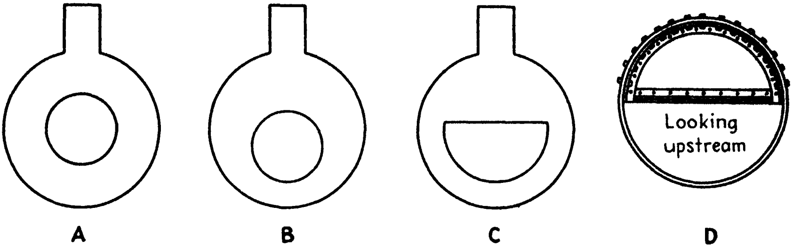


FIG. 3-30.—Position and shape of the hole in an orifice plate vary to suit the requirements.

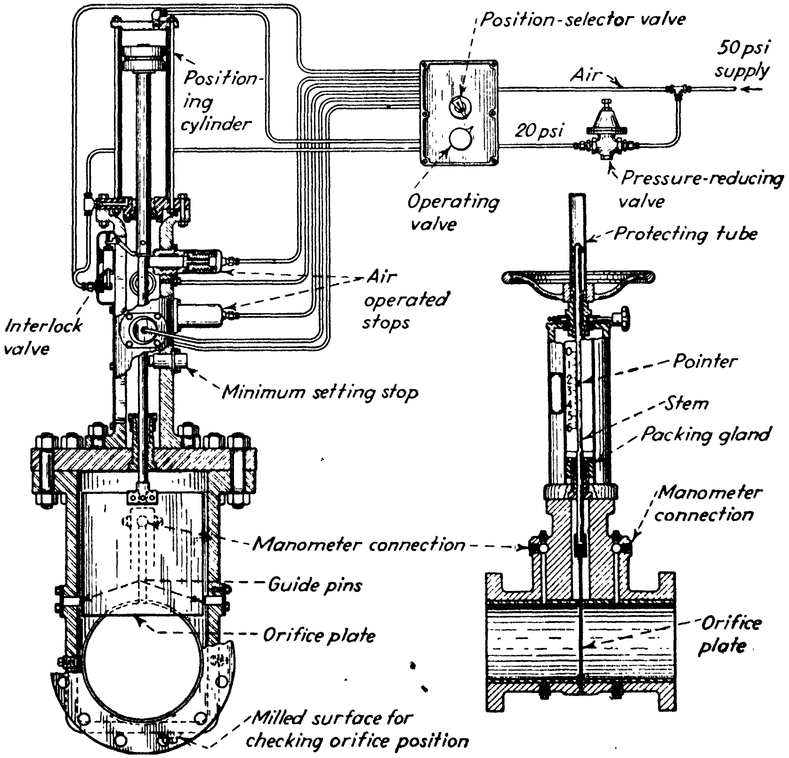


FIG. 3-31.—An adjustable orifice can be positioned by a power device or handwheel. (*Courtesy of Bailey Meter Co.*)

Where flowmeters must operate over wide ranges, adustable orifices (Fig. 3–31) are available. They consist of a square-bottom plate slightly wider than the pipe. The lower edge of the plate forms a segmental orifice similar to Fig. 3–30*C*, the size of the segment depending on how far the plate is lowered. It can be operated manually like a gate valve or be fitted with a power device for remote control. Figure 3–32 illustrates how the shape of an orifice affects the location of the low-pressure tap.

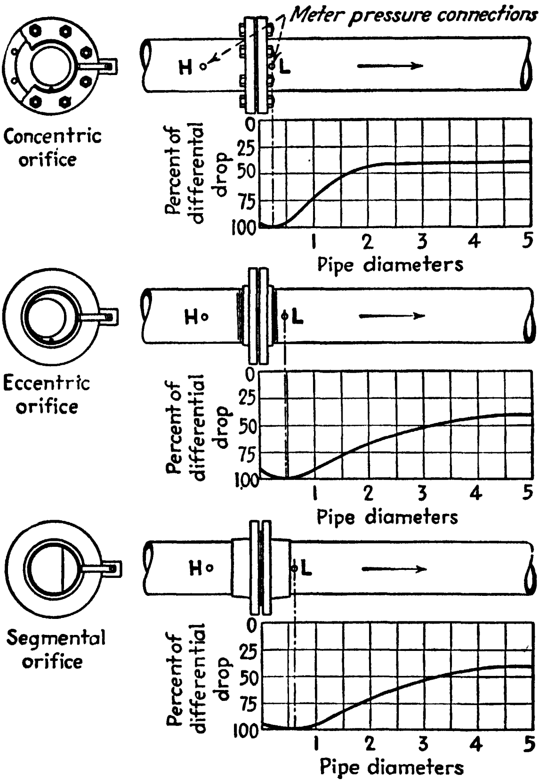


Fig. 3–32.—Curves aligned with main pipe show how orifice shape affects the location of low-pressure tap. (*Courtesy of Bailey Meter Co.*)

Orifice openings may have a sharp edge (Fig. 3–33*A*) or a square edge (Fig. 3–33*B*) if the plate is not more than 1/32 in. thick. For thicker plates, the downstream edge is beveled to reduce the thickness of the orifice rim to 1/32 in. In Fig. 3–33*C*, the low-pressure tap is in a ring that serves as the orifice-plate holder and is clamped between pipe flanges. The orifice plate in Fig. 3–33*B* is carried on a steel ring that has radial differential-pressure taps that connect to annular grooves extending around the full circumference. These grooves are intended to average the pressure over the orifice face and minimize the effects of uneven flow distribution across the pipe section caused by bends and valves.

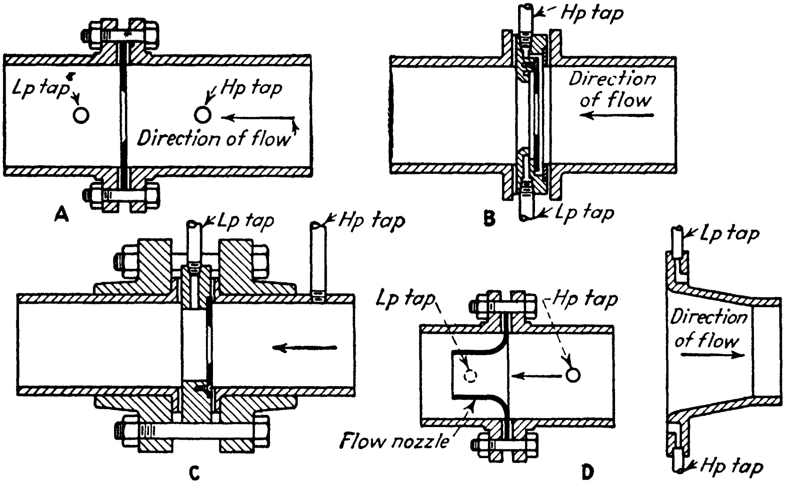


FIG. 3–33.—Typical orifice and flow-nozzle installation features.

Another form of differential-pressure device is the flow nozzle (Fig. 3–33*D*), a short cylinder with a flared entrance. The flow nozzle is recommended where a rugged device is required and where the expense of a venturi tube is not justified. It also finds wide use where orifices cannot satisfactorily handle the maximum capacity desired. A flow nozzle has about 60 per cent greater capacity for a given throat diameter than has a thin-plate orifice.

Venturi *tubes* (Fig. 3–34*A*), more costly than thin-plate orifices or flow nozzles, are used extensively on water systems and other large water-flow lines and where it is important to keep the net pressure loss to a minimum. With a reasonably steady flow, venturi tubes are accurate at any pressure, up to moderate temperatures, and at capacities limited only by the size of the pipe in which they are installed. Pressure taps are made through annular chambers, known as *piezometer rings,* which communicate with the interior of the tube through small vent holes. Opposite each vent hole is a tapped hole in the outer casing. All the holes are plugged except the two used for the manometer connection. The remainder serve as cleanouts.

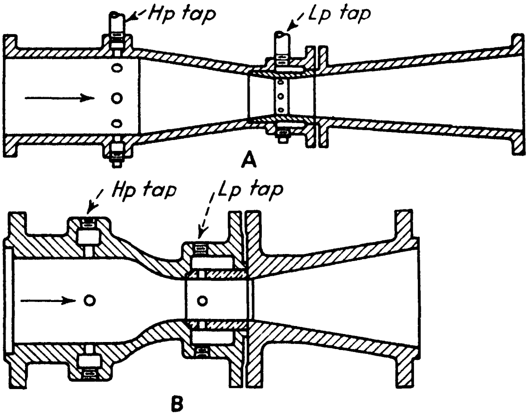


FIG. 3–34.—Venturi nozzle *B* is somewhat shorter than a venturi tube *A*.

A venturi *nozzle* (Fig. 3–34*B*) is used where space is too restricted for a standard venturi tube. The head loss is about double that of a tube, but the initial cost is less. The capacity of both devices is about the same.

The following section is adapted from an article by R. E. Sprenkle, Bailey Meter Co., *Power,* July, 1945.

Piping-arrangement studies of fluid flow, made with air, gas, and water in the laboratories of the American Gas Association, University of Pennsylvania, Case School of Applied Science, Ohio State University, and many industrial concerns, have produced data to formulate *Piping Requirement Standards,* or basic rules upon which to make piping recommendations for installing flowmeter orifices (Fig. 3–35). They are divided into eight schedules; seven designate the length of straight pipe needed for orifices or flow nozzles, and the eighth covers venturi tubes. In addition, these standards specify that no thermometer well, bulb or socket, resistance element or other similar unit should be located closer than 6 pipe diameters to the primary element on the outlet side or closer than 15 diameters on the inlet side.

The required length of straight pipe preceding and following the orifice or nozzle increases with both diameter ratio and complexity of the piping arrangement. The diameter ratio is *d/D,* in which *d* is the diameter of the primary element throat and *D* is the internal pipe diameter.

To make the standard schedules easier to understand, simple layouts convert them into terms of typical piping arrangements commonly met in industrial applications and apply the particular schedule to each. Consider bends or fittings in the same plane: diagram 1 (Fig. 3–36) shows that the orifice or flow nozzle can immediately follow a long-radius bend if the following conditions are met: (1) there must be at least 10 diameters of straight pipe preceding the long-radius bend, (2) the diameter ratio of the orifice or flow nozzle must not exceed 75 per cent, and (3) the radius of the bend must be five pipe diameters or more.

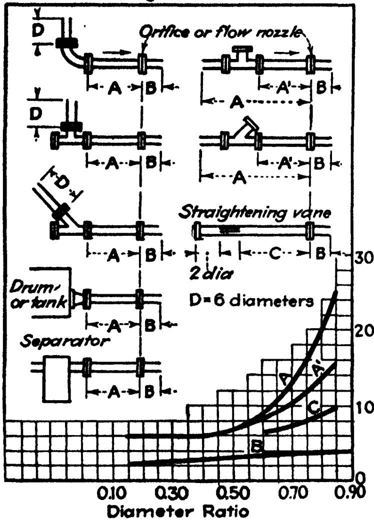
Diagram 2 differs from 1 in that there must be a length of straight pipe between the bend and orifice, or nozzle, if the diameter ratio of the primary element exceeds 75 per cent. If a 10-diameter length of straight pipe precedes the bend, it can again be considered as the equivalent of so much straight pipe, as in diagram 1. The length of straight pipe *following* the bend plus the developed length of the bend can therefore be assumed to meet the requirements of schedule 1 (Fig. 3–35). (Refer to Fig. 3–35 for all schedule references.)

Occasionally two orifices or flow nozzles are installed in series to operate check meters (diagram 3, Fig. 3–36). A particular application is made when a producer sells quantities of steam, gas, or liquids to a consumer, each party to the contract having an independent meter to check billing charges. Here there must be at least 10 diameters of straight pipe between the two primary elements to allow complete pressure restoration from the first orifice or flow nozzle before the second is reached.

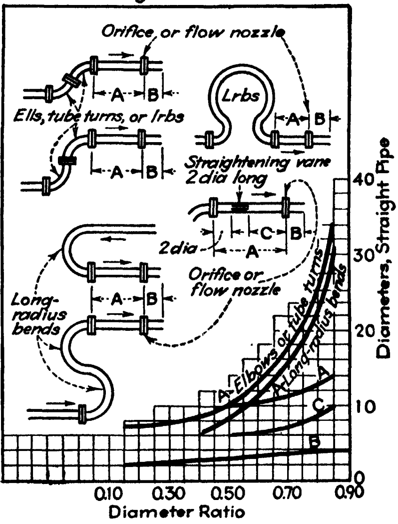
Diagram 4 illustrates a point not readily apparent from schedule 5 but important nevertheless. If other fittings or bends precede an expander or reducer, the length of straight pipe between such fittings and the primary element must meet the requirements of the schedule that represents the bend combination. In diagram 4, for example, when two bends in the same plane precede the reducer, the total length of straight pipe between the last *bend* and the orifice or flow nozzle must agree with schedule 2. However, only the lengths specified by schedule 5 are required between the *reducer,* or expander, and the orifice, or nozzle. Obviously, if long lengths of straight pipe precede the reducer or expander, only the requirements of schedule 5 apply.

Flow entering one end or side of a tee (diagram 5) or a Y fitting (diagram 11) and splitting into two separate exit flows creates a disturbance in that fitting, which requires that somewhat longer lengths of straight pipe be used to smooth out the flow (schedule 2). Use the elbow curve (schedule 2), be-

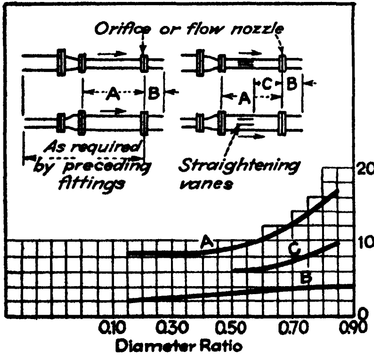
Schedule 1
For Orifices and Flow Nozzles,
All Fittings in Same Plane



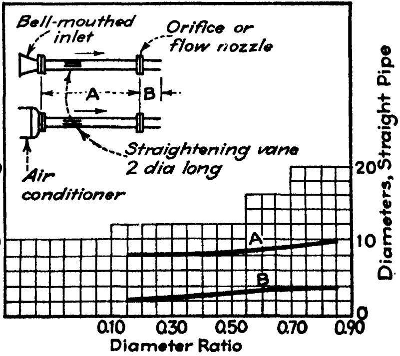
Schedule 2
For Orifices and Flow Nozzles,
All Fittings in Same Plane



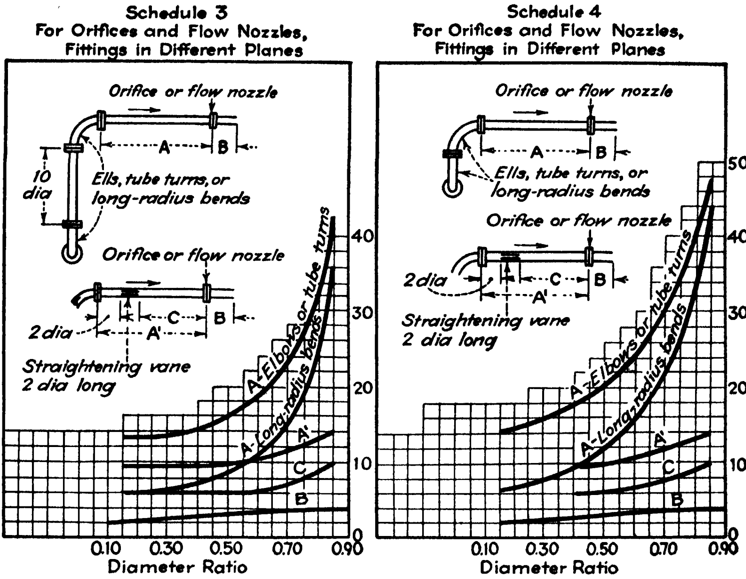
Schedule 5
For Orifices and Flow Nozzles
With Reducers and Expanders



Schedule 6
For Orifices and Flow Nozzles,
In Atmospheric Intake

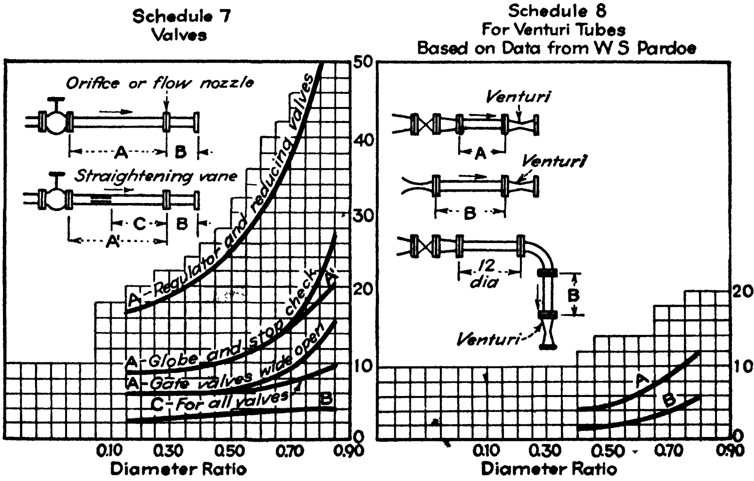


	0.0–0.50 ratio	0.50–0.60 ratio	0.60–0.70 ratio	0.70–0.80 ratio
Fittings allowed on outlet side in place of straight pipe	1. Tees 2. 45 Ells 3. Gate valves 4. Separators 5. Y-fittings 6. Expansion jts.	1. Tees 2. Expansion jts 3. Gate valves 4. Y-fittings 5. Separator (if inlet neck is one dia long)	1. Gate valves 2. Y-fittings 3. Separator (if inlet neck is one dia long)	1. Gate valve 2. Long-radius bend



Schedule 3
For Orifices and Flow Nozzles,
Fittings in Different Planes

Schedule 4
For Orifices and Flow Nozzles,
Fittings in Different Planes



Schedule 7
Valves

Schedule 8
For Venturi Tubes
Based on Data from W S Pardoe

FIG. 3-35.—These specifications show how to lay out pipe lines and locate orifices and flow nozzles where least fluid disturbance occurs.

cause the turn through a tee is similar to that through an elbow. When the inlet flow enters from both ends of the tee (diagram 6) or from one end and the side (diagram 8) with discharge emerging from the other end, use schedule 4 requirements for elbows The reason is that two or more entrance flows mixing in a single fitting produce a turbulence equal in magnitude to that created by two turns at right angles to each other.

Diagrams 7 and 9 show other multiple-entrance flows. Remember that, when several entrance lines successively discharge into a common header (diagrams 8 and 9), the magnitude of disturbance produced may increase with the number of such inlet flows; and it is therefore sometimes necessary to use a straightening vane in addition to the lengths of straight pipe as required by schedule 4 for elbows.

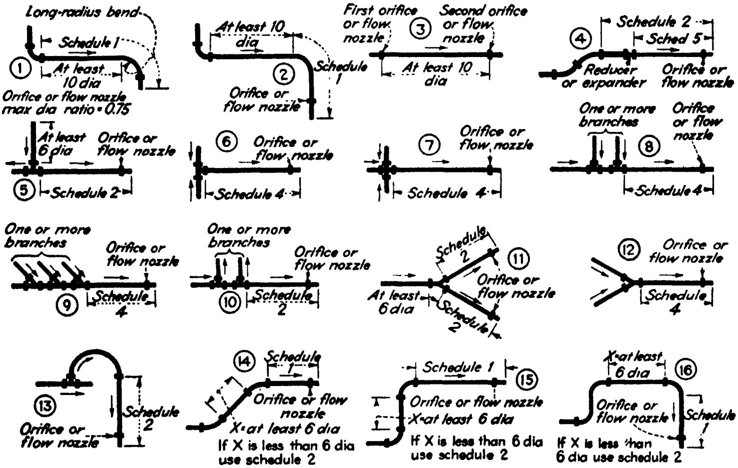


FIG. 3–36.—Multiple-entrance flows produce disturbances of such magnitude that they are particularly difficult to eliminate.

Diagram 12 shows a typical arrangement for joining two lines into one, *e.g.*, two steam leads from a double-outlet boiler that merge into one pipe before connecting to the header. To eliminate the disturbance created by a Y fitting, this arrangement requires the maximum lengths for elbows called for in schedule 4.

Diagrams 13 to 16 give simple applications of the requirements of schedules 1 and 2 to piping systems having two or more bends all in the same plane. If six diameters or more separate these bends, schedule 1 applies; but, if there are less than six diameters of straight pipe, schedule 2 applies.

Isometric diagrams 17 to 22 in Fig. 3–37 show a series of typical take-off connections from a header to a turbine, blower, pump, or compressor, or to a distribution line. In every instance, the header tee must be considered as having the same effect as an elbow and thus as a 90-deg bend in the flow path.

The simplest arrangement of this group is shown in diagram 17, where a long-radius bend closely follows the tee at the header discharge. Since the plane of the bend is at right angles to the plane of flow through the tee, there are, in effect, two bends at right angles to each other, which require schedule 4 piping lengths for long-radius bends. If an elbow or tube turn is used instead of a long-radius bend, follow schedule 4 requirements for elbows. If a piece of straight pipe shorter than 10 diameters follows the header tee (diagram 18), apply schedule 3 requirements for long-radius bends instead.

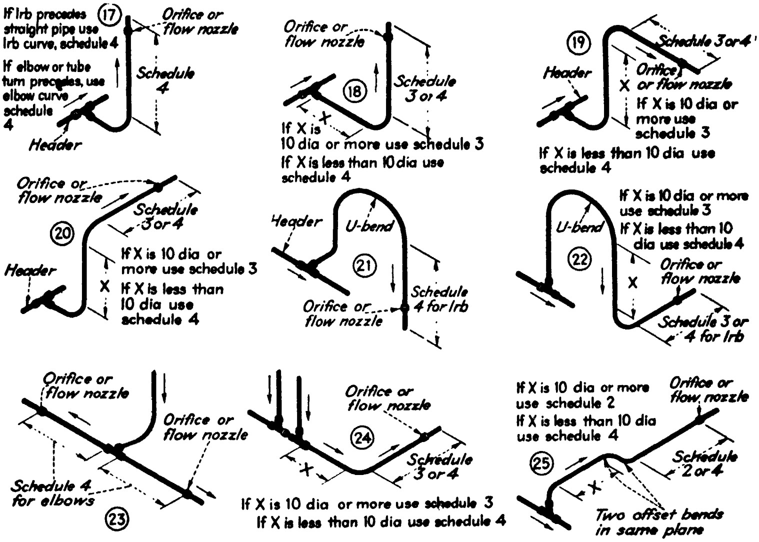


FIG. 3-37.—Where component bends are not in the same plane, the schedule used in Fig. 3-35 depends on the amount of straight pipe between preceding bends.

In diagram 19, schedule 3 or 4 requirements apply in spite of the fact that the two bends preceding the orifice or flow nozzle are in the same plane. The reason is that the flow turns through the header tee, preceding these bends, at *right angles* to the plane of the two bends, and the stricter requirements therefore apply The same procedure applies to diagram 21, in which another bend ahead of the U bend is at right angles to it and thus dictates the requirements of schedule 4 for long-radius bends.

Diagrams 23 and 24 show flow entering a header from one or more inlets. In both, consider the header tee as a bend, so that, together with the bend either preceding it, as in diagram 23, or following it, as in diagram 24, a two-bends-at-right-angles combination results, and schedule 4 must be applied.

The schedule 4 curve for elbows applies to diagram 23, whereas the schedule 4 curve for bends applies to diagram 24.

Consider offsets in the same category as full bends if the amount of offset is two or three pipe diameters. Diagram 25 illustrates an application where two bends at right angles precede the offset.

Diagrams in Fig. 3–38 show typical outlets coming directly from a boiler drum and superheater header. Several important points are worthy of mention in connection with these piping arrangements.

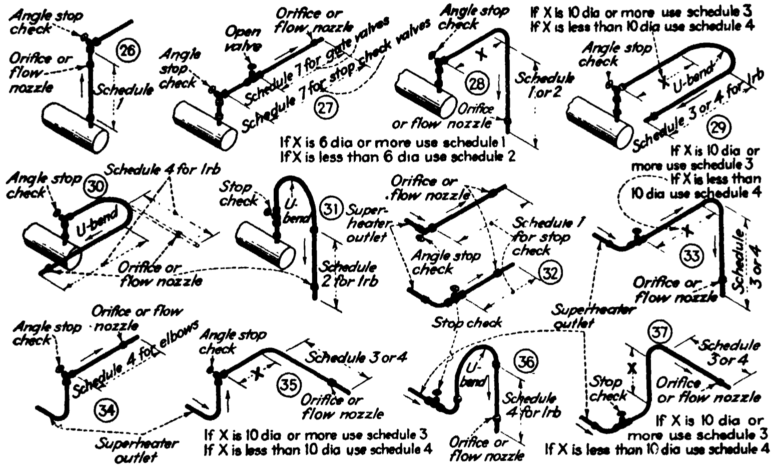


FIG. 3–38.—When bends or turns closely follow an angle stop-check valve, the latter is considered to have the same disturbing effect on fluid flow as an elbow.

The first is that an angle stop-check valve must always be considered as an elbow and included as an *integral part of the piping arrangement that immediately follows it.* You cannot start at the valve outlet flange and consider only the piping arrangement beyond. This is clearly illustrated in diagrams 28, 29, and 30 in Fig. 3–38. In diagram 28, the angle valve forms the first bend of a two-bends-in-the-same-plane combination, thus calling for the requirements of schedule 1 or 2; whereas, in diagrams 29 and 30, the angle valve is the first bend in a two-bends-at-right-angles combination, requiring the use of schedule 3 or 4 for long-radius bends.

If a bend instead of a straight pipe precedes the angle stop-check valve, as at the superheater outlet (diagram 34), this bend must also be considered an integral part of the piping arrangement. To illustrate: Compare the piping arrangement of diagram 34 with 27, and of 33 with 28. Diagram 34 is similar to 27 except that a bend precedes the angle valve, at right angles to it. This dictates that schedule 4 requirements for elbows should be followed instead of schedule 7. Similarly, the bend preceding the angle valve in diagram 33 is at right angles to the bend following the valve, thus requiring schedule 4 for long-radius bends; whereas diagram 28, with the same piping arrangement following the angle valve itself, requires only schedule 2.

Diagram 27 illustrates an important point in schedule 7, namely, when two valves are close to each other in series, the one having the stricter requirements governs, regardless of its position in the assembly. In other words, you cannot take the requirements of an open gate valve alone, as in diagram 27, just because the gate valve is closest to the orifice or nozzle, unless the distance between the two valves is equal to or greater than the difference in the requirements of these two valves, as given in schedule 7.

For example, assume a 75 per cent diameter-ratio orifice is to be installed in the piping arrangement shown in diagram 27. This orifice would have to be 18 pipe diameters beyond the stop-check valve for satisfactory results despite the fact that the gate valve that follows the angle valve requires only 11 diameters of straight pipe between it and the orifice or nozzle. This means that the straight pipe between the angle valve and the open gate valve in this installation must be 7 diameters in length.

In applying these schedules (Fig. 3–35) give preference to longer lengths of straight pipe *A* rather than to the alternate use of straightening vanes with much shorter pipe lengths, *C*. Straightening vanes do not correct all forms of disturbances and, in fact, may tend to perpetrate rather than eliminate irregular flows.

Although certain fittings are permitted on the outlet side of the primary element (tabulation in Fig. 3–35), the preferred arrangement is to use lengths of straight piping *B* called for in each schedule. If a control valve follows the orifice or nozzle, install at least five or six diameters of straight pipe between the orifice or nozzle and valve regardless of the diameter ratio.

These diagrams do not represent the only satisfactory piping arrangements for accurate metering. Many others are equally good or better. However, they serve to illustrate the principles involved in putting the Piping Requirement Standards into practice so that piping systems can be laid out to give greatest accuracy to the metering element.

When an orifice plate is to be installed in a pipe flange, cut the gaskets to match the outside diameter of the orifice plate and the inside diameter of the flanges. Sticking the gaskets to the plate with shellac makes it easier to center the complete unit between the flange bolts.

Locate the center of the pressure-tap holes, and drill them perpendicular to the pipe axis. Thread the holes, and remove all burrs from the inside of the pipe. When preparing the pressure-tap pipes, thread them so as to make a tight connection when their ends are screwed in flush with the inner wall of the main pipe. If the ends protrude into the pipe, they must be filed down flush. The importance of making the pressure taps perpendicular to the main line is illustrated in Fig. 3–39. Even though taps are flush with the inner wall, the pressure in tubes *a* and *c* is affected by the velocity of the moving fluid. Tube *a* shows a lower and tube *c* a higher level than that in tube *b*. Discrepancies would be greater if the tube ends were not flush with the pipe wall.

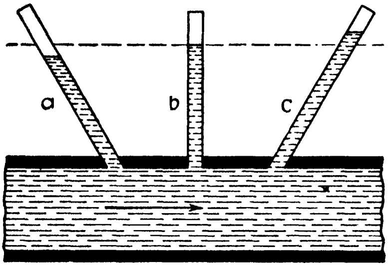


FIG. 3–39.—For accurate readings, pressure taps must be perpendicular to pipe axis.

On steam lines, the pressure taps are made into the side of the pipe on a horizontal plane passing through the pipe axis. The taps are provided with radiator reservoirs (Fig. 3–40) so that the steam will condense and keep the manometer piping filled with water. The reservoir provides storage space to take care of evaporation in case the pressure drops below the flash point during rapidly changing flow. An offset in the entrance pipe ensures that the reservoir will be full of water at all times.

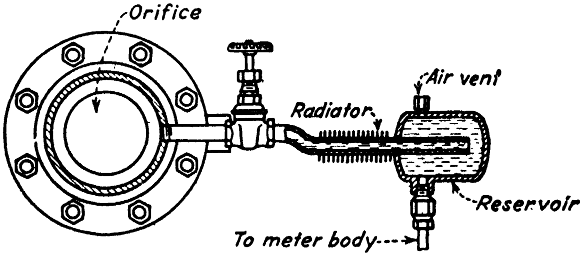


FIG. 3–40.—Radiating fins on the pipe or reservoirs condense the steam.

Meters for measuring steam flow in horizontal lines are connected to the pipe line as in Fig. 3–41*B*. When possible, the meter element should be installed below the orifice level so that there will always be a positive head of water on both legs of the manometer.

When the meter must be placed above the orifice, make piping connections as shown by the dotted lines, provided that the steam pressure is over 10 psi. Here the piping drops below the reservoirs a distance equal to the maximum differential head that will be on the meter. It then goes vertically to the meter where air vents are

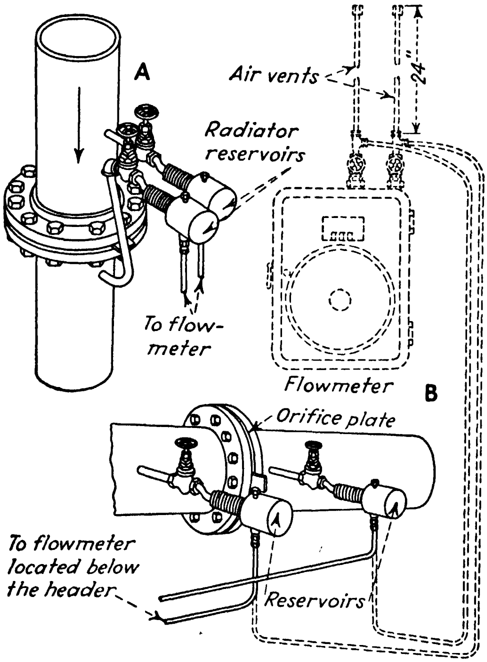


FIG. 3–41.—On a vertical pipe, mount the reservoirs at the same level *A*. Connections for mounting a meter above or below a horizontal line *B*.

provided. On vertical lines (Fig. 3–41*A*), the reservoir on the lower tap should be raised so that its center line is on a level with that of the upper reservoir.

When the fluid being measured contains sediment, install settling chambers (Fig. 3–42). The cutaway section shows how inlet connections extend nearly to the bottom of the chamber with the meter lines coming off at the top. Sediment can be blown out of the chambers periodically through the drain lines. Leaking blowoff valves cause the meter to register incorrectly, and they must therefore be kept tight.

Whenever a corrosive fluid or a viscous or highly volatile liquid is to be measured, install separating chambers (Fig. 3–43) to prevent the fluid from entering the meter mechanism and connecting lines. The separating chambers permit filling the meter lines with water. For oil and other liquids that do not mix with water, the chamber is divided into two parts by the vertical partition. If the liquid is lighter than water, pressure taps from the pipe line go to the top of the chamber, and meter connections come off the bottom. When the measured liquid is heaviest, the connections are reversed. Gage glasses on the side permit adjusting the water levels to the halfway mark.

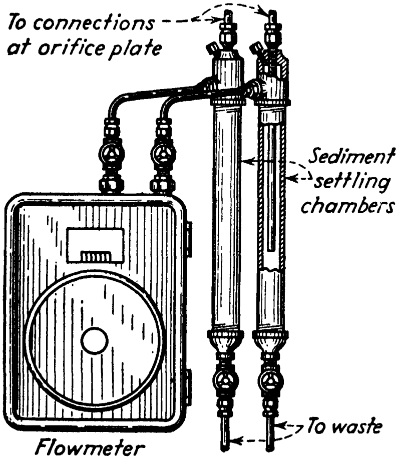


FIG. 3–42.—Sediment chambers prevent suspended matter reaching the flowmeter.

For liquids such as brine that mix with water, a separate chamber is needed for each tap. In these, the partition does not go to the top. The upper part of each chamber is filled with a third liquid that, being lighter than water and the measured liquid, will not mix with either. The meter connection and the tap line are brought in at the bottom on opposite sides of the partition so that the third liquid transmits pressure from one side of the partition to the other. The separating chambers can be provided with electric or steam-heating coils for liquids that must be kept hot to flow freely.

For air or gas, meter connections are made as in Fig. 3–44. The taps are usually made into the top of a horizontal run of pipe, from which the connecting lines extend vertically upward 1 or 2 ft before bending toward the meter, to keep the lines free from moisture. It is immaterial whether the meter is mounted above or below the pipe-line level.

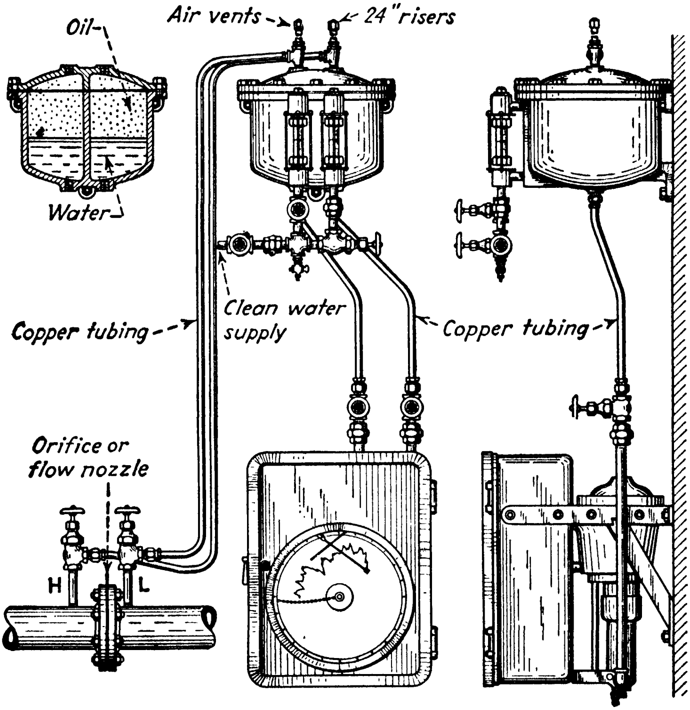


Fig. 3–43.—Sealing chambers keep corrosive fluids out of the meter element.

When the flow is pulsating, such as steam to an engine or water from a reciprocating pump, a flowmeter of the head type reads high. The inertia of the mercury or oil in the meter body and the water in the connecting piping prevents the meter from responding correctly. In general, if the pulsations do not exceed 5 per cent of the static pressure, the meter accuracy will be satisfactory. Pulsations can be reduced by installing one or more throttling orifices, a large receiver, an air chamber for water, or a combination of these devices in the line.

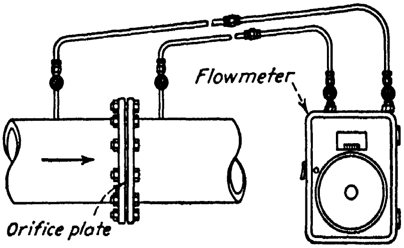


FIG. 3-44.—For air or gas, slope the pressure-tap lines away from the meter element.

CHAPTER IV

TEMPERATURE ELEMENTS

Unlike pressure, temperature or *thermal potential* cannot be measured directly but must be inferred from its effect on a substance whose characteristics are understood. This inference is based on one of the following: expansion of a liquid, gas, or solid; vapor tension of a liquid; change of electrical resistance; electrical potential produced by dissimilar metals in contact; intensity of total radiation or of a particular band of wave lengths of radiation given off by a hot substance; and change in state of a liquid, gas, or solid.

The liquid-expansion-in-glass thermometer is one of the most commonly used devices for measuring temperature in industrial processes and general utility services. Shapes vary, since the tube may be (1) bent at an angle with the scale to meet installation requirements (Fig. 4–1*B*) or (2) adjustable within a narrow range (Fig. 4–1*A*). The first type uses a sealed-guard bulb to measure the temperature of fluids under pressure, and the second uses an open-guard bulb to measure air temperatures.

A pencil thermometer for laboratory and plant tests carries the scale markings directly on its glass tube (Fig. 4–2). For use in pipes or vessels, a separate well is screwed into the container wall and filled with a heat-transfer liquid such as oil or mercury, and the thermometer is inserted (Fig. 4–3). A guard screws into the well to protect the thermometer against breakage (Fig. 4–4).

Figure 4–5 illustrates a thermometer that indicates maximum and minimum air temperature. Bulb *A* and part of *B* are filled with a liquid such as alcohol. The connecting capillary, which is calibrated in degrees, contains a quantity of mercury and two metal floats. Although carried along with the mercury, each float has enough friction against the tube wall to hold it in place as the mercury recedes. The shape of the float permits alcohol to pass freely. A small magnet is used to reset the floats.

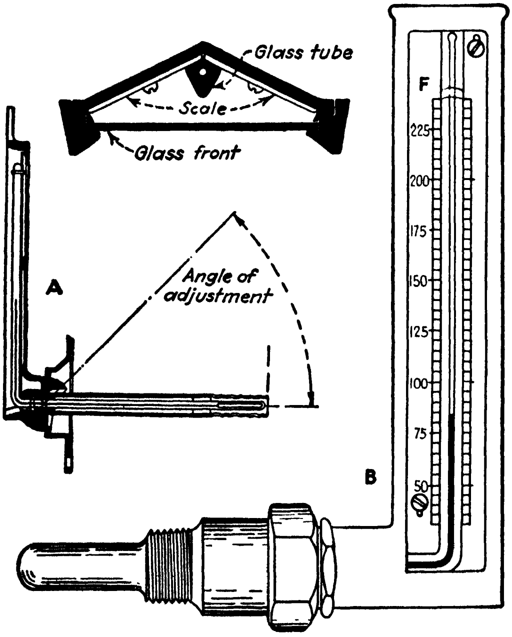


FIG. 4-1.—Thermometer shapes vary. The tube can be bent at any angle to meet installation requirements.

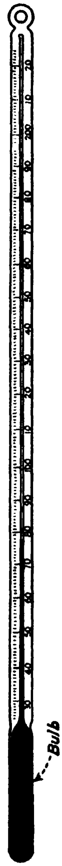


FIG. 4-2.—Pencil thermometer.

In air-conditioning and cooling-tower work, it becomes necessary to find two temperatures of the air—*dry bulb* and *wet bulb*. The dry-bulb temperature is that shown by a regular thermometer. To read the wet-bulb temperature, the bulb is covered with a wet cloth while the air whose temperature is being measured is passed rapidly over the bulb. The thermodynamic wet-bulb temperature

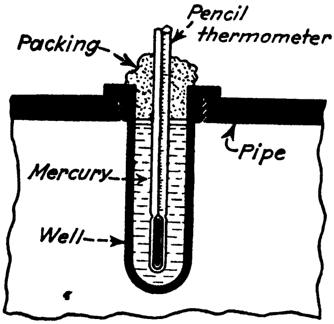


FIG. 4-3.—Pencil-thermometer well.

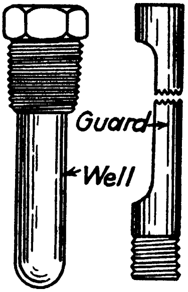


FIG. 4-4.—Thermometer well and guard.

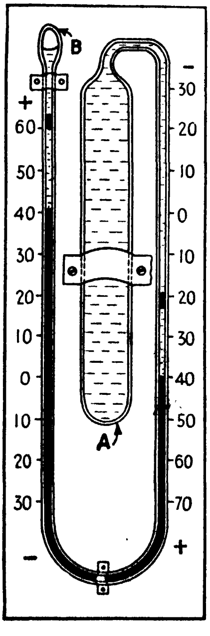


FIG. 4-5.—Maximum and minimum temperature is registered by mercury moving two metal floats.

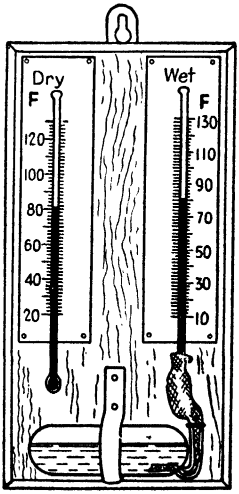


FIG. 4-6.—Dry- and wet-bulb thermometers.

is the temperature at which a liquid or solid (in this instance, water or ice) by evaporating can bring the air to saturation adiabatically at the same temperature. If both the wet- and the dry-bulb temperatures are known, the relative humidity and amount of moisture present can be found from a psychrometric chart. Also the wet-bulb temperature of the entering air is the lowest adiabatic equilibrium temperature to which water passing through a cooling tower can be cooled. In Fig. 4–6, the dry- and wet-bulb thermometers are mounted side by side, and a water container keeps the bulb sleeve constantly wet. Figure 4–7 shows the water pan and cloth arrangement for the bulb of a remote-reading thermometer.

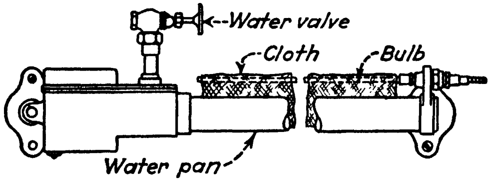


FIG. 4–7.—Water pan and cloth for metal bulb.

Glass thermometers with mercury serve for temperatures of —38 to +650 F; quartz tubes with mercury and filled with nitrogen under pressure (to prevent the mercury from boiling at 675 F) are good for 1000 F. Toluol, alcohol, and pentane serve as the liquid for temperatures of —143, —171, and 328F, respectively. Glass thermometers are calibrated for either total or partial immersion; if total-immersion units are not completely immersed, they read low for temperatures above the surroundings, and vice versa. For accurate results, thermometer indications must be corrected by the amount A for stem temperature.

$$kn\ (T - t) = A$$

where constant $k = 0.00009$ for Fahrenheit and 0.00016 for Centigrade thermometer readings.
$n =$ number of degree divisions thermometer fluid emerges.
$T =$ bulb temperature.
$t =$ stem temperature.

A pressure-measuring device connected to a closed system and filled with an expansible medium can be calibrated to read the temperature at the exposed sensitive surface of the system. The dial or pressure thermometer consists of a bourdon spring or bellows connected through a capillary tube to a hollow metal bulb to form a pressuretight system containing the thermometric fluid (Fig. 4–8). The internal pressure changes with a change in temperature at the bulb and causes the bourdon spring or bellows to expand or contract. The motion of the bourdon-tube tip moves the pointer to indicate the temperature on a dial.

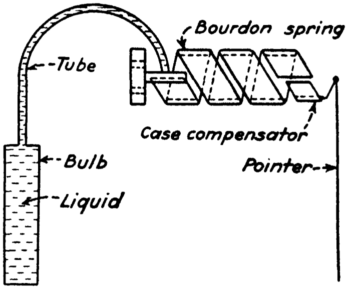


FIG. 4–8.—Liquid-filled thermometer has the bulb, tube, and bourdon spring completely filled under high pressure.

Pressure-actuated thermometers are divided into four classes.

Class 1 uses a liquid-filled system whose operation is based on the volumetric expansion of a liquid (other than mercury) caused by a temperature change. The dial is evenly graduated (Fig. 4–9). The present temperature limit is 750 F.

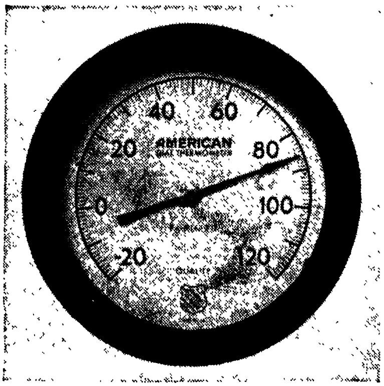


FIG. 4–9.—Liquid-filled thermometer scale is evenly divided.

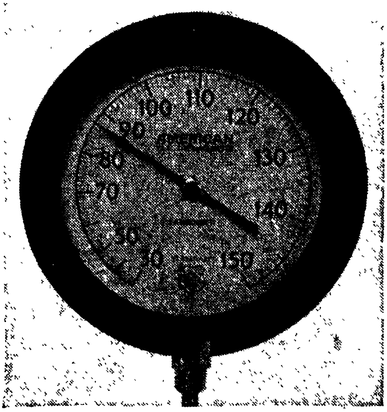


FIG. 4–10.—Vapor-pressure thermometer scale divisions get broader up scale.

Class 2 uses the vapor-tension system in which temperature variation changes the vapor pressure of a volatile liquid such as alcohol, butane, propane, ethyl ether, sulphur dioxide, aniline, ethyl or methyl chloride, toluol, and hexane. The dial or chart scale has uneven divisions that become larger up the scale (Fig. 4–10). The present temperature limit is 750 F.

Class 3 is the gas-filled system (usually nitrogen) in which an inert gas expands with an increase of temperature. The temperature limit is 1000 F.

Class 4 is a special division of class 1, definitely specifying mercury. The temperature limit is 1200 F.

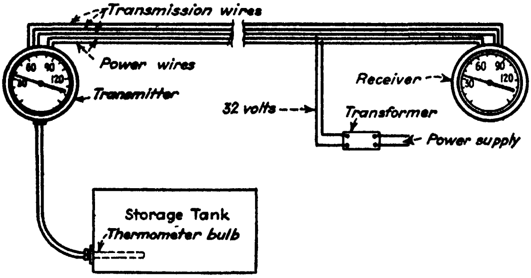


FIG. 4–11.—Mercury-filled thermometers are powerful enough to operate transmitters.

In general, capillary tubing for these instruments should not be longer than 200 ft, because the time lag between the bulb and the instrument may cause difficulties, especially when used with a controller.

Liquid-filled thermometers (classes 1 and 4) are highly evacuated and filled solid with a liquid under initial pressure (Fig. 4–8). Since the mercury-filled instrument combines all the advantages of the devices in the other three classes and is typical of class 1, it will be described first. The bulb, capillary tube, and actuating device are made of steel or steel alloy, which will not amalgamate with mercury. Joints are welded to provide a pressuretight system of great strength.

Good practice calls for internal working pressures as high as 2,500 psi at the temperature-range maximum. This high pressure permits the use of a stiff bourdon tube; and, since the pressure change in a mercury system is much greater than in other liquid-filled units, this device can drive additional indicating instruments such as the electrical transmitter in Fig. 4–11. Scale graduations are uniform over its entire range.

Temperature changes along the capillary tube (Fig. 4–12) introduce errors that must be compensated for. Because this tube is filled with mercury, it may be considered as a second bulb, its effect on the pointer deflection depending on the ratio of the mercury volumes in the capillary tube and bulb. Using extremely fine bore tubing minimizes the error.

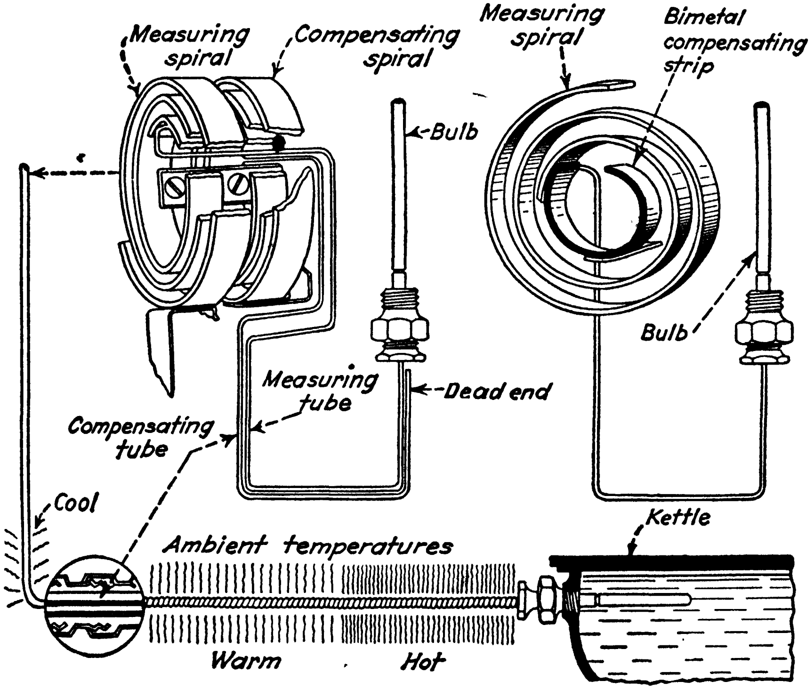


FIG. 4–12.—Ambient temperature changes along the capillary tube introduce errors that must be compensated for, or the instrument will be inaccurate. (*Courtesy of Brown Instrument Co.*)

Since line error is directly proportional to capillary length (a function of volume) and there is a limit to how small the bore can be made, additional compensation must be incorporated in the instrument for long capillary tubes. The simplest method of partial compensation connects one or more extra bourdon springs to the mercury system (Fig. 4—13) in parallel with the actuating spring but keeps them free of the pointer linkage. One spring absorbs one-half the mercury expansion, thus cutting in half the line-error effect on the main spring. Line error reduces in exact ratio to the number of extra springs added, but the bulb volume must be increased in direct proportion to the number of springs added.

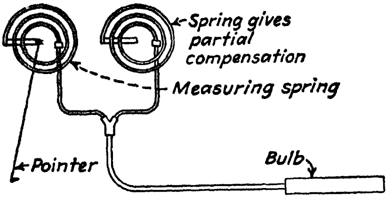


Fig. 4-13.—Second bourdon spring (not connected to pointer) gives partial compensation for ambient temperature changes.

Full compensation is obtained by using an extra bourdon spring and capillary tube, but no bulb (Fig. 4-12). The extra capillary is the same length as the operating tube and encased in the same protecting armor. The two bourdon springs connect to the pointer through differential linkage so that the compensating-spring movement subtracts from the operating-spring movement (Fig. 4-14) to position the pointer according to the temperature change at the bulb.

One form of capillary compensation uses a relatively large bore capillary tube, which contains a full-length invar wire (Fig. 4-15). The coefficient of expansion of the wire is so related to that of the tube and mercury that any expansion or contraction of the tube and mercury is accurately counteracted at the point of fluctuation. Units of this type in which the compensation does not place an extra bourdon spring inside the instrument case must be compensated for temperature variations inside the case that affect the bourdon spring. The compensator usually consists of a bimetallic strip (Figs. 4-12 and 4-15) connected between the actuating spring and pointer in such a way that it neutralizes any movement of the actuating spring caused by temperature changes in the case.

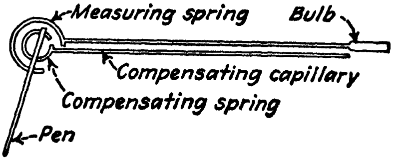


Fig. 4-14.—Second capillary tube and bourdon spring compensate for temperature variations.

Mercury-filled pressure thermometers must be corrected for bulb elevation only when this elevation exceeds a certain height. Because mercury exerts a hydrostatic pressure of 6 psi per ft height, this pressure adds up materially when the instrument and bulb are located at widely separated elevations. On the other hand, mercury-filled units are assembled and operate under high initial pressures, 200 psi minimum and 2,500 maximum; and small differences of several feet in elevation do not require correction. A mercury-filled thermometer bulb averages the temperatures to which it is exposed. This eliminates the effects of temperature stratification or zones. Bulbs can be 10 to 15 ft long for flue-gas measurement. These units respond quickly to bulb-temperature changes (small thermometric lag), and their great power makes them well suited to carry the added load of extra indicators. The unit in Fig. 4–11 operates a selsyn motor to transmit readings to a distant point.

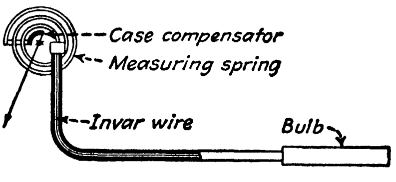


FIG. 4–15.—Invar wire and bimetallic strip provide compensation. (*Courtesy of Taylor Instrument Cos.*)

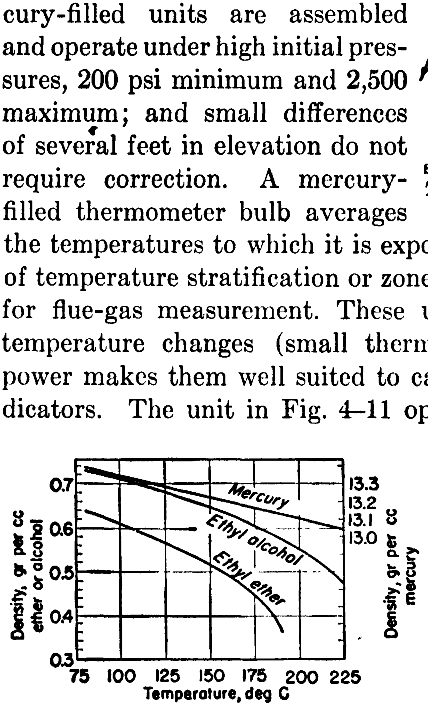


FIG. 4–16.—Temperature-density curves of filling liquids.

Thermometers using filling liquids other than mercury have the same handicaps that affect mercury units as well as other peculiarities. These liquids have coefficients of expansion three to eight times that of mercury and also higher factors of compressibility. For instance, ether is compressed thirty-three times and alcohol twenty-two times more than mercury when under 150 psi pressure. Figure 4–16 compares density-temperature characteristics of mercury, alcohol, and ether. Under certain conditions, a mercury-filled thermometer will exert four times as much power as one filled with ether. Since line error is nearly proportional to the coefficient of expansion of the liquid, it will be greater in a thermometer filled with other liquids than in one filled with mercury.

Because other filling liquids have a higher coefficient of expansion than mercury, their bulbs can be smaller for the same temperature range. The smaller bulb helps to reduce lag; but, since the heat conductivity of mercury is forty to fifty times greater than for other liquids, the over-all sensitivity of the mercury-filled unit is greater. As with mercury, the other liquid fillings have an initial internal pressure of several hundred pounds per square inch and, from a practical standpoint, no correction for bulb elevation is necessary.

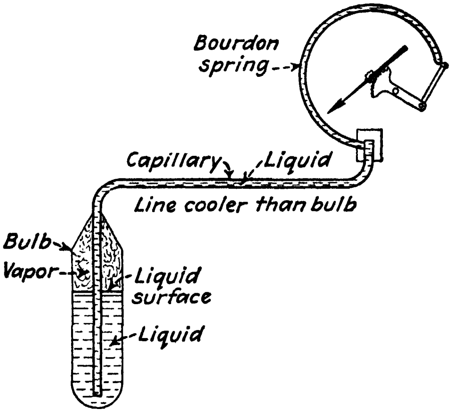


FIG. 4–17.—In the vapor-pressure device, the bourdon spring and capillary tube are filled with liquid.

Vapor-pressure thermometers derive their name from being actuated by the vapor pressure of a volatile liquid contained in the pressuretight system. If the system is evacuated and then partly filled with a volatile liquid (Fig. 4–17), a definite relation exists between the liquid temperature and the pressure of its vapor. To assure a true vapor pressure at all times, the system is so filled that the bulb always contains a free liquid surface and vapor space. Capillary lengths may be up to 200 ft, and no particular limitations apply to bore size except that, if it is too small, fluid friction will cause some lag in the instrument response. Vapor pressure at a given temperature is different for the various liquids (Fig. 4–18).

Because power for operation depends on the pressure increase

for a given increase in bulb temperature, the vapor-pressure thermometer is most powerful and accurate in the upper portion of its range. It is important therefore that these units be selected so that the working point falls in the upper half of the scale and that the range is as small as possible. Scale graduations become progressively wider from the bottom to the top of the dial.

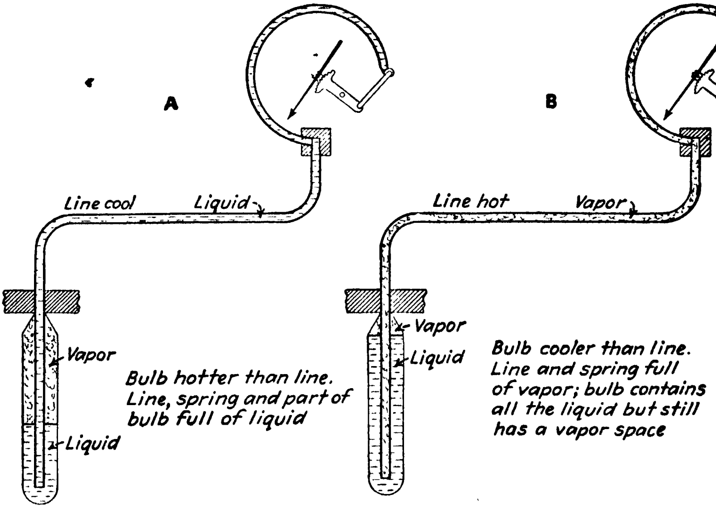


FIG. 4–19.—Condition inside a vapor-pressure thermometer when bulb is hotter and colder than capillary tubing.

As long as the bulb is the hottest part of the system, the line and bourdon spring are filled with liquid (Fig. 4–19*A*). If the line and spring become hotter than the bulb, they will be filled with vapor, and all the liquid will be in the bulb (Fig. 4–19*B*). In either case, the free liquid surface is in the bulb, and the correct bulb temperature will be indicated as soon as conditions reach equilibrium.

When the line and spring first become hotter than the bulb, the liquid momentarily becomes superheated and unstable although under the same pressure as that in the bulb. Vapor bubbles soon form that break the liquid column in the line, progressive flashing continues, and the pointer will not indicate true temperature until all the liquid has been vaporized or forced into the bulb and cooled to bulb temperature. During this process, pressure surges build up in the line, as in Fig. 4–20. When the line and spring cool down, the action of the pointer is again erratic until all the vapor in the

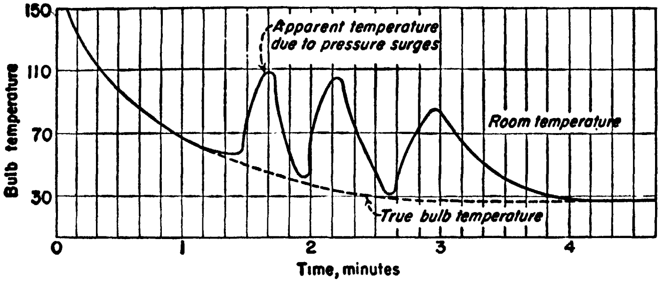


FIG. 4–20.—Pressure surges inside a vapor-pressure thermometer as tube temperature rises above that of bulb.

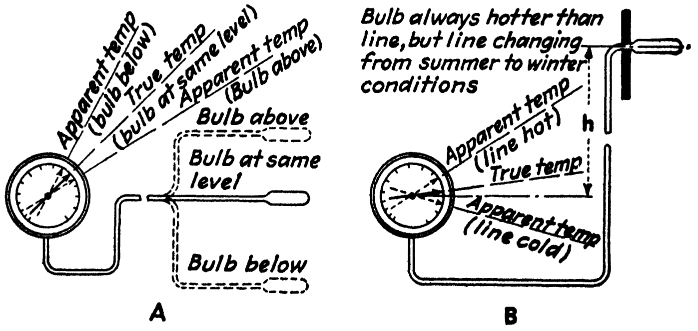


FIG. 4–21.—Positive and negative bulb elevations affect vapor-pressure-thermometer accuracy unless capillary tube always remains full of vapor.

line condenses. If these reversals are likely to occur in service, specify that the instrument be provided with necessary compensation. This is a must for a bulb installed with either a positive or negative elevation in relation to the indicator.

Vapor-pressure thermometers must be corrected for either positive or negative bulb elevations unless the line and spring always remain filled with vapor. When the bulb is above the instrument, the pointer is set back; likewise, with the bulb below, the pointer is set ahead so as not to indicate the hydrostatic head (Fig. 4–21*A*). Distant-reading instruments using an elevated bulb are subject to slight error resulting from temperature changes along the line. Hydrostatic head depends on liquid density, but density changes with temperature (Fig. 4–16). The pointer adjustment for hydrostatic head is therefore correct for only one line temperature (Fig. 4–21*B*); but the error can be reduced by correcting for bulb elevation at the mean line temperature, using a filling liquid of low coefficient of expansion and selecting as stiff a bourdon spring as possible.

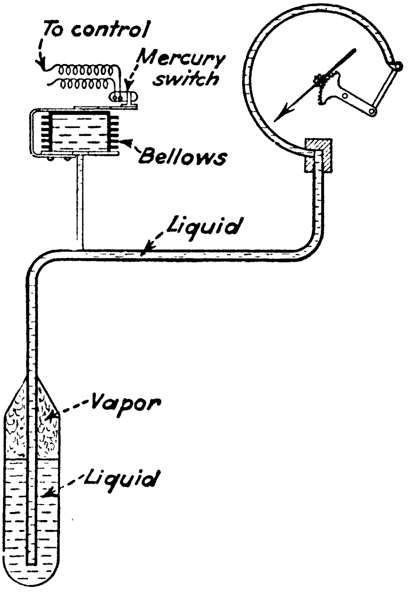


FIG. 4–22.—Vapor-pressure unit is powerful enough to operate one or more responsive elements.

A vapor-pressure system will operate one or more responsive elements from the same bulb without using special line compensation or bulb size. For instance, one bulb can operate a bourdon-spring indicator and a bellows-operated switch or other controller (Fig. 4–22). These thermometers do not indicate the hottest temperature to which the bulb is exposed. The temperature at the free liquid surface in the bulb determines the vapor pressure and thus the temperature registered by the pointer, which will be a value between that of the hottest and that of the coldest part of the bulb. They have a small inherent thermometric lag, *i.e.*, they give quick response.

A combination vapor-tension and liquid-filled system (Fig. 4–23) has the spring, capillary tube, and a bellows completely filled with liquid. The bellows provides a seal between the capillary and the bulb; the latter contains a quantity of volatile liquid whose vapor pressure acts on the bellows to position the instrument pointer. Practically, the device is self-compensating for line error, because any temperature change along the tubing expands or contracts the liquid, which does nothing more than reposition the bellows. This has no effect on bulb vapor pressure, because it depends on the temperature and not on the volume of the bulb. This unit substitutes ideally for a vapor-pressure instrument where line temperatures are likely to vary above and below bulb temperature.

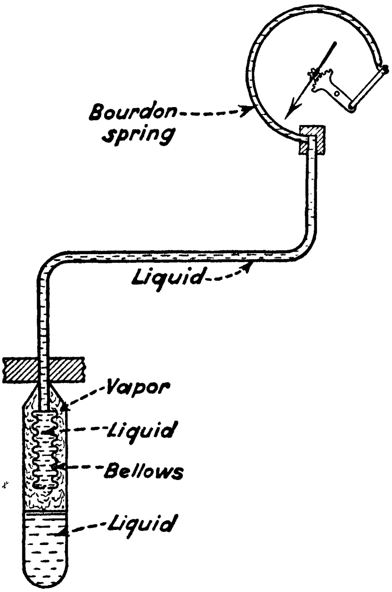


FIG. 4–23.—Combination vapor-pressure and liquid-filled system uses a bellows inside the bulb.

Gas-actuated thermometers are filled with inert gas, usually nitrogen. Operation is in accordance with the gas laws: If the gas volume is constant, its pressure will increase if the temperature of all or any part of the gas is increased. If all the gas were contained in the bulb or if the bulb, line, and spring were heated to the same temperature, the pressure rise would be directly proportional to the absolute temperature, Fig. 4–24*A*. Usually, only the bulb is heated; therefore, some gas must expand into the line and spring, making the final pressure slightly less because of their volume *B*.

Just as in a liquid-filled thermometer, the capillary and bourdon spring introduce an error from temperature effects along the line. This error can be reduced to tolerable limits but never eliminated by the usual methods of compensation, because it is a function of the square of bulb temperature rather than only a function of line temperature as in a liquid-filled system. Gas-filled thermometers are not subject to bulb elevation errors and are free from bulb-and-line temperature reversals. They are accurate (the hydrogen-gas unit is accepted as a primary standard in thermometry for temperatures within its scope) but lag in responding to temperature changes.

The unit has small power, and the indicating mechanism must be light.

Because of larger diameter bulbs—used to reduce line error—compressibility of the gas, and slower heat-absorption rate, gas thermometers respond more slowly than mercury or vapor-pressure devices. The order of quick response is vapor-pressure, liquid-filled, and gas-filled systems. Where sockets or wells are required, their heat-transfer characteristics have a lot to do with the instrument response. The over-all thermometric lag (speed of response) in pressure thermometers is influenced by the kind of filling fluid, size and shape of bulb, length of capillary and size of its bore, and the fluid surrounding the bulb. The lag is greater if the bulb is in air, gas, or superheated steam than if it is in a liquid or saturated vapor. Agitating the surrounding liquid further reduces lag.

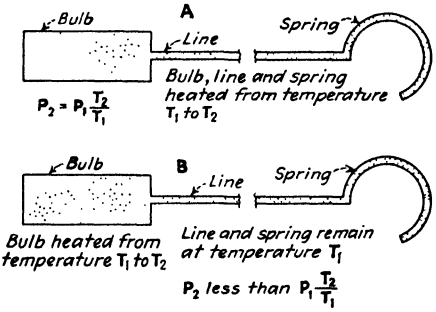


FIG. 4–24.—The pressure inside a gas-filled system increases if the temperature of any part rises.

In most processes, the material in kettles, tanks, and vats must be kept constantly in motion, and the temperature varies throughout the material. General preference is to locate the thermometer bulb where the velocity is highest, because this point gives the best representative temperature and the minimum transfer lag. Always immerse the entire bulb in the fluid being measured. Where protective sockets or wells are required, select one having walls as thin as possible. Carefully consider differences in expansion between the bulb and the socket and, make the fit as neat as possible. This speeds up heat transfer. Adding a liquid to the socket also helps.

Sockets are a necessity on high-pressure vessels and in corrosive or erosive liquids. They permit removing the bulb for repairs or testing but introduce lag in instrument response.

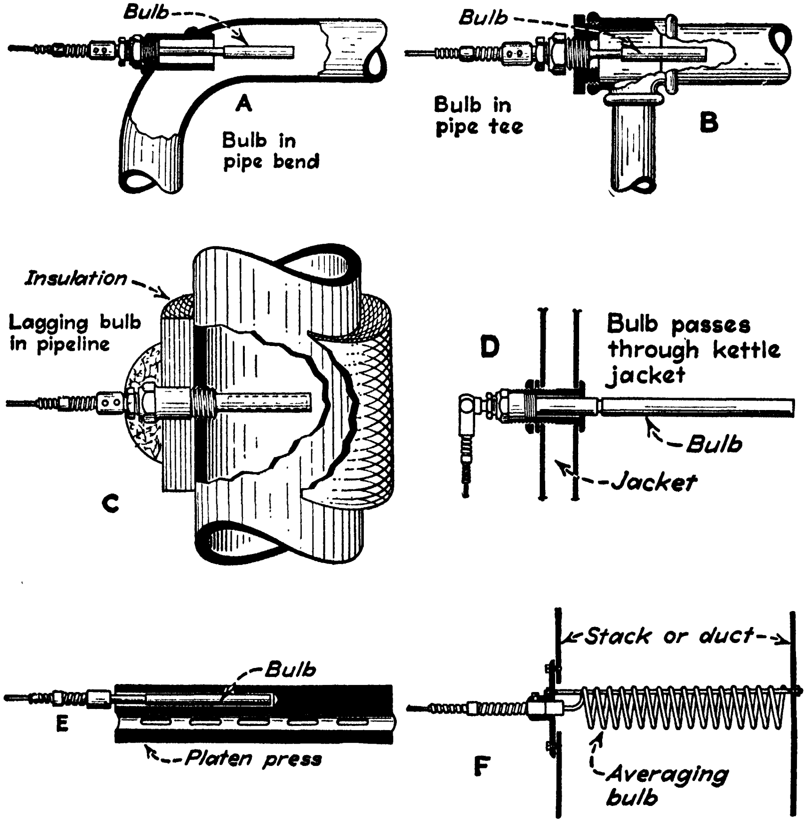


FIG. 4–25.—For best results, put the thermometer bulb in such a place that flow over its entire surface is assured. (*Courtesy of The Bristol Co.*)

The best location for a bulb in a pipe line is at a bend (Fig. 4–25*A* and *B*). This permits using a long bulb and assures flow over its entire surface. If the pipe or vessel is insulated, use a socket with an extension neck so that the lagging can be brought up close and over the extension if necessary (Fig. 4–25*C* and *D*).

For platen presses, drill the bulb holes for a slide fit (Fig. 4–25*E*). Because gas temperatures in chimneys, ovens, driers, and dry-houses are more variable and difficult to measure accurately than liquid temperatures, a long bulb serves best to obtain the average temperature. If the space is limited, use a coil unit and support it as in Fig. 4–25*F*.

Radiation also becomes important when measuring gas temperature, because heat transfer from the gas to the bulb is relatively poor, and the heat may easily be lost by radiation to surrounding objects. The best solution is to place the bulb in the highest velocity path and place polished metal shields between it and other objects such as duct walls. If a bulb is inserted in a duct bend, it will register a higher temperature in the outer radius than in the inner one, because the gas velocity is higher at the outside wall of the bend. In every case, locate the bulb where it will not be damaged by agitators or workmen climbing ladders to and from manholes. Rotating drying rolls and retorts require special fittings where the bulb enters the roll shaft (Fig. 4–26).

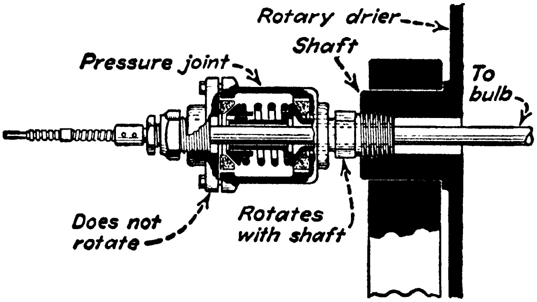


FIG. 4–26.—Use special fittings on drying rolls and retorts.

Here is a 12-point comparison of the four types of pressure thermometers:

Mercury-filled Types

1. Solidly filled with mercury under high initial pressure. Deflection of bourdon spring measures direct expansion and contraction of mercury as bulb is heated or cooled.
2. Most expensive to build. Entire system all-steel welded construction.
3. Does not ordinarily require correction for bulb elevation.

4. Not affected by reversal of bulb and line temperatures.

5. Compensation required for error of long lines or for large changes in line temperature. Compensation may be either partial or full, as required.

6. Working range between — 40 and + 1200 F. Range can be as short as 30 and as long as 1200 F.

7. Scale graduations uniform. Accuracy high over whole scale.

8. Bulb relatively small, especially for long ranges. Bulb volume varies inversely with length of range.

9. Bulb can be made quite long to obtain average temperature. Is true averaging type.

10. Has good heat sensivitity, influenced by size and proportions of bulb.

11. Most powerful and rugged of all types.

12. Will not stand appreciable overtemperature, unless specially protected.

Vapor-pressure Types

1. System partly filled with volatile liquid. Bourdon spring deflected by vapor pressure of liquid in bulb.

2. May be fabricated entirely of nonferrous materials.

3. Will not have appreciable line error unless bulb has considerable elevation above or below instrument case, when a pointer correction must be made for hydrostatic head. Should never be installed with bulb elevated if range is such that bulb and line temperatures may cross or reverse each other, unless system is specially compensated.

4. Erratic pointer fluctuations if bulb and line temperatures reverse, unless system is specially compensated.

5. Requires compensation if bulb is elevated.

6. Large selection of ranges, including temperatures as low as — 40 and as high as 750 F, but not in one range.

7. Scale not uniform. Divisions increase progressively through range. Accuracy poor in lower portion of range.

8. Bulb can be made small if it is always hotter than line and case; otherwise bulb must be large enough to take all liquid from line and spring.

9. Will not indicate true average or hottest temperature of bulb. Bulb should be kept as short as possible.

10. Has good heat sensitivity. Suitable for indicating rapidly changing or fluctuating temperature.

11. Inherently sturdy and accurate, provided that filling-liquid and bourdon-spring characteristics are properly matched.

12. Must be protected against even moderate overtemperature.

Gas-filled Types

1. Filled with an inert gas under initial pressure. Bourdon spring deflected by changes in pressure as gas in bulb is heated or cooled. Absolute pressure in system nearly proportional to absolute temperature.

2. Can be fabricated entirely of nonferrous materials.

3. Does not require correction for bulb elevation.

4. Not affected by bulb and line temperature reversals.

5. Does not normally require line error compensation other than bulb, which must be large to reduce line error. Large bulb may be difficult to install.

6. Working ranges between — 60 or less and + 1000 F. Range can be as short as 150 and as long as 1000 F.

7. Scale graduations uniform, accuracy high over whole range.

8. Bulb can be made quite long, but minimum length is limited.

9. Is not true averaging type but will indicate approximately average temperature if temperature differences along bulb are not too great.

10. Large bulb and poor heat conductivity of gas reduces heat sensitivity.

11. Less rugged than other types.

12. Will not stand appreciable overtemperature.

Liquid-filled Types

1. Solid filled with a noncorrosive liquid under moderately high initial pressure. Bourdon-spring deflection measures direct expansion and contraction of liquid as bulb is heated or cooled.

2. Less expensive to manufacture than mercury-filled when compensation is not required. Can be made of copper, brass, or bronze with soldered connections.

3. No error due to reasonable bulb elevation.

4. Not affected by reversal of bulb and line temperatures.

5. Has much greater line error than mercury-filled type. Line-error compensation required except under most favorable conditions.

6. Working range between — 40 or less to about + 750 F. Range may be as short as 30 F.

7. Scale graduations uniform over the shorter ranges; not uniform for ranges much longer than 300 F.

8. Bulb inherently small; volume varies nearly inversely to length of range.

9. Is of true averaging type but, because of small volume, difficult to build bulb sufficiently long for averaging.

10. Small bulb gives high heat sensitivity.

11. Stands next to mercury-filled type for ruggedness. Has approximately 25 per cent power of mercury-filled type for given change in temperature.

12. Will not stand appreciable overtemperature.

When orders or inquiries covering pressure thermometers are sent, the following information should be given to the manufacturer:

1. Type preferred: mercury, liquid, vapor pressure, gas.
2. Working range: give maximum, minimum, and working temperature.
3. Temperature scale: Fahrenheit or centigrade.
4. Line length: after fabrication, the length cannot be increased except by complete rebuilding.
5. Size of dial preferred.

6. Type of bulb required: plain, union-connected, separable-socket, extra-sensitive, averaging.

7. Nature of fluid in which bulb or socket will be immersed; if fluid is corrosive, give material from which container is made, with alternate recommendations for other suitable materials. Will line armor be subjected to corrosion? If so, of what nature?

8. Size of bulb: give minimum and maximum size that can be installed.

9. Line temperature: give maximum, minimum, and normal temperatures to which line and spring will be subjected.

10. Bulb elevation: give distance, and whether above or below instrument.

11. Temperature reversals: (*a*) Will bulb always be hotter than line and spring? (*b*) always colder than line and spring? (*c*) sometimes hotter, sometimes colder than line and spring?

12. *Bulb elevation must be given; also variations in line temperatures.*

Bellows have their field of usefulness, mostly for controlling temperature in refrigeration, water heating, air-conditioning, and process work. The bellows itself may contain the filling fluid; or a system consisting of bellows, capillary tube, and bulb similar to those for pressure thermometers may be used. Service requirements determine whether the devices are liquid- or gas-filled or vapor-pressure units. For low-temperature work, vapor-pressure units require highly volatile liquids to produce power enough to operate switches and valves. Since these units are frequently exposed to room temperature, the amount of liquid added is carefully computed to flatten the vapor-pressure curve at temperatures above the working range. Thus the liquid is entirely evaporated when heated to slightly above the working point, and damaging overpressure is prevented.

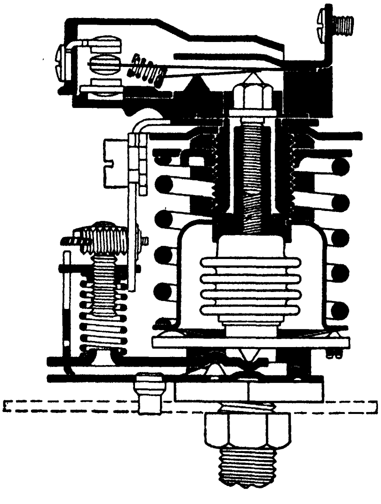


FIG. 4-27.—Switches can be operated by either integral bellows or remotely located pressure-thermometer bulbs. (*Courtesy of Allen-Bradley Co.*)

Units can be designed to function up to a certain temperature and exhibit no further pressure increase above this point. The unit in Fig. 4–27 operates a control switch and can be adapted for use as a room thermostat or, fitted with a bulb and capillary, for process control. The thermostatic switch (Fig. 4–28) serves to start and stop small refrigerating plants. The room thermostat (Fig. 4–29) is a bellows or disk-operated valve used in pneumatically controlled air-conditioning systems. The volatile liquid in this device has a boiling point of 50 F and at 70 F develops about 6 psi, the pressure varying with temperature. Expansion and contraction of the disk under this pressure change operate the supply and exhaust valves and control the compressed-air flow to operate the valve and damper motors. Air wastage is a minimum, because supply and exhaust valves never open at the same time.

With the system at equilibrium, both valves are closed, and the controlled valve or damper is properly positioned. A temperature rise causes the disk to expand and force the supply valve off its seat. Air then enters the exhaust-valve chamber and flows to the controlled-valve motor. When the air pressure in the exhaust-valve chamber acting on the exhaust-valve diaphragm balances the bellows pressure, the exhaust-valve-body assembly moves to the left, and the supply valve closes. When the room temperature falls, the bellows contracts, and the pressure in the exhaust-valve chamber, acting on its diaphragm, moves the exhaust-valve body to the left; the spring opens the exhaust valve until the bellows and air pressure again balance. Thus just enough air is bled to waste to reposition the controlled valve or damper in conformity with the room temperature as felt by the bellows.

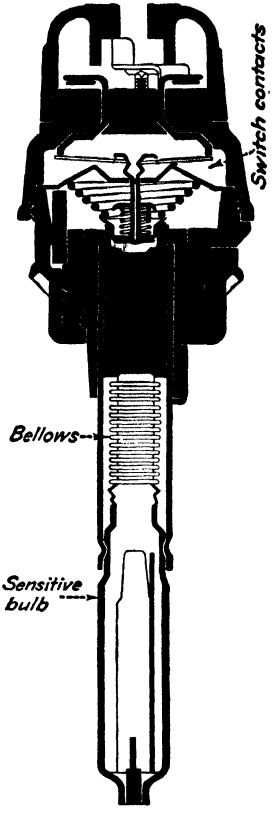


FIG. 4–28.—Bulb- and bellows-operated switch used on refrigerating systems.

The bimetallic temperature-indicating and -controlling device operates on the differential expansion of two different strips of metal brazed, welded, or riveted solidly together. One metal (invar or other nickel-iron alloy) has a low coefficient of linear expansion, and the other (brass or steel alloy) has a high one. (Brass expands twenty times as much as invar per unit of temperature rise.) Thus, if a strip of this bimetal is anchored at one end, temperature changes make one metal expand more than the other. Since they are fastened together, the strip bends, and the movement of the free end can be used to close electric contacts or operate lightweight dampers.

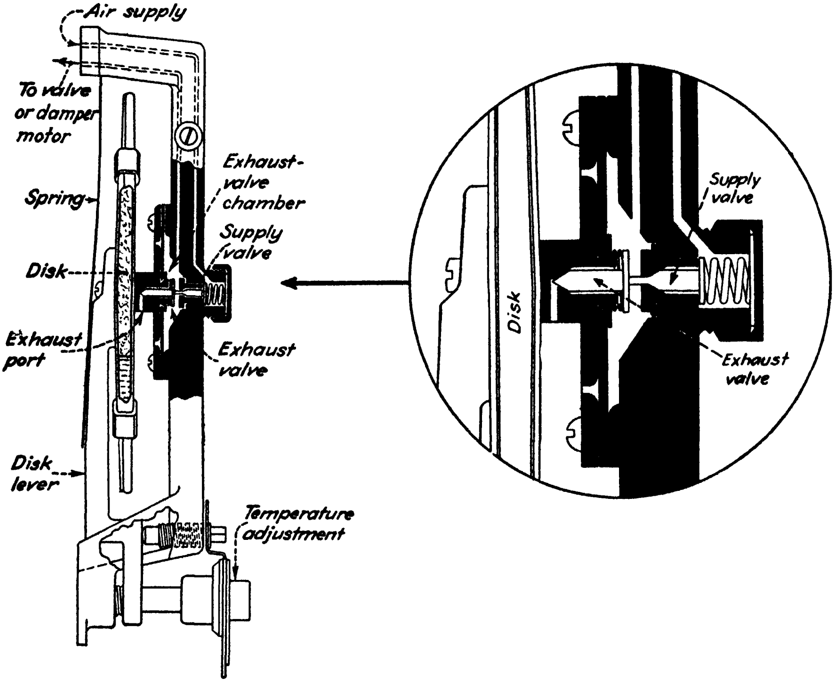


FIG. 4-29.—Vapor-pressure disk serves as a room thermostat to control a pneumatically operated valve or damper. (*Courtesy of The Powers Regulator Co.*)

Increasing the strip length gives greater free-end movement for the same temperature change, and increasing the bimetal thickness increases the force developed by the strip. This additional force is obtained at a sacrifice in instrument response, however, because a longer time is required to bring the thicker metal up to the surrounding temperature. To take advantage of the greater deflection of a long strip and yet keep the device within a reasonable size, the bimetal is coiled in a spiral (Fig 4–30) or bent in some other shape. Fortunately the metals in the bimetallic strip have about the same modulus of elasticity; therefore, they can be designed to work within their elastic limit.

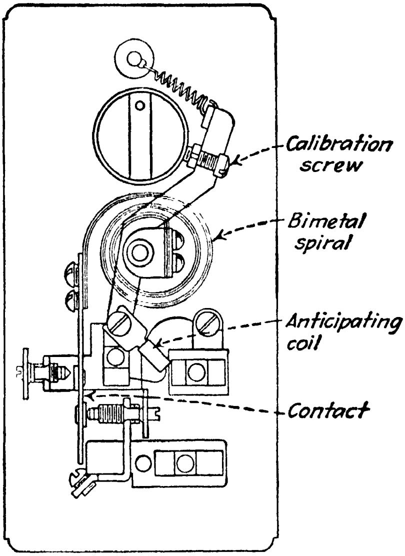


FIG. 4–30.—Coiling the bimetallic strip gives greater contact movement per degree of change. (*Courtesy of White-Rodgers Electric Co.*)

Devices of this kind are not so accurate as pressure thermometers because of hysteresis (lag in movement on temperature reversals), and they serve mostly for temperature measurement or control at atmospheric pressure up to 600 F. Figure 4–31 illustrates the schematic arrangement of a typical bimetal thermostat. When the bimetallic strip bends to close the control circuit, a permanent magnet holds the contacts closed until a temperature rise causes the strip to exert sufficient force to overcome the magnetic pull and open the circuit. Adding this magnetic load to the strip would permit room temperature to overshoot before the strip could exert enough force to open its contact, and a small anticipating coil (resistance heater) is therefore connected into the circuit and placed near the strip. This slight additional heat affects the strip just enough to make up for the magnetic pull. The heater in Fig. 4–31 has its own parallel circuit, but another common arrangement puts it in series with the load.

Thermostats of this kind can be adapted for night setback (to maintain lower temperature at night than in the daytime) by installing another small resistance heater near the bimetallic element. A time clock then energizes and deenergizes the heater to maintain the thermostat several degrees warmer than room temperature during the night.

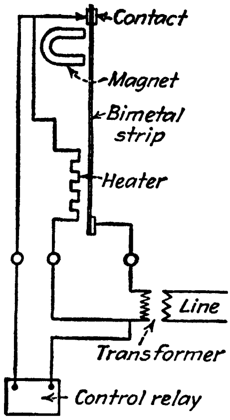


FIG. 4–31.—Anticipating coil (electric heater) opposes magnet to help open contacts. (*Courtesy of Detroit Lubricator Co.*)

The device in Fig. 4–32 consists of two bimetallic spirals mounted inside glass bulb *A* and serves as a flame-failure relay on furnaces. Radiant energy is picked up by concave reflector *B* and focused on small bimetallic coil *C*, which causes it to close contacts *D* and *E*. If the flame fails, the bimetallic element cools and opens its contacts. A compensating bimetallic coil *F* counteracts any effect on coil *C* by surrounding ambient temperature.

The direction of rotation of the turbine wheel (Fig. 4–33) depends on the temperature of the driving gas. The turbine blades of bimetal construction are not held rigidly and therefore bend and reverse their direction of curvature as their temperature goes above or below the fixed control point. At this temperature, the blades are straight, and the turbine therefore does not rotate. Connected through a gear train, the unit positions dampers according to air temperature.

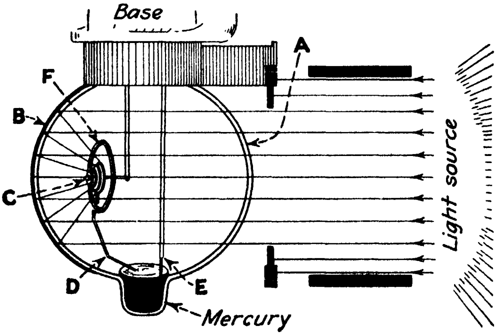


FIG. 4–32.—Bimetallic strip acts as a flame-failure device for furnaces. (*Courtesy of The Mercoid Corp.*)

Instead of a bimetallic strip, a rod of one metal (invar or other nickel alloy) may be enclosed in a tube of another metal (brass, steel, or steel alloy). Figure 4–34 shows how one end of the rod fastens to the expansion sleeve, leaving its other end free to operate a small air valve. Since the tube expands more than the rod, it repositions the free end of the rod on temperature changes. In the unit in Fig. 4–35, a carbon rod is the negative member.

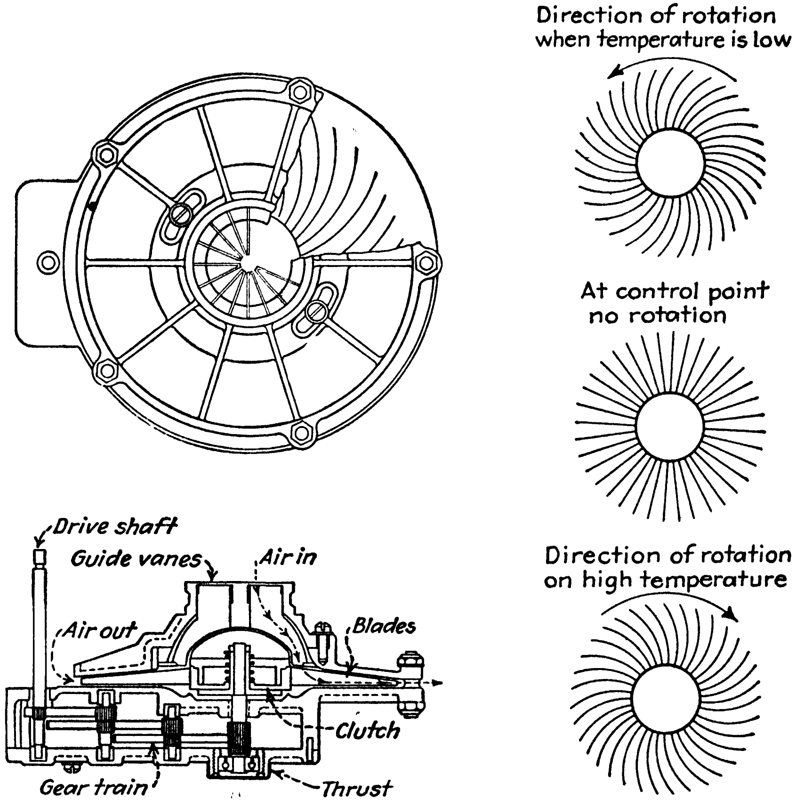


FIG. 4–33.—Bimetallic turbine blades bend according to temperature change. (*Courtesy of General Electric Co.*)

A thermostat sensitive to both air temperature and radiant energy (used to control radiant-heating systems) consists of a hollow copper sphere 6 to 8 in. in diameter and blackened on the outside. An internal sump containing a volatile liquid is fitted with a small electric heating coil. Heat from the coil creates a vapor pressure inside the sphere which remains constant as long as total heat loss from the sphere remains at the desired rate. If the room gets uncomfortably warm, vapor pressure increases because of smaller heat loss from the sphere. This pressure acts on a diaphragm-operated switch to reduce the heat supply to the room. In another instrument of this type, only an electric heater is used to warm the air inside the sphere and expand it against a diaphragm-operated switch.

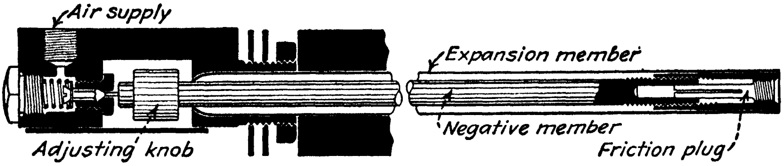


FIG. 4–34.—A metal rod having a low coefficient of expansion fastens to one end of a high-coefficient-of-expansion tube to control an air valve. (*Courtesy of The Powers Regulator Co.*)

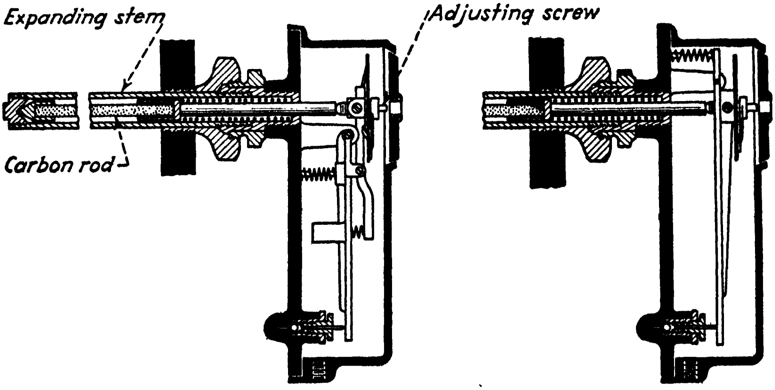


FIG. 4–35.—This device uses a carbon rod as the negative element. (*Courtesy of C. J. Tagliabue Mfg. Co.*)

The resistance thermometer consists of a carefully made coil of nickel or platinum wire having a relatively high coefficient of resistance which serves as the sensitive element (Fig. 4–36). Connected into a Wheatstone-bridge circuit, the resistance of the coil is compared with that of a standard resistance either by automatically adjusting the bridge or by the deflection of a galvanometer. Since the resistance of the sensitive-element coil is always the same at a given temperature and changes with the temperature, the indicating or recording instrument scale can be calibrated in degrees of temperature.

The simple indicating instrument circuit (Fig. 4–37*A*) uses a millivoltmeter calibrated to read temperature and connected across two junction points of the bridge. A battery or other d-c source connects to the opposite points. Coils *a*, *b*, and *c* are standard resistances mounted near the instrument, and *x* is the sensitive coil. The resistance of all four coils is usually the same at room temperature. Any variation in ambient temperature changes the value of the three instrument resistors, but this change is slight and usually can be disregarded. The greatest change occurs in the sensitive coil, which upsets the bridge balance and deflects the millivoltmeter pointer.

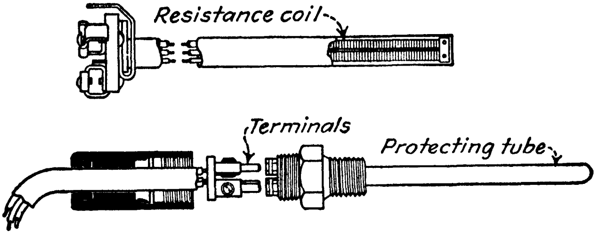


FIG. 4–36.—Resistance search coils with protecting tube and terminal plug. (*Courtesy of The Bristol Co.*)

Another instrument (Fig. 4–37*B*) mounts balancing resistors S and S_1 on a disk and automatically adjusts them according to galvanometer deflection to bring the instrument pointer back to zero. The position of the resistor disk indicates the temperature. Calibration is such that, at any disk position, resistance between contacts V and V_1 is always equal to that between V_1 and the junction A_1. Thus ratio a/b is always unity.

When the disk is rotated to reduce galvanometer deflection to zero, resistance between contact V and junction C_1 is equal to that between junctions C_1 and A_1. Thus an increase in sensitive-coil resistance caused by a temperature change is balanced by including more of the slide wire S in the r arm of the bridge. Accurate indications are obtained by carrying three leads from the instrument to the sensitive coil. Two of the leads, of equal resistance, go from opposite arms of the bridge to the coil, and the third (current supply) has no effect on the bridge balance. The two-lead system is rarely used, because variation in wire resistance affects the bridge balance.

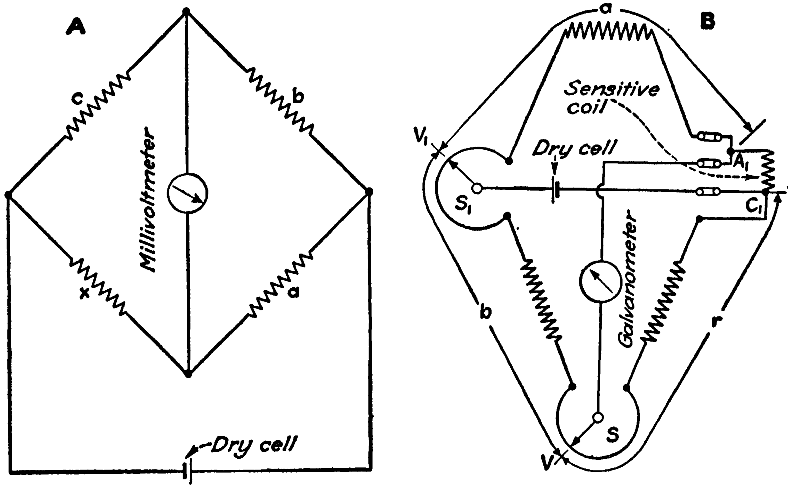


FIG. 4-37.—Wheatstone bridge showing resistance relationship used with a three-lead coil. (*Courtesy of Leeds & Northrup Co.*)

Lead resistance normally varies only with ambient-temperature changes. The three-lead system therefore does not upset bridge balance, because the same resistance change takes place in two arms of the bridge, and hence they cancel each other. Resistance thermometers are good for —330 to +1800 F, and the sensitive element can be located a considerable distance from the indicator or recorder. (The distance is limited only by the resistance of the leads.)

Pyrometric instruments that measure temperature operate on the

(1) fusion, (2) calorimetric, (3) thermoelectric, (4) optical, and (5) total-radiation principles. Fusion cones, made of material having a definite melting temperature, serve for checking furnace temperatures and, when mounted on an insulated rod, for testing live electric conductors and switches for heating. In calorimetric devices, the temperature rise of a known weight of water in a container of known weight and specific heat is used to calculate the temperature of a metallic article of known weight and specific heat placed in the water.

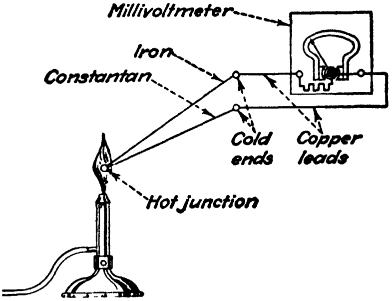


FIG. 4-38.—Heat applied to the hot junction of a thermocouple generates voltage in the meter circuit.

Thermoelectric devices consist of a millivoltmeter or potentiometer circuit, which includes a thermocouple as the sensitive element. The thermocouple is made by fusing one end of two dissimilar metal wires together. The free ends (cold junction) then connect to the indicator circuit, and the fused end (hot junction) is placed where the temperature is to be measured. When the hot junction is heated so that its temperature is higher than that of the cold junction, a small voltage is generated, and a current flows in the closed circuit (Fig. 4-38). Since the voltage generated is proportional to the difference in temperature between the hot and cold junctions, the indicator scale can be calibrated directly in degrees. The cold-junction temperature is assumed to be 32 F, and compensation is added to the instrument to correct for variations; or the cold junction temperature must be measured and a correction factor used. Temperature-conversion equations follow:

$$\text{Temperature F} = 9/5\,\text{C} + 32$$
$$\text{Temperature C} = 5/9\,(\text{F} - 32)$$

NOTE: For temperatures below freezing, the thermometer temperature must be read as plus or minus and the problem solved algebraically.

When copper wires connect the thermocouple to the instrument, the cold junction is at the point where the wires connect to the thermocouple leads. If the dissimilar metal wires that form the thermocouple run to the instrument or selector switch, the cold junction is at their terminals. Thermocouple-metal wires must always be used for leads to instruments that contain cold-junction temperature compensators. Copper and copper-nickel alloy wires serve as leads for platinum and platinum-rhodium thermocouples because they have similar thermoelectric characteristics. Cold-junction correction can be made by a bimetallic element that shifts the pointer zero, a copper-nickel coil whose resistance varies with temperature (Fig. 4–39*A*), or a manually adjusted rheostat (Fig. 4–39*B*).

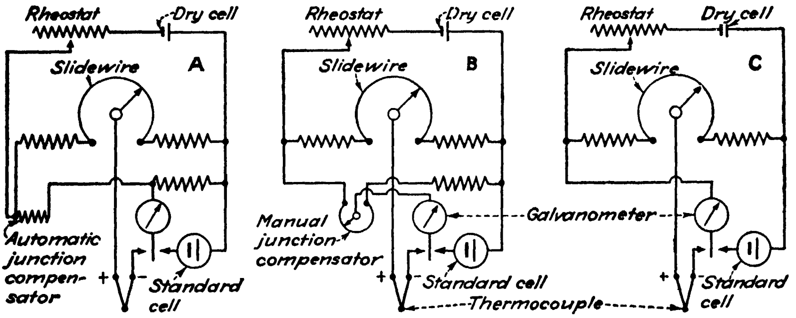


FIG. 4–39.—Automatic resistance coil, manual rheostat, andbimetallic strip (not shown) correct for cold-junction temperature. (*Courtesy of Leeds & Northrup Co.*)

If the instrument has no means of adjusting for variations in cold-junction temperature, results are best if this temperature is held constant. Otherwise the instrument is inaccurate, because the points where thermocouple wires connect to the leads or instrument terminals (cold junction) act as additional thermocouples that generate voltage in the circuit. Holding the temperature constant keeps this voltage at one value and permits the use of one correction factor with meter indications. For practical purposes, the factor is 1 F for each degree of difference between the cold-junction temperature and that at which it was calibrated. For platinum, use 0.5 F.

There is considerable difference in the voltage generated by thermocouples made from various metal combinations (Fig. 4–40).

The emf per degree rise in temperature is the thermoelectric power of a couple and is used as a basis of comparison for the different types. In general, a couple having low thermoelectric power requires a delicate indicating instrument, and one possessing greater power has broad utility.

Although the more common thermocouple metals are shown in Fig. 4–40, the combinations of tungsten-graphite, tungsten-molybdenum, and carbon-silicon carbide have been tested in molten steel. Others considered are graphite-boron, carbon-molybdenum, and carbon-tungsten carbide. Millivolts generated by several of these metals at 3000 F are as follows: tungsten-molybdenum, 2.5 mv; tungsten-graphite, 38 mv; carbon-silicon carbide, 480 mv or almost ½ volt.

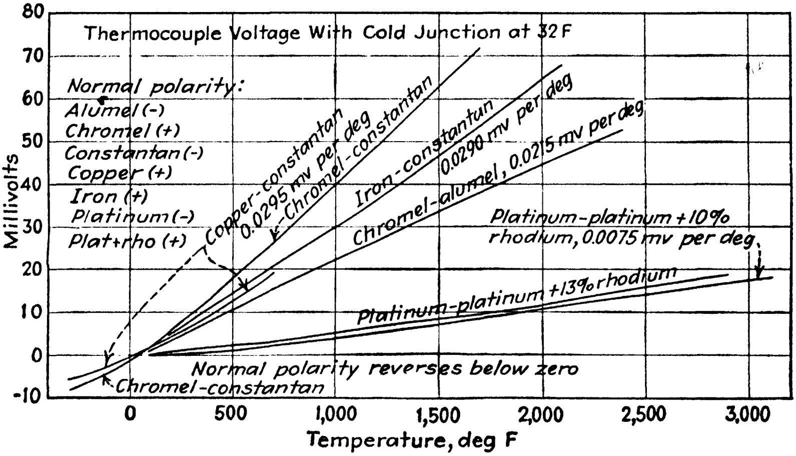


FIG. 4–40.—Thermoelectric power of different thermocouple combinations.

Because perfect junction of the two metals is a necessity, clean the two wire ends down to bright metal, and twist them together tightly two or three times for additional support (Fig. 4–41*A*). Use an oxygen-illuminating gas torch adjusted to give a neutral flame. Hold the twisted junction in the flame until both wires are a dull red. Then dip them in borax flux (no flux on platinum couples), and return to the flame. Keep the wire whose melting point is highest in the hottest part of the flame, and manipulate it until both wires melt into a bead (Fig. 4–41*B*). The bead may also be formed with an electric arc by connecting the thermocouple wires to the positive electrode and drawing the arc with a graphite pencil connected to the negative electrode. Platinum couples are usually annealed at 2700 F by passing electric current through them for several hours.

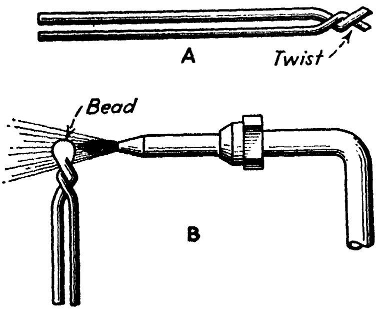


FIG. 4–41.—Twist the wires tightly, and fuse the ends into a bead.

Thermocouple protection (Fig. 4–42) consists of (1) wire insulation, (2) a primary tube to protect the element against contaminating atmospheres, and (3) a secondary tube to protect the primary tube against breakage from falling material or rough handling.

Although it is possible to measure thermocouple temperature with a millivoltmeter, the method introduces the resistance of leads and connections. Consequently, a potentiometer circuit, as in Fig. 4–39, is considered best. Here current from a battery flows through the main circuit of fixed resistors and slide wire. By means of the throw-over switch, a standard cell is periodically connected into the circuit through the galvanometer, and the instrument is standardized by adjusting the battery rheostat until the galvanometer deflection is zero. With the throw-over switch in the opposite position, the thermocouple is connected to the instrument, and the galvanometer is balanced by moving the slide wire. When the potentiometer circuit is balanced, no current flows through the ther-

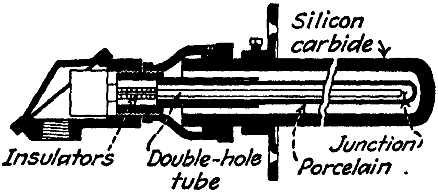


FIG. 4–42.—Protective covers for a thermocouple. (*Courtesy of Wheelco Instruments Co.*)

mocouple leads, and the effect of their resistance does not appear. The position of the slide wire arm can be used to indicate temperature, and the instrument can be made recording by using galvanometer deflection to control a motor that adujsts the slide-wire-arm position. The same mechanism cuts in the standard cell periodically and adjusts the battery rheostat.

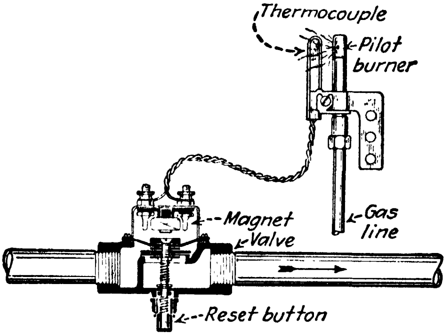


FIG. 4–43.—Flame generates voltage in magnet circuit to hold fuel valve open.

Current generated in a thermocouple can also be used to operate control valves or relays in signaling systems. The unit in Fig. 4–43 is mounted at the pilot burner and holds the magnetic valve open as long as the burner is ignited. If the flame fails, the magnet is deenergized, and the valve closes.

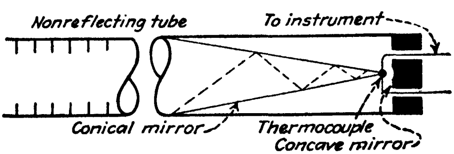


FIG. 4–44.—Total-radiation pyrometer uses conical and concave mirror to focus rays on thermocouple.

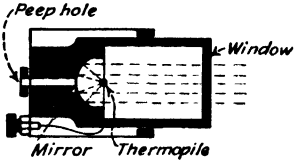


FIG. 4–45.—Peephole facilitates sighting the pyrometer.

Since thermocouples are unsatisfactory for measuring temperatures above 3000 F, instruments that use the radiations from a hot body have been developed. One device, the radiation pyrometer, measures the intensity of all radiations. Another, the optical unit, measures the intensity of those of a particular wave length.

The total-radiation pyrometer concentrates the rays from a particular area of the hot body on a thermocouple or thermopile (group of thermocouples in series), and the thermocouple potential indicates the intensity of all radiation reaching it. The unit in Fig. 4–44 consists of a nonreflecting tube, conical mirror, and concave mirror that concentrate the rays on a thermocouple. Figure 4–45 uses a mirror and Fig. 4–46 a lens to concentrate the rays on a thermopile. The latter device has a nickel coil to compensate for variations in cold-junction temperature. Both instruments employ a peephole for aiming the pyrometer at the hot body. Devices of this kind are available as indicators, recorders, and controllers.

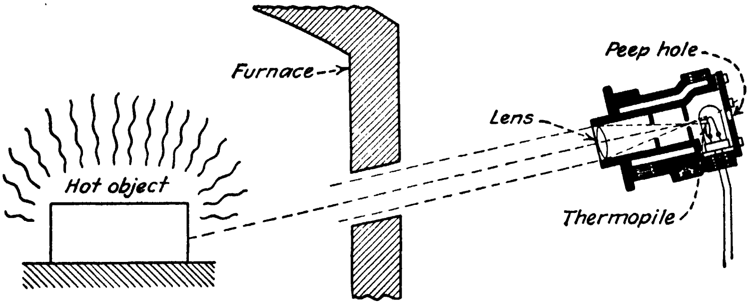


FIG. 4–46.—Pyrometer uses a lens to concentrate the rays on a thermopile.

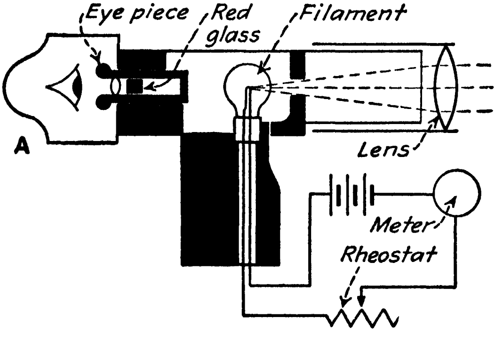


FIG. 4–47.—Optical pyrometer compares radiation with that of a known source.

The optical pyrometer compares the intensity of radiation from a standard source with that emitted by the body whose temperature is being measured. Many types are available, but the commonest method uses the disappearing filament (Fig. 4–47). Light from the hot body enters the lens and is viewed through the eyepiece. The temperature of the filament is then altered by adjusting the rheostat until the filament outline just disappears into the background of the hot object. The filament current deflects the meter pointer over a scale calibrated in degrees. The instrument in Fig. 4–48 uses a potentiometer to measure the filament current. The operator, after regulating the filament brightness by adjusting the rheostat, balances the potentiometer with the slide wire, which is calibrated directly in degrees.

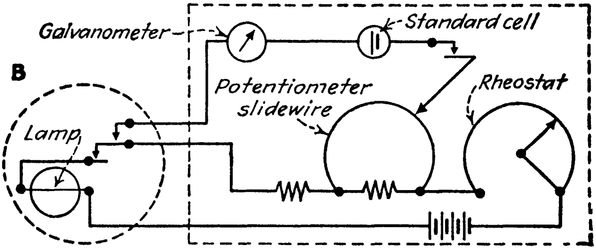


FIG. 4–48.—Pyrometer with potentiometer metering circuit.

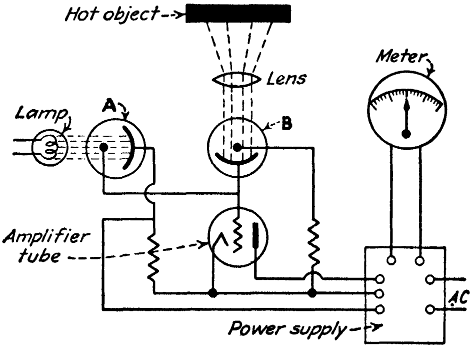


FIG. 4–49.—Photoelectric cells compare hot object with a lamp filament.

For temperatures of 1400 to 2250 F, the filament and object are compared directly; for higher temperatures, the operator inserts a calibrated screen between the filament and the hot object to reduce the brightness to a point comfortable to the eye. This slight disadvantage is overcome by photoelectric tubes (Fig. 4–49), which eliminate the need for visual comparison.

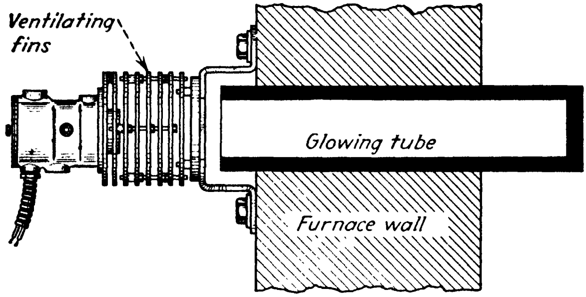


FIG. 4–50.—Pyrometer with glowing tube inserted in a furnace wall. (*Courtesy of The Bristol Co.*)

Tubes *A* and *B* measure light from a standard lamp and that from the hot object for comparison. Both tubes form part of an a-c bridge that includes an amplifier tube instead of a galvanometer. Light from the hot body in striking tube *B* varies its resistance and unbalances the bridge. The amplifying tube adjusts the current flow to the comparison lamp to balance the bridge. The indicator then measures the lamp current in terms of the measured temperature. Devices of this kind also serve as controllers.

A pyrometer can be used to obtain the temperature of molten metal by observation of the metal surface or a second surface at the same temperature. A closed-end refractory or graphite tube (Fig. 4–50) below the metal surface or an open-end tube immersed in the metal and kept clear by an air stream both provide a surface that may be observed.

CHAPTER V

CHARACTERISTICS OF MOTOR-OPERATED VALVES

Control valves are classified as pilot-operated and self-operated. In the first, a separate pilot controls the admission of outside pressure to the valve motor. In the second (Chap. VI), the operating pressure comes from the fluid being handled and acts directly on the operating motor. Both of the above classifications are further subdivided into (1) direct-acting, in which the unit decreases its opening as the operating pressure is increased; and (2) reverse-acting, in which the valve opens wider upon an increase in the operating pressure.

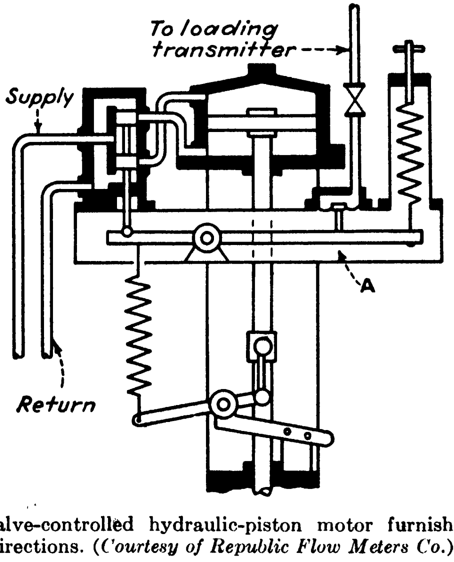


FIG. 5-1.—Pilot-valve-controlled hydraulic-piston motor furnishes power in two directions. (*Courtesy of Republic Flow Meters Co.*)

Pilot-controlled valves, which will be described here, serve best on complicated systems or where the valve requires more power than is available from the fluid being handled. Here the operating power can be taken from an outside source, and the primary instrument response can be amplified pneumatically, hydraulically, or electrically before application to the main valve motor.

Before selecting the valve for a specific job, consider the character of control desired and the conditions under which the valve must operate, as follows: (1) Which motor will furnish the best operating power—a hydraulic-piston (Fig. 5–1), bellows (Fig. 5–2), or diaphragm motor (Fig. 5–3)? (2) Shall the controlled medium be turned on or off if the valve-operating power fails? This determines whether a direct- or reverse-acting valve will serve best (Fig. 5–10*A* and *B*). (3) Is inner-valve construction to be butterfly or globe design? If globe, does the job require a single- or double-seat unit?

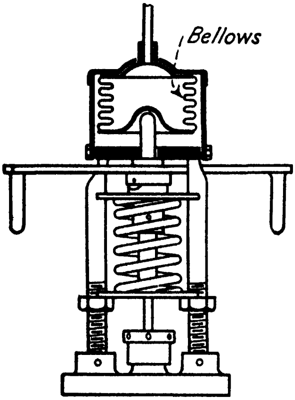


FIG. 5–2.—Single-acting bellows uses an opposing spring.

Mechanical operating power comes from a fluid supplied through a control pilot actuated by a sensitive primary element. Hydraulic-piston motors usually power in two directions, whereas bellows and diaphragm units, generally single-acting, must be opposed by a spring (Fig. 5–4) or weight (Fig. 5–5) that powers the valve in the opposite direction. Because the bellows and diaphragm motors operate similarly, each consisting of a balanced set of equal and opposing forces—a compressed spring or weight opposed by a diaphragm or bellows having a given area and unit of applied pressure—we shall discuss only the diaphragm-operated unit. A change in fluid pressure plus or minus moves the valve stem by acting on a flexible member made of natural or synthetic rubber with a fabric insert or treated-leather. Metal diaphragm valves are described in Chap. VI.

Both spring and weight loading have certain advantages, but valve design and the service it is to render determine the choice. Spring loading is constant only at a given compression; the tighter

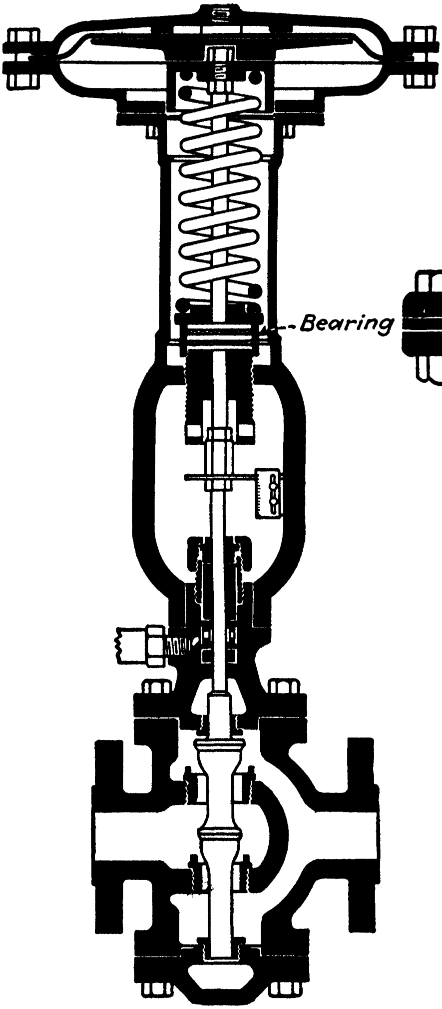


FIG. 5-3.—Flat diaphragm uses a loading spring to produce movement in two directions. (*Courtesy of Fisher Governor Co.*)

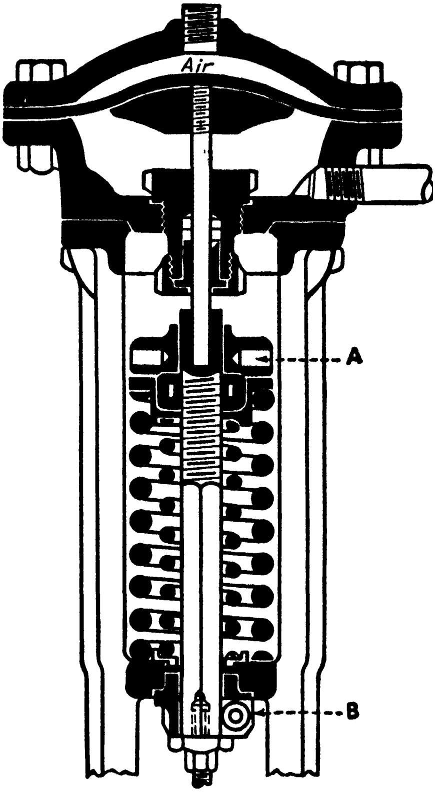


FIG. 5-4.—Rounded diaphragm head with spring loading and roller guides for the stem. (*Courtesy of A. W. Cash Co.*)

a spring is compressed the more force it exerts. A spring-loaded valve gives its best performance when the flow is constant; a variable demand that requires numerous valve positions changes the spring loading at each new flow. Weights are preferable under some conditions, because effective loading remains constant regardless of the valve position.

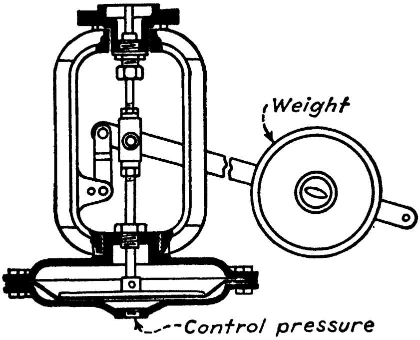


FIG. 5-5.—Weights give constant effective loading regardless of position.

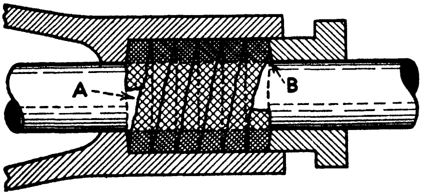


FIG. 5-6.—Rope packing tightens unevenly.

Diaphragm power is absorbed by raising the weight or compressing the spring, overcoming friction and resisting unbalanced load thrust. The motor must have a large margin of power so that these outside influences do not cause unwanted valve movement. This means a large diaphragm with a correspondingly powerful spring. The unit in Fig. 5-4 uses a roller-bearing spring-adjusting button *A* to prevent transmission of the torsional movement of the spring, which occurs during valve movement, to the valve stem, increasing its friction. Roller guides *B* eliminate side strain and keep the stem aligned, which reduces its friction and increases packing life.

When one spring works inside the other, they are wound in opposite directions to prevent intermeshing of turns. To give uniform load distribution and stress, the outer spring usually carries 60 to 70 per cent of the total load, the inner spring carrying the remainder. In dismantling a valve using inner and outer springs, the latter may be found to have a greater over-all length than the inner one. Do not attempt to equalize the lengths by grinding the long one or shimming the short one, because this may upset the loading of the two springs.

Stem friction must be held to a minimum by installing new packing instead of overtightening old packing. It should also be kept lubricated. When inserting new material, use individual rings. Winding the coil packing around the stem until the gland is full, even though it hugs the gland wall instead of the valve stem, produces the condition in Fig. 5–6. Pressure on the overlapping ends causes one section *A* to be filled with loose fibers and another section *B* to be filled with unduly compressed fibers. To stop leakage through loose areas, the fibers already compressed must be squeezed still further. This not only extrudes initial lubricant from the packing but also increases friction and wear on the valve stem. The stick lubricant placed in chamber *A* (Fig. 5–7) is forced in the gland by turning setscrew *B*. On valves not equipped with a lubricator, apply several drops of oil to the stem packing gland occasionally.

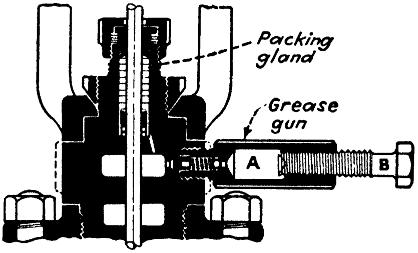


FIG. 5–7.—Jackscrew and grease pocket permit easy lubrication of stem guides and packing.

Diaphragms, usually of natural or synthetic rubber, have fabric inserts and may be flat disks or molded to predetermined shapes for maximum flexibility. Shapes vary. Figures 5–3 and 5–5 fit flat diaphragm plates, and Fig. 5–4 is for a rounded one. To produce maximum power throughout valve travel, the diaphragm must remain in full contact with the moving *plate* (Fig. 5–8*A*). Stretching or bowing away from the plate *B* decreases the effective area as travel increases (Fig. 5–9).

Frequent grinding of valve seats or incorrect adjustment of the valve-stem length may cause separation that decreases the power output sharply just when the spring force is greatest. This means that a unit in pressure increase makes the valve travel farther at the beginning of its stroke than near the end, and valve travel is not proportional to diaphragm pressure throughout the full stroke.

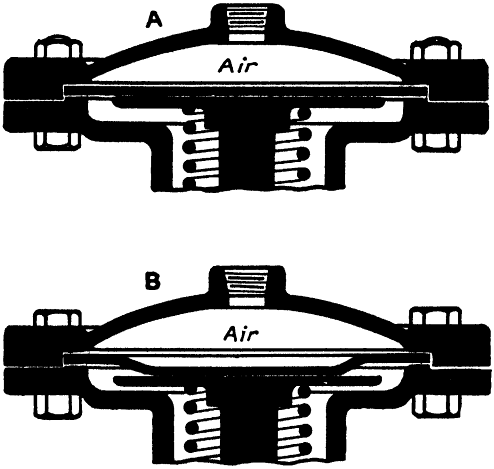


Fig. 5–8.—To produce maximum power throughout valve travel, diaphragm must remain in contact with moving head *A*, instead of bowing away, *B*.

Unless the diaphragm power is restored by applying a higher operating pressure, the valve remains incorrectly positioned.

The only power a diaphragm can contribute toward moving the valve is the difference between the motor power and the opposing spring or weight force. At each valve position, the two forces must balance; and, with each new motor-power movement, the diaphragm must receive sufficient additional power to overcome the loading force before movement occurs.

Assume three valves operating at (1) 100 psi with a 20 sq in. diaphragm, (2) 30 psi with a 63 sq in. diaphragm, and (3) 10 psi with a 132 sq in. diaphragm (Fig. 5–10*C*, *D*, and *E*.) Suppose we increase the pressure 1 psi on the 100-psi valve. Acting on 20 sq in., this produces a 20-lb increase in force opposing the springs. Since spring deflection is proportional to the applied load, the valve moves in proportion to the 20-lb increase. In like manner, 1 psi increase on the 30-psi valve is 63 lb; and 1 psi increase on the 10-psi unit

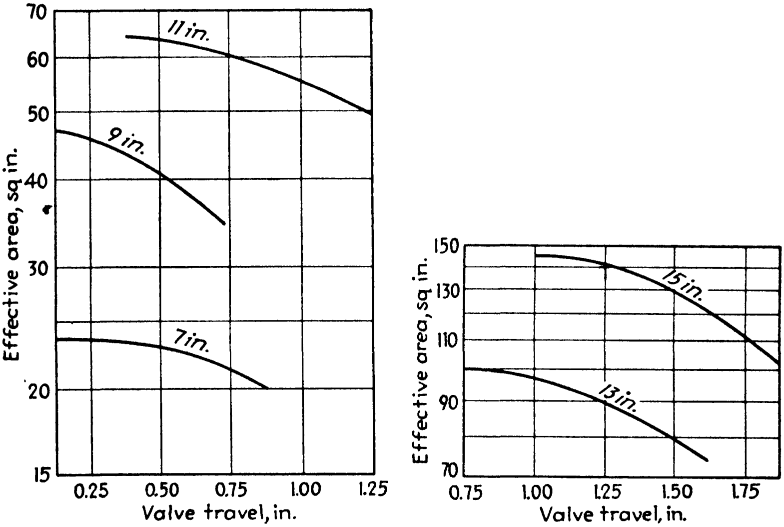


FIG. 5-9.—Change in effective diaphragm area at different valve positions.

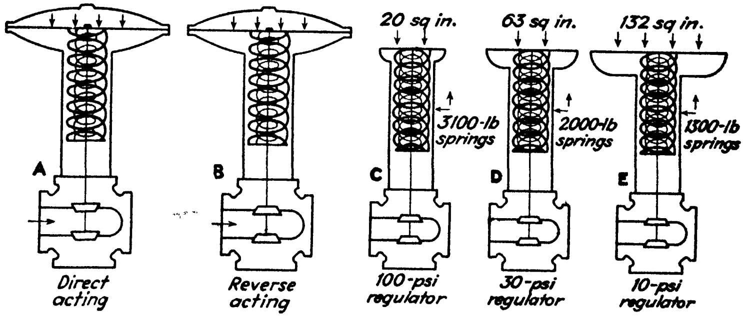


FIG. 5-10.—Diaphragm travel is proportional to the change in applied pressure.

is a 132-lb increase in force on the springs, which produces a corresponding proportional deflection and valve travel.

The spring load shown at each valve is the force required to compress the coils until they are "dead," or touching one another.

Therefore the 1 psi increase in pressure produces a deflection of 20/3,100 or 1/155, 1/32, and 1/10 of the total spring compressibility in each respective valve. If the total compressibility of each spring is 3 in., the change of 1 psi in operating pressure moves the valves 0.019, 0.094, and 0.30 in., respectively. Thus the 10-psi valve moves fifteen times farther than the 100 psi one. Considered from another angle, pressure on the 100 psi diaphragm must increase 15 psi before it moves as far as the 10 psi unit moved on the 1 psi change.

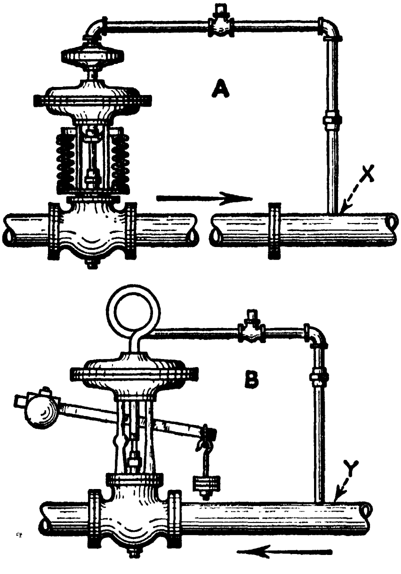


Fig. 5–11.—Self-operated valve holds reduced pressure at *A* and back pressure at *B*.

Going to higher pressures with self-operated valves, even greater pressure variation can be expected before the unit moves any appreciable distance. Higher pressure means lower sensitivity. In general, self-operated valves are satisfactory for service below 25 psi and for handling fairly constant volumes up to 150 psi. Figure 5–11*A* shows a self-operated direct-acting diaphragm valve maintaining reduced pressure in line *X*, and Fig. 5–11*B* shows a reverse-acting unit maintaining back pressure in line *Y*.

The most important characteristic of a diaphragm motor is its hysteresis, failure of the valve stem to assume the same position for the same motor pressure on upward and downward strokes. The distance between the increasing and decreasing pressure lines (Fig. 5–12*A*), in terms of percentage of full travel, is the hysteresis. It accumulates from spring and diaphragm hysteresis and from stem guide-bearing and stuffing-box friction. Excessive hysteresis produces a wide loop *B*; tight packing alone may account for 10 per cent. Both curves indicate the linear relation between the diaphragm pressure and the stem movement when the diaphragm remains in full contact with the plate throughout its stroke.

Assume that the diaphragm has an area of 100 sq in. and that the applied pressure is 7 psi. Thus the diaphragm exerts a 700-lb force to compress the spring and set the valve. Suppose the primary element now calls for a new valve position that should be obtained with 7¼ psi air pressure or an increased force of 25 lb. If the total valve friction equals this 25-lb force, no movement occurs until the controlled medium (pressure, temperature, flow, etc.) changes enough to admit more pressure to the diaphragm. The valve then jumps to a new position beyond that needed to supply the demand, which causes hunting, or cycling, of the control.

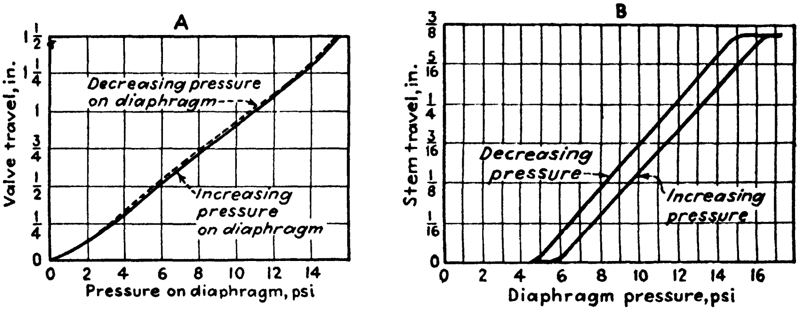


FIG. 5-12.—Stem travel for pressure change on diaphragm. *A* shows minimum hysteresis, *B* effects of excessive friction.

Valves have cast-iron or bronze bodies for lower temperatures and pressures, and alloy-steel ones for high temperatures and pressures. The trim may be cast iron, bronze, monel, stainless steel, or any suitable material that can be cast, machined, or ground to suitable shape. Where erosion is serious, stellited seats and disks find use; and hard stainless steel serves where both erosion and corrosion are bad.

Inner valves are roughly classified as (1) quick opening, (2) gradual opening, and (3) full throttling. The quick-opening valves may be of either the bevel-seated or the low-lift V-port type. The gradual-opening valves may be of either the high-lift V-port or the parabolic type—ratio plug and throttling.

Beveled-disk (Fig. 5–13) or rectangular-port valves serve on simple, high-sensitivity, small-time-lag applications usually for open-and-shut service. With stable load and constant line pressure, they operate satisfactorily between 10 and 90 per cent opening. However, they do not suit applications that require flow graduated to correspond to the load. Here the need is for *characterized* valves.

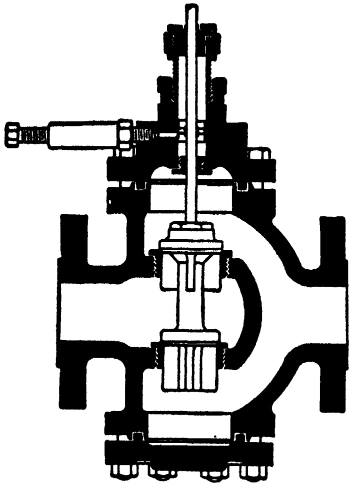


Fig. 5–13.—Quick-opening bevel-disk double-seat construction.

Assume a beveled-disk valve controlling a load that requires 10 units of flow (Fig. 5–14). If the valve lifts 1/10 unit, the flow changes 11 units, or 110 per cent of the original. Now suppose the valve supplies a flow of 40 units. Taking the same deviation that occurred before (1/10 unit change in lift), the flow changes only nine units, or 22 per cent of the original. A valve such as this requires a change in controller sensitivity to compensate for load changes if the same character of control is to be maintained for each load condition. The valve is therefore not suitable for applications that require flow through the valve to be graduated to correspond to the load.

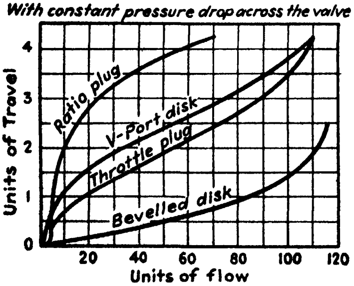


Fig. 5–14.—Flow-lift characteristics of different designs of inner valves.

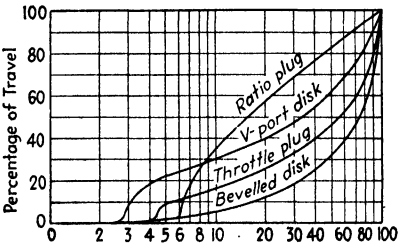


Fig. 5–15.—Flow-lift characteristics of valves plotted on semilogarithmic paper.

To give properly graduated flow in relation to load requires that the valve have a characteristic that gives a small change in flow for a given increment in travel at low lifts (corresponding to light load) and large changes for the same increments at high lifts, *i.e.*, equal increments of lift should give equal percentage increments in flow. This percentage characteristic is desirable for throttling control because it assures similar control action under light and heavy load conditions without changing the controller sensitivity. Such a characteristic appears as a straight line in Fig. 5–15, which shows various valve characteristics plotted on semilogarithmic paper.

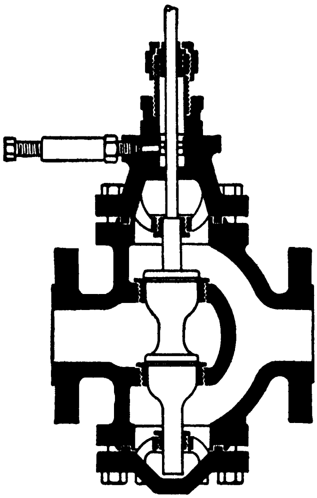


Fig. 5–16.—Ratio-plug double-seat valve with lower guide.

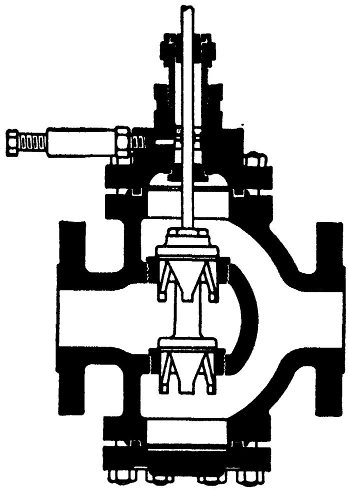


Fig. 5–17.—Double-seat V-port valve has high control range.

Characterized valves—ratio plug (Fig. 5–16), V-port (Fig. 5–17), and throttle plug (Fig. 5–18)—suit applications with changing loads because they most nearly approach the ideal characteristics of giving equal percentage increments in flow at equal increments of lift throughout the stroke. In the ratio-plug valve (Fig. 5–15), each uniform increment of lift results in a flow-rate change that is practically a constant percentage of the flow existing prior to each change in lift. It has the lowest capacity of the three throttling types but is recommended for difficult jobs involving large load changes. Having small clearance at low lifts, the ratio plug should not handle liquids carrying suspended solids or coking materials.

The V-port valve uses a beveled disk carrying a skirt fitted with several V-shaped ports. The V shape and arrangement gives constantly increasing port openings for each unit lift under constant pressure drop. It has the greatest controllable range of the characterized valves and a capacity as high as any.

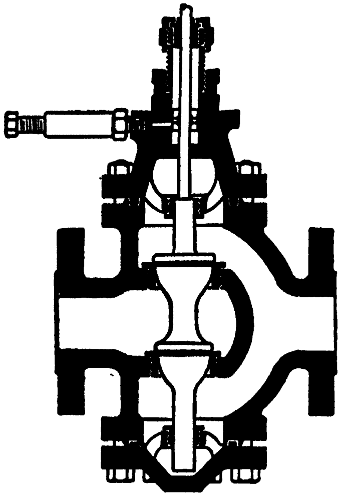


Fig. 5-18.—Throttle-plug design resists fouling at low lifts.

Although similar in construction, the throttle-plug valve gives larger flows at low lifts than the ratio plug at equivalent lifts. Developed for applications requiring better characteristics than beveled-disk valves, it also serves where resistance to wear is important. Having considerable port area at low lifts, the throttle plug does not *coke* or become fouled by suspended solids. It handles large pressure drops particularly well. When comparing equal percentage flow-lift characteristics, ratio plug ranks first, throttle plug second, and V port third.

Too much stress cannot be placed on the importance of sizing the valve for the service it is to render. When calculating size, determine accurately the pressure drop actually existing in the system. Never select a valve that gives minimum pressure drop, as is the practice when calculating pipe size. The purpose of a control valve is to create sufficient pressure drop to restrict the flow to that quantity called for by the controller. Naturally, the valve should not be too small, but it is equally important that it should not be too large. If it is oversize, it operates erratically, and the seat and disk are subjected to severe wire drawing because of the restricted opening.

Pressure losses in pipe lines vary as the square of the fluid velocity; and, in actual practice, the valve will not have constant pressure drop across it throughout full flow range, because some of the normal drop is absorbed by lines and fittings ahead of and beyond the valve. The nominal pressure drop across a valve can be completely absorbed by the line loss at peak flow, putting the valve and primary element completely out of control. A velocity of 5 ft per sec (for liquids) holds the line-pressure drop to a reasonable minimum at maximum flow. Figure 5–19 shows the loss in capacity and altered characteristic when a 1-in. ratio-plug valve (having good characteristics) is used with 30 ft of 1- and 1½-in. pipe. Beveled-disk, V-port, or throttle-plug valves show greater discrepancy, because of their greater relative capacities, size for size.

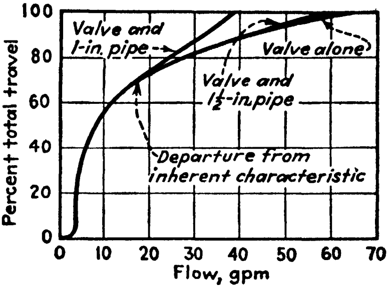


FIG. 5–19.—Flow lift of 1-in. ratio-plug valve with 30 ft of pipe.

When selecting a valve, consider whether it is to be single- or double-seat construction. Single-seat valves (Fig. 5–20) are impractical for throttling control if the line pressure fluctuates widely, unless the controller sensitivity is high or the valve small. Above 2 in., unbalanced thrust becomes a problem unless the line pressure is low. If the pressure drop across the valve is constant, the unbalanced load is constant and merely equivalent to an additional spring load opposing the diaphragm action. Usually both high and low side pressures vary with a change in valve opening. Turbulence and velocity effects also increase with flow.

As an illustration, assume a single-seated valve with a port area of 2 sq in. and an effective diaphragm area of 50 sq in. For a certain flow through the valve, the inlet pressure might be 200 psi and the discharge pressure 100 psi, giving a pressure drop of 100 psi, and an unbalanced load of 100 × 2 = 200 lb. The air pressure on the diaphragm required to just balance this load is 200 ÷ 50 = 4 psi. For the assumed opening, the total air pressure on the diaphragm might be 7 psi, of which 3 would be effective in positioning the valve, and the other 4 psi would be required to overcome unbalanced thrust on the valve disk.

Now suppose the load changes so that primary element calls for a greater valve opening. As the valve opens, the inlet pressure may drop and the discharge pressure increase. This decreases the differential pressure and produces exactly the same effect as though a lighter spring was suddenly placed on the valve. Air pressure already on the diaphragm opens the valve wider than it should, because of the sudden decrease in unbalanced thrust.

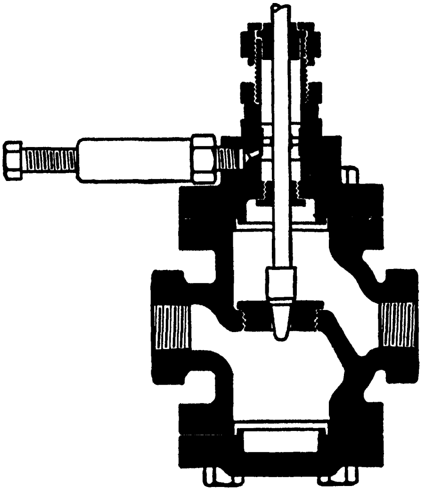


FIG. 5–20.—Unbalanced thrust is a problem in single-seat valves above 2 in.

Line pressure entering between the seats of a double-seat valve produces a semibalanced unit and eliminates the effects of varying line pressure to some extent, but they cannot be relied upon to give tight closure. Semibalancing permits the valve to operate at high pressure differentials because the diaphragm need not overcome full line pressure acting on the disk, as in a single-seat valve. Even in semibalanced valves, unbalanced forces are not always negligible. Such a valve may be decidedly unbalanced when handling flow under pressure. The pull chart (Fig. 5–21) shows unbalanced forces acting on a valve at different openings. The zero line, through the middle of the curve, represents balanced conditions. Starting at *X*, unbalanced forces try to lift the valve from its seat. After the valve lifts, pressure drop and dynamic forces change the lifting to a closing force with its highest value at approximately one-quarter lift. From here on, force changes to the opening direction, reaching full value at wide-open position.

The rate of flow through a regulating valve depends primarily on (1) the area of the valve opening and (2) the pressure drop across the valve. If the pressure drop is subject to wide variations, the feeding rate varies widely for a given valve opening. Pressure variations may be caused by pump characteristics, by faulty pump governors, or by friction loss in the line. A centrifugal pump always operates at the intersection between its head-capacity-characteristic curve and the system-head curve; altering the shape of either changes the pumps operation.

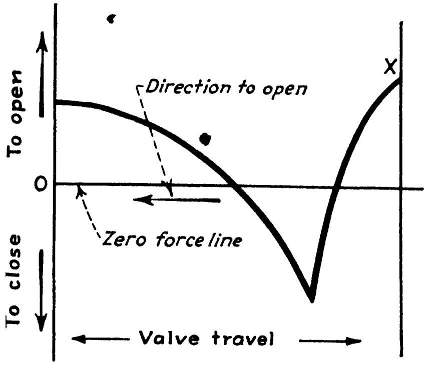


FIG. 5–21.—Thrust on a semibalanced valve throughout its stroke.

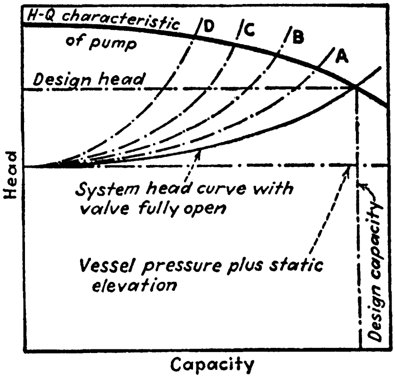


FIG. 5–22.—Curves show how valve position affects system head.

A regulating valve changes the system head (curves *A* to *D*, Fig. 5–22) as it moves toward the closed position. Pump head pressure then varies according to the intersection of the system head curve with the *H-Q* characteristic curve and influences the pressure differential across the regulating valve. Constant pressure drop can be maintained across the control valve by a differential-pressure control installed as shown in Fig. 5–23, if the latter responds accurately to pressure changes. With constant pressure drop across the control valve, the flow is proportional to the valve opening regardless of load changes.

The capacity of a valve depends on the pressure drop across it. Therefore, when laying out a piping system or when selecting a valve, check the available pressure drop carefully. For example, consider an installation where liquid is heated by a closed steam coil that is equipped with a condensate trap. The steam may be supplied at 100 psi and the trap may discharge to atmosphere; but this does not mean that available pressure drop across the valve is 100 psi. Some pressure will be lost in the supply piping, there will be additional drop through the coils, and finally pressure will be needed to force the condensate through the trap. Thus there will be considerably less than 100 psi pressure differential across the valve.

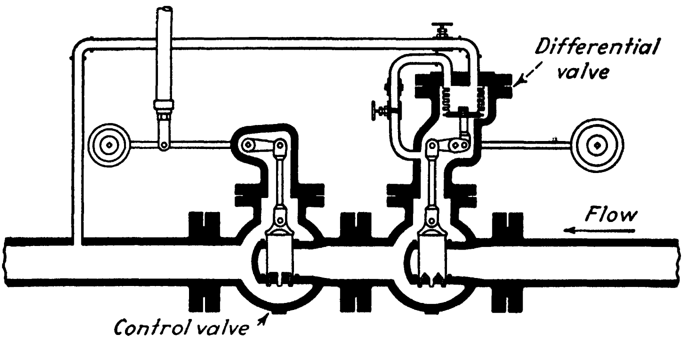


FIG. 5-23.—Differential valve installed ahead of main control valve. (*Courtesy of Northern Equipment Co.*)

Excessive valve hysteresis or unbalanced thrust may prevent the valve from assuming a position that corresponds exactly to primary-element output pressure. The controlled medium then deviates farther until the primary element moves and changes the controller pressure sufficiently to set the valve. A valve positioner (Fig. 5-24) supplied from a separate air source corrects for this and eliminates hunting. It assures that the valve takes a position having a definite relation to the primary-element position regardless of outside opposing forces such as packing friction and spring hysteresis.

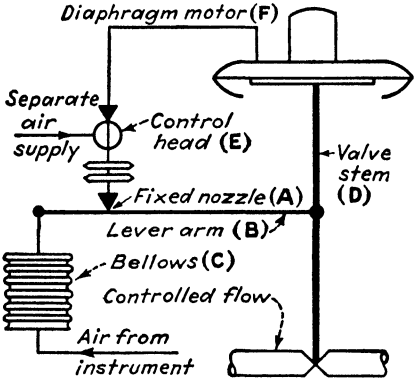


FIG. 5-24.—Schematic arrangement of a valve positioner.

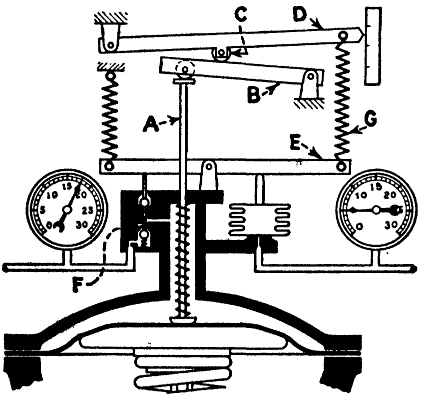


FIG. 5-25.—A positioner assures positive valve setting. (*Courtesy of C. J. Tagliabue Mfg. Co.*)

Here the loading pressure goes directly to the positioning device, mechanically connected to the valve stem, that applies higher pressure increments to the main diaphragm, based on the pressure of the primary element but modified by the actual valve position.

Any displacement of the valve from its correct position acts through the positioner to change the diaphragm pressure an exact amount to position the valve correctly.

In operation, the air discharge from a fixed nozzle *A* is controlled by the movement of a lever arm *B* held tangent to it. One end of the lever connects to bellows *C* supplied with air pressure from the primary element. Any movement of the bellows moves the lever. The other end of the lever connects to valve stem *D* and is moved by it.

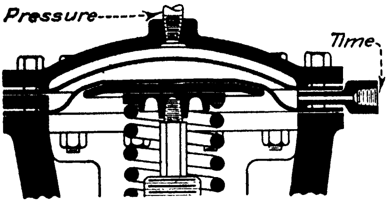


FIG. 5-26.—Second diaphragm is used with a time controller.

Changes in air pressure from the primary element expand or contract the bellows and move the lever toward or away from the nozzle. This throttling action on the nozzle operates a control head *E* that readjusts pressure on the diaphragm *F* until the nozzle and lever are again tangent. Because the air supply comes from an independent source, the pressure on the valve diaphragm can be greater or less than that acting on the bellows.

In the positioner (Fig. 5–25) stem *A*, adjustable on its upper end, is held against the valve diaphragm by a spring. One end of lever *B* is fulcrumed, and the other rests on stem *A*. The rider *C* is set in position for full valve travel and connects lever *B* with *D*, which is fulcrumed at one end. A spring connects levers *D* and *E*. Control lever *E* acts like a balance and operates reverse-acting pilot valve *F*. The spring force on the left end of lever *E* is of permanent magnitude. Two variable forces act on the right end; their magnitude changes inversely to each other, but the sum always equals the spring force on the left end. The variable force of the right-hand spring is governed by the valve-stem position and that of the bellows by the applied operating pressure.

The pilot-valve pressure and that applied to the diaphragm comes from an outside source. The operation is such that the bellows movement upsets the balance arm to operate the pilot until the main valve moves and changes spring force *G* enough to rebalance arm *E*. Another improvement for complicated control problems is the double-diaphragm top (Fig. 5–26). With this arrangement, pressure or temperature control applies operating fluid to the upper diaphragm, and a process time controller (Fig. 5–27) operates a valve to apply air pressure to the auxiliary chamber and shut off the process.

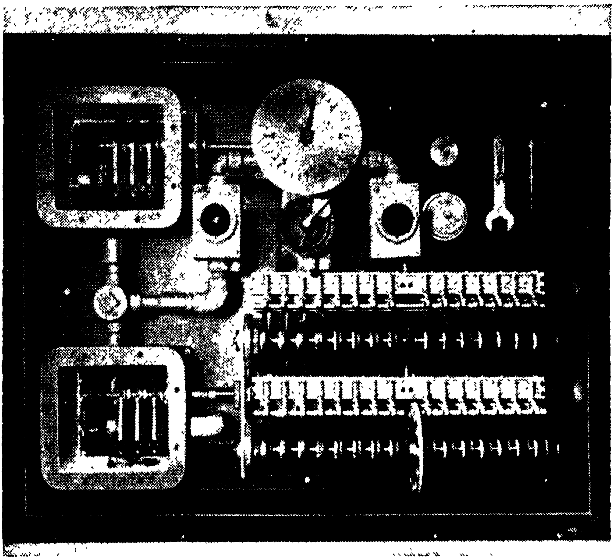


FIG. 5–27.—Time controller with synchronous-motor clock and series of air valves (*Courtesy of Automatic Temperature Control Co., Inc.*)

Wide flexibility when controlling large as well as small flows is a requirement of good valves. This range of control is partly determined by flow design of valve passages and partly by the traveling characteristics of the valve plug. High turbulence and eddy currents, caused by poor design, cut down valve capacity at high flows and decrease its range before full capacity is reached.

Likewise, throttling characteristics at very small flows will be determined by lift-flow relationship of the inner valve. The valve must be able to throttle the smallest flow required without overtraveling and starting a surge. It should not, when first opened, permit too great a flow, followed by a quick closing, and then a surging cycle of opening and closing. The minimum flow that can be handled by any valve is limited. Below this minimum, good control will be sacrificed and surging and wire drawing will probably result. The actual range will, of course, depend on the size, type, and design of valve. If the control range is too great for a particular valve, it is best to use two valves or a small one to handle low and normal flow and a manual by-pass for maximum loads that occur rarely.

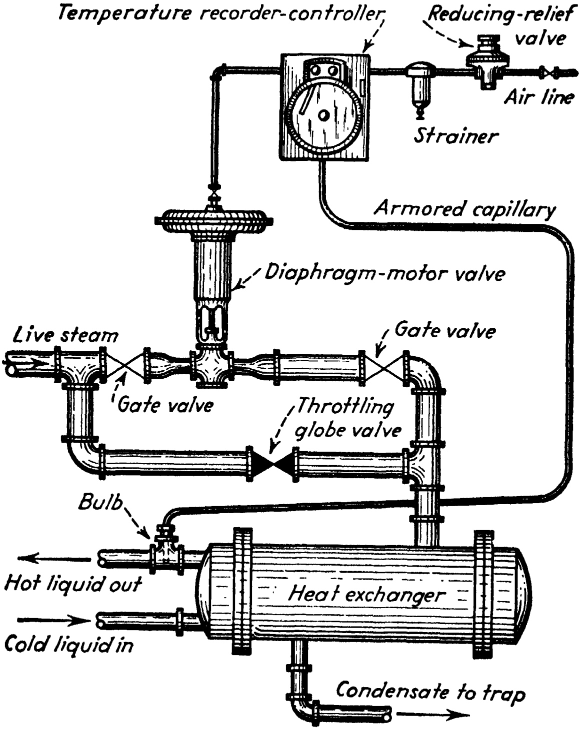


FIG. 5-28.—Install by-pass line and throttling valve to permit inspecting the control valve.

On severe requirements, where control must be fully automatic at all times, install two valves in parallel. Set the valve springs so that the second valve will not start lifting off its seat until the first is wide open. This arrangement provides accurate control at low flows and gives large capacity for maximum flows. The liquid capacity of a diaphragm valve varies as the square root of the pressure drop across the valve. For noncritical steam, air, or gas flow, capacity varies as the square root of the product of pressure differential and low-side pressure; and, for critical flow, capacity varies directly as high-side pressure. Liquid and gas capacities must be corrected for specific gravities or temperatures other than standard; steam flow must be corrected for quality of superheat. A liquid acts more like a solid because it does not expand or compress like a gas. A successful liquid-control valve must therefore have a large port area, or the flow will be greately restricted on heavy demands.

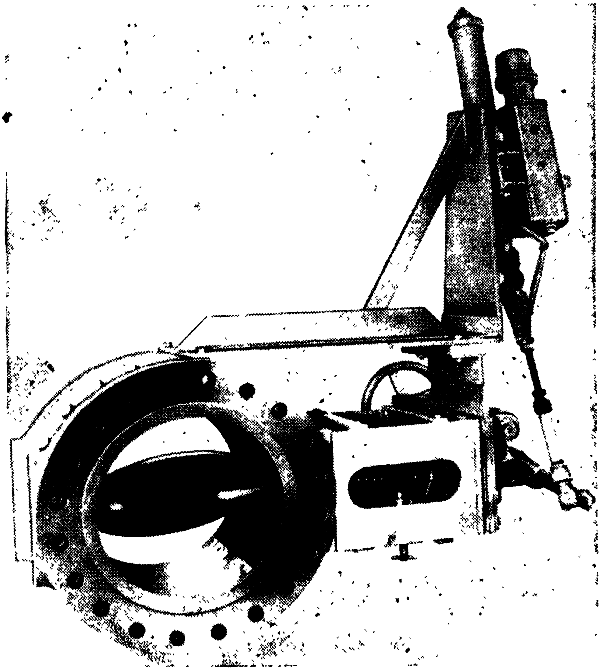


FIG. 5-29.—Butterfly valves find wide use with all kinds of fluids. (*Courtesy of R-S Products Corp.*)

Most manufacturers provide capacity curves or charts for their valves, which gives maximum flow for specified conditions. The valve should be selected so that it will normally be not more than 60 to 75 per cent open. The capacity of a double-ported valve is greater than that of a single-seat valve of the same nominal pipe size, but it will not have twice the capacity, other things being equal.

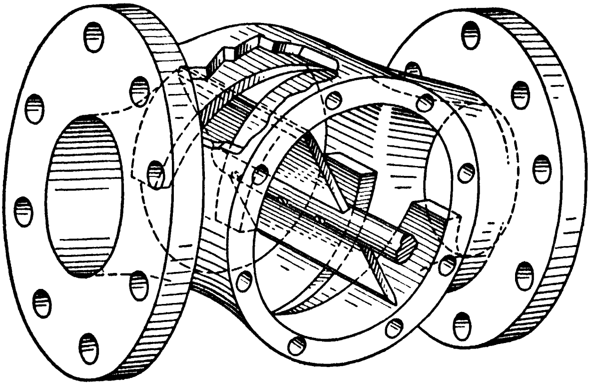


FIG. 5-30.—Butterfly valve with V ports. (*Courtesy of Hagan Corp.*)

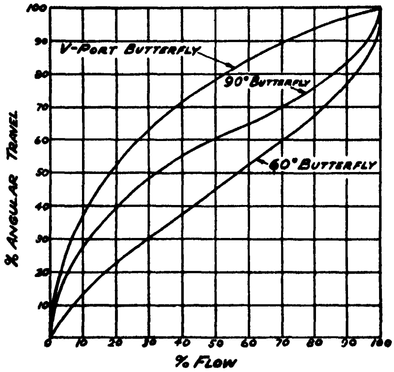


FIG. 5-31.—Curves for different butterfly valves.

Install the control valve in a straight run of pipe with a shutoff valve on either side and a by-pass line around the main valve so that it can be removed for repairs (Fig. 5–28). The by-pass also permits operating the system on hand control when starting up or shutting down. When reducing from pipe size to the valve and up to pipe size again, use bell reducers or streamlined fittings to reduce pressure drop to a minimum. Have the valve accessible for inspection; do not install it in crowded corners. Mount it vertically for best operation, because this puts less wear on moving parts.

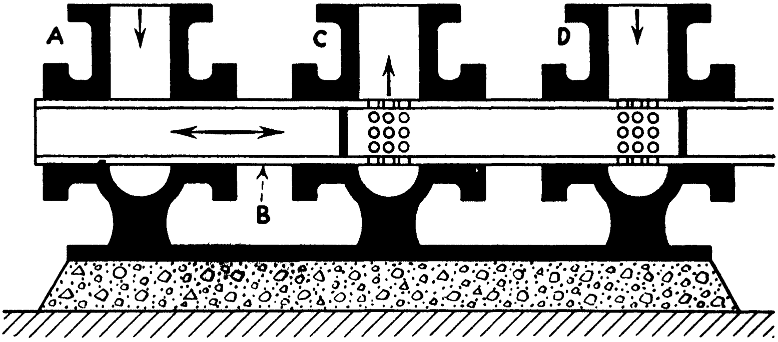


FIG. 5–32.—Sliding-tube valve delivers liquid from either of two sources to a common outlet.

When ordering or specifying a control valve, send the manufacturer this information:

1. Nature of fluid handled; is it corrosive or erosive?
2. Variations in flow (minimum, normal, maximum).
3. Allowable pressure drop across valve (minimum, normal, maximum).
4. Initial, or high-side, pressure and maximum pressure.
5. Flowing temperature and maximum temperature.
6. Gravity—API, Baumé, or specific (referred to air or water).
7. Can valve be operated with a manual by-pass?
8. Should valve be direct- or reverse-acting?
9. Can valve be double-port or must it operate on dead-end service?
10. Is service throttling or on-and-off?

The butterfly valve (Fig. 5–29) built in large sizes finds wide use in handling fluids of all kinds. The modern unit is precision-machined and has a beveled vane set at an angle with the valve body so that it tends to close under pressure drop. Figure 5–30 shows a V-port butterfly valve, and Fig. 5–31 shows the relation between angular movement and flow.

The sliding-tube valve (Fig. 5–32) automatically delivers water from either *A* or *D* to a common outlet *C*. The choice of inlets is automatically made according to hydrostatic pressure prevailing at the normally used inlet. When this pressure falls, shift-over occurs through a hydraulic cylinder and piston attached to sliding tube *B*. Two groups of holes in the tube walls, inside the dead-ended area, are so spaced that they line up with main outlet *C* and either inlet *A* or *D*.

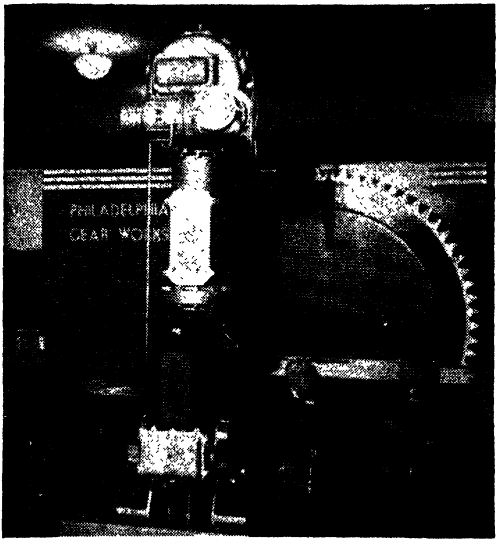


FIG. 5–33.—Electric-motor-operated valve. (*Courtesy of Philadelphia Gear Works.*)

Large electric-motor operators (Fig. 5–33) can close a 24-in. valve in 3 sec to a predetermined seat pressure without danger of jamming.

CHAPTER VI

SELF-OPERATED PRESSURE VALVES

The self-operated valve is entirely satisfactory for many services but has certain limitations. With single-seat valves, the regulated pressure varies slightly with variations in primary pressure, but this deviation is minimized in double-seat valves. The controlled pressure also drops slightly from minimum to full flow because the regulated pressure must change before the valve moves.

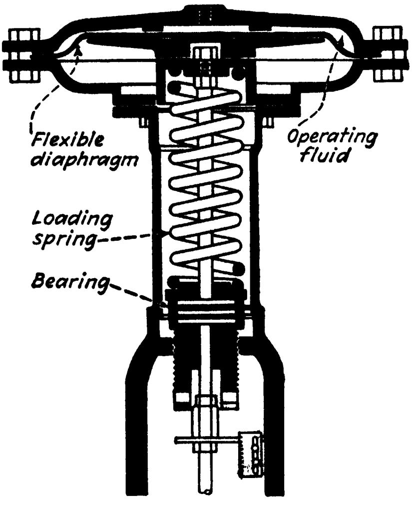


Fig. 6-1.—Flexible diaphragm is made of natural or synthetic rubber with fabric insert. (*Courtesy of Fisher Governor Co.*)

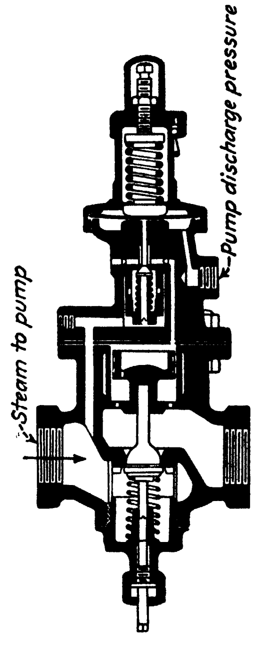


Fig. 6-2.—Metal diaphragm acted on by pump discharge pressure controls admission of steam to top of piston. (*Courtesy of Foster Engineering Co.*)

Here we shall consider those which reduce and relieve pressure, control back pressure, and govern pumps and fans. In general, these units are operated by a diaphragm (Figs. 6–1 and 6–2), piston (Fig. 6–3), or solenoid (Fig. 6–4). Operating motors are opposed by weight, spring, or fluid pressure. Although all these motors can be connected for on-off (fully open, fully closed) operation, the motor of the diaphragm- or piston-operated valve will hold it at any intermediate position between open and closed. This is not true of the solenoid-operated unit, however. If it is open when the coil is energized, it must be closed when the coil is deenergized. The only alternative is to have a separate solenoid for each intermediate position (Fig. 6–5).

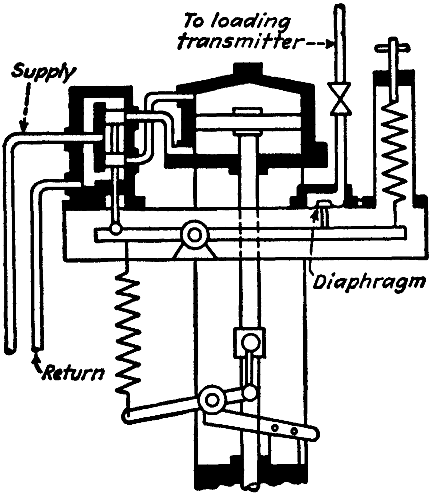


Fig. 6–3.—Piston motor piped to receive operating fluid from an outside source.

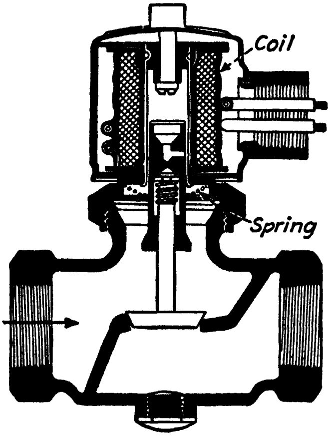


Fig. 6–4.—Single solenoid valves have only two positions, open and closed.

A diaphragm and piston are often combined in one unit. The valve in Fig. 6–6 is operated by air pressure from a pneumatic controller acting on a rubber-diaphragm motor (not shown). The first downward movement of the valve stem opens the pilot valve *A*, which admits pressure through passage *B* to the top side of the piston. Thus pressure on top of the piston balances that on the

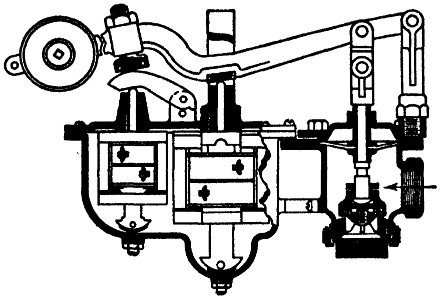


FIG. 6–5.—Two solenoids give the valve three positions.

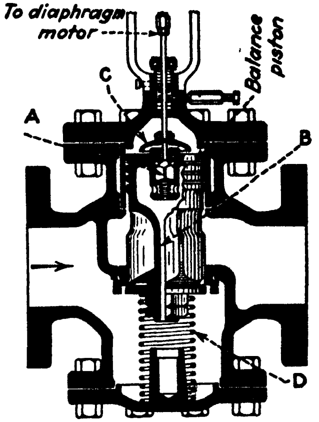


FIG. 6–6.—Pressure to balance the valve disk is admitted on initial stem movement. (*Courtesy of Leslie Co.*)

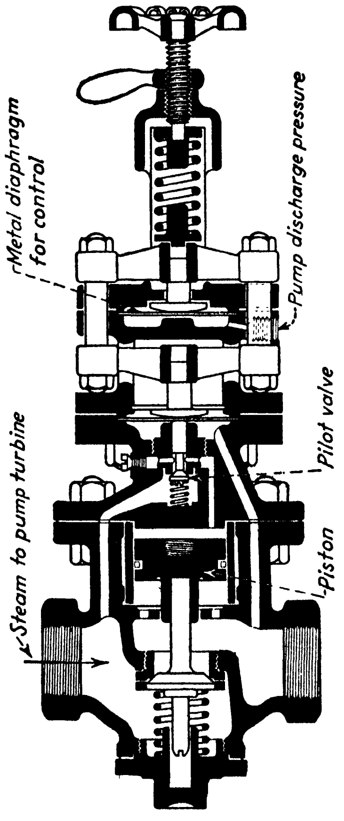


FIG. 6–7.—Piston-operated valve controlled by metal-diaphragm

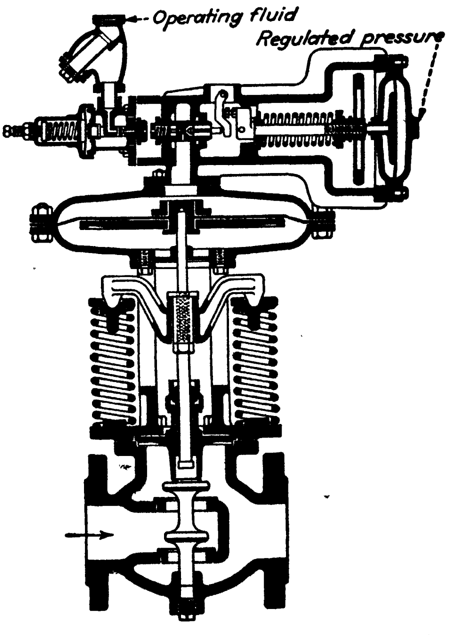


FIG. 6–8.—Main valve carries external pilot and small operating-pressure reducer. (*Courtesy*

underside of the valve disk. Continued downward movement of the valve stem pushes strong back *C* against the piston to move the valve against spring force *D*. This balancing of the piston eliminates unbalanced forces in single-seat construction, and the single seat provides tight shutoff on dead-end service. In Fig. 6–7, a piston controlled by a metal diaphragm operates the main valve. The self-operated regulator can be controlled by either an internal (Fig. 6–7) or an external (Fig. 6–8) pilot valve. The pressure being controlled acts on the pilot-valve diaphragm, and a slight pressure variation is ample to actuate the pilot valve and permit the operating fluid to exert full force on the main diaphragm or piston.

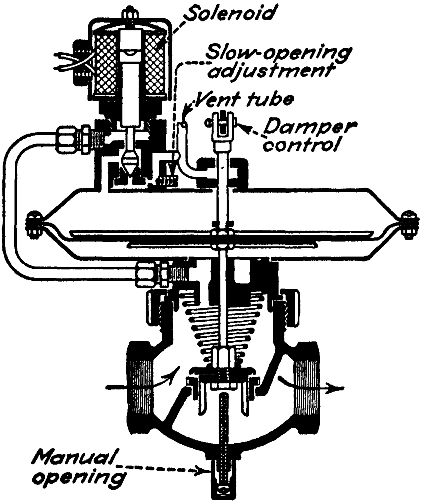


FIG. 6–9.—Solenoid pilot admits pressure to or bleeds it from the upper diaphragm chamber. (*Courtesy of General Controls Co.*)

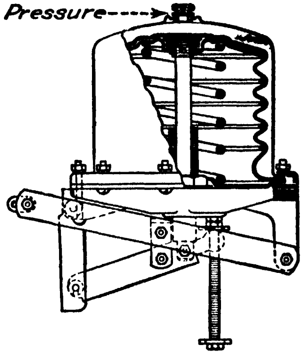


FIG. 6–10.—Air-pressure-operated bellows moves ventilating dampers.

The unit in Fig. 6–7, fitted with monel diaphragms, is used as a pump governor. In operation, the handwheel is adjusted to compress the spring and move the external yoke downward until the pilot valve opens. This admits steam from the main-valve inlet to the top of the piston, opening the main valve and admitting steam to the pump. Reduced steam pressure acts on the lower diaphragm. The pump-discharge pressure is piped back to the upper-diaphragm chamber. These pressures react on the diaphragms to control the pilot valve, which in turn positions the main valve to maintain a constant pump-discharge pressure. Metal diaphragms have great strength and can operate in contact with steam (pilot lines should drain condensate away from the diaphragm) but lack sufficient flexibility to allow much valve movement unless the diameter is large.

The solenoid-operated valve in Fig. 6–4 is constructed to be open when the coil is energized. The solenoid pilot (Fig. 6–9) allows pressures above and below the diaphragm to equalize when the coil is deenergized. When the plunger and pilot stem move up, the upper diaphragm chamber is isolated from the lower, and the pressure in the former bleeds out an adjustable vent. Line pressure then opens the main valve and the air damper. The valve always closes quickly, but the opening time can be changed by turning the bleed-vent adjusting screw. Although air dampers are often operated by the fuel-valve motor, a separate bellows device (Fig. 6–10) can be incorporated in a more complex system. Because of its compact design, it may be mounted inside duct work.

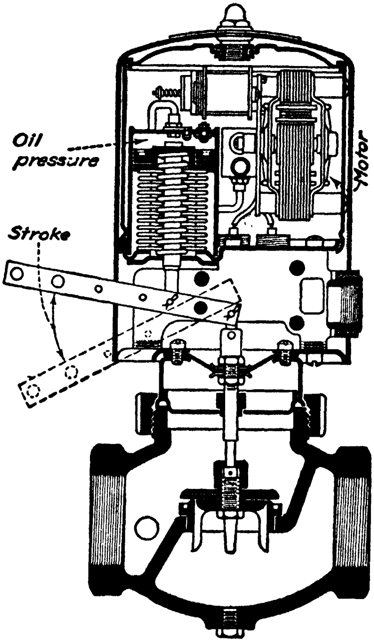


FIG. 6–11.—Motor-driven pump supplies oil to operate bellows. (*Courtesy of General Controls Co.*)

The hydraulic-motor valve (Fig. 6–11) is designed to control gas feed and furnace dampers. A motor-driven pump supplies oil under pressure to open the valve. Oil pressure is maintained by a magnetically actuated release as long as the valve mechanism remains energized. Should the control fail or call for a fuel shutoff, a magnetic release vents the oil pressure, and a spring closes the valve. The unit in Fig. 6–12 also controls gas feed to a furnace, the electric relay receiving its operating current from a thermocouple in the pilot-burner flame. Failure of this flame automatically shuts off gas flow.

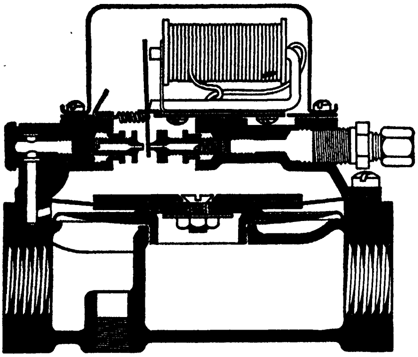


Fig. 6–12.—Electric relay receives current from thermocouple in pilot flame. (*Courtesy of General Controls Co.*)

A device of considerable importance to boiler operation is the nonreturn stop-and-check valve (Fig. 6–13). After it has once been opened by manual screwing up of the handwheel, the valve operates automatically as a nonreturn device to cut off reverse steam flow if a boiler tube ruptures. Some valves have a second seat to prevent steam flow if a steam header bursts.

The unit in Fig. 6–14 fills the requirement for services that must be shut off quickly if a line breaks or a vessel ruptures. When large vessels are not strengthened against external pressure, the vacuum breaker (Fig. 6–15*A*) will open if the internal pressure drops below the danger point. The air-inlet valve (Fig. 6–15*B*) protects large pipe lines against vacuum when used on gravity systems.

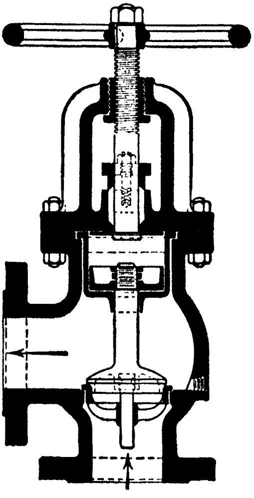


Fig. 6–13.—After being opened manually, this valve operates as a nonreturn device. (*Courtesy of Foster Engineering Co.*)

To close itself, one turbine-bleeder nonreturn valve (Fig. 6–16) uses an oil cylinder and springs in addition to reverse flow. Hydraulic pressure from the turbine-oiling system forces the piston to the top of its stroke and compresses the springs. Then the valve spindle is free to slide back and forth along the valve stem—opening to permit steam

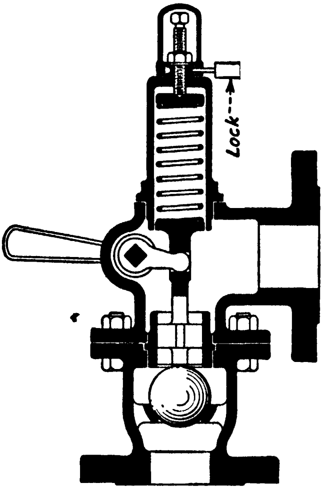


FIG. 6-14.—Ball valve closes quickly on downstream line failure.

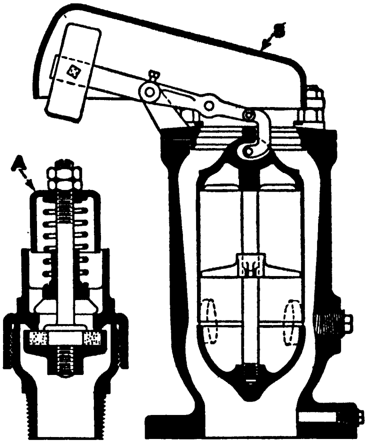


FIG. 6-15.—Valve *A* protects vessels against vacuum and *B* breaks the vacuum in gravity-flow pipe lines.

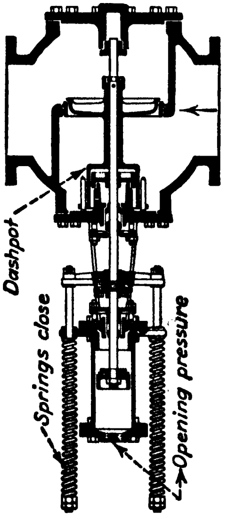


FIG. 6-16.—When valve is open, disk is free to slide on its spindle. (*Courtesy of Davis Regulator Co.*)

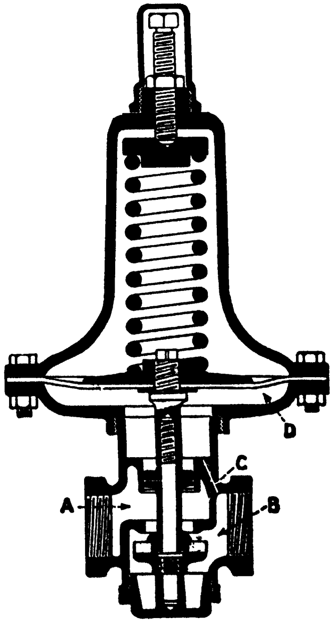


FIG. 6-17.—Self-operated pressure-reducing valve uses soft diaphragm.

flow from the turbine and closing on reverse flow. Should the emergency governor act, a trigger that trips the main throttle valve releases oil pressure in the cylinder, and spring compression pulls the bleeder valve closed. A fiber ring, on top of the piston, seats against the upper cylinder seat to prevent oil leakage when the piston is at the top of its stroke. Spring compression should not prevent this seating. A closed dashpot in the valve body cushions the valve-spindle movement.

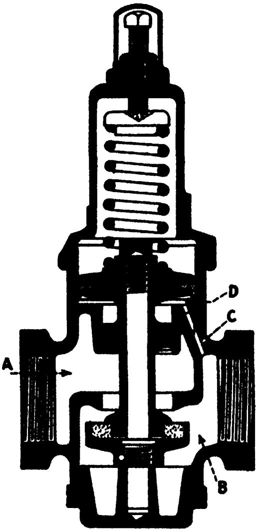


Fig. 6–18.—Piston operates this pressure-reducing valve. (*Courtesy of Foster Engineering Co.*)

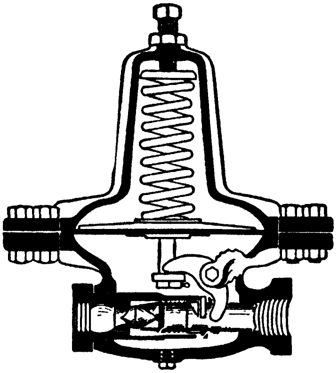


Fig. 6–19.—Inner valve has streamlined design. (*Courtesy of A. W. Cash Co.*)

Self-operated pressure-reducing valves all operate like the diaphragm valve (Fig. 6–17) and the piston valve (Fig. 6–18). Spring force and incoming pressure at *A* tend to hold the main valve open until discharge *B* fills with fluid at the required reduced pressure. Flow upward through passage *C* to chamber *D* forces the valve toward the closed position until its opening just maintains a reduced pressure corresponding to the spring force. Figure 6–19 uses a streamlined inner valve. The unit in Fig. 6–20 reduces the pressure in two steps with two independent diaphragms. Full primary pressure enters the first stage and is reduced to an intermediate pressure set by the manufacturer. The second stage, manually set by an adjusting lever, reduces the intermediate pressure to the required value. This device regulates gas pressure for cutting and welding torches.

In pressure-loaded regulators, fluid under pressure serves in place of a spring to load the diaphragm. The unit in Fig. 6–21 is employed for primary pressures up to 4,000 psi and delivers pressures up to 1,000 psi. Turning the pilot tension screw clockwise into the housing loads the diaphragm to the correct pressure as shown by an attached pressure gage. The tension screw is then released. To reduce the diaphragm-loading and delivery pressure, the dome relief valve is opened until the desired value is reached.

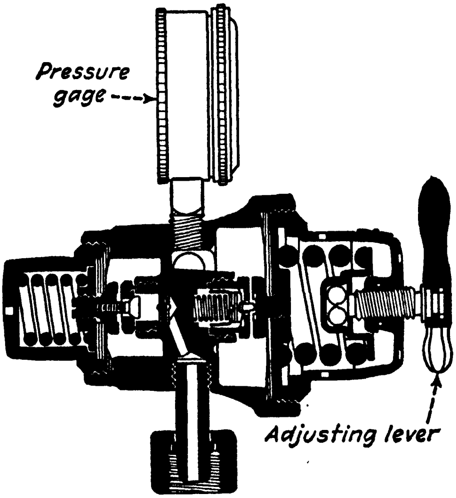


FIG. 6–20.—Regulator reduces pressure in two steps.

The unit in Fig. 6–22 has its dome charged with air or other gas. Pressure then passes through a restricting orifice in the separating plate to the operating diaphragm. The dome pressure is always slightly higher than that at the valve outlet to overcome the force of the two springs acting to close the valve. If the diaphragm fails, pressures on both sides of it equalize, and the springs, together with incoming pressure, force the valve to its seat. The valve is separate from the diaphragm rod to eliminate binding and reduce stem friction. A pressure tight sleeve around the operating stem eliminates the need of packing. The contours of the upper and lower separating plates fit those of the diaphragm and prevent damage to it by high one-sided pressure.

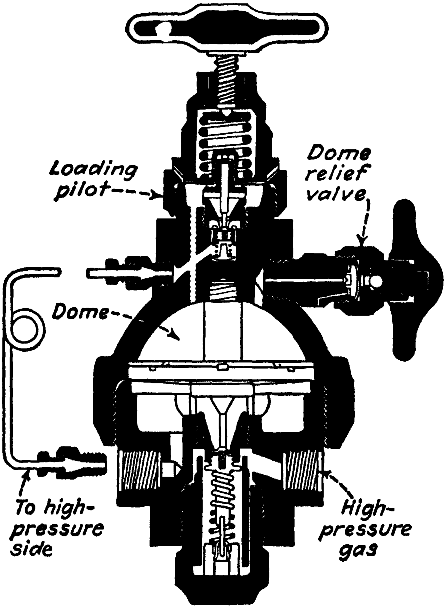


FIG. 6-21.—Dome-loaded unit for extremely high pressures. (*Courtesy of Victor Equipment Co.*)

The air dome may be charged periodically from an ordinary hand air pump, a compressed-air line, high-pressure carbon dioxide, or air bottles. With gas-loaded domes, ambient temperature variations affect the loading pressure and therefore that of the regulated fluid. The automatic dome-loader controller (Fig. 6-23) takes care of ambient temperature variations. It operates from a pressure supply up to 3,000 psi and maintains correct pressure inside the dome. When the temperature affects the dome air, the changed pressure reacts on the diaphragm in the controller to admit more air to the dome or bleed some out of the relief valve. Normally both controller valves remain closed.

The valve in Fig. 6–24 consists of a flexible tube surrounding two metal sleeves containing rectangular ports. The tube expands to open and is closed by hydraulic pressure applied between it and the outer casing. The valve serves for fluids containing sand, cuttings, or other fine solids.

Altitude valves are pressure-regulating units that maintain a set level (pressure) and prevent overflowing of tanks. For the unit in Fig. 6–25, a separate pilot assembly can be located wherever convenient. The height of liquid (pressure) to be maintained is set by adjusting the pilot-valve spring. The main valve is double-acting and suited for single-pipe systems. (The liquid is pumped to the tank and then backflows through the valve to the distribution system when the inlet pressure falls below the tank pressure.) For a double-pipe system, check valve *A* is removed and the openings plugged.

The valve, which is normally closed, opens by pressure under the disk. When the desired tank head is reached, pressure on the diaphragm of the pilot valve opens it and admits pressure to the diaphragm of the three-way valve. The latter closes its drain port and admits pressure to the top of main-valve piston *B*, which closes the main valve.

When the supply pressure falls below the tank static head, pressure on the pilot-valve diaphragm bleeds off through check valve *A*. A spring closes the pilot valve, which cuts off pressure to the three-way valve diaphragm. Diaphragm pressure then bleeds off through an adjustable orifice; the three-way valve

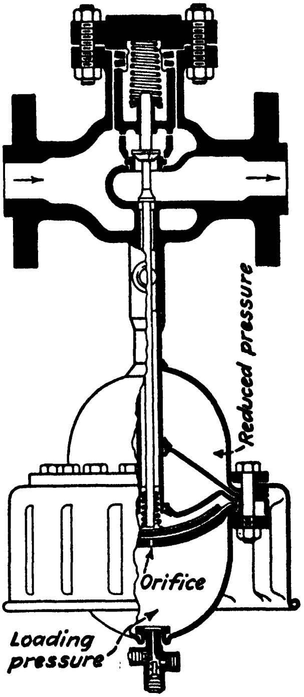


FIG. 6–22.—Dome-loaded unit uses pressuretight sleeve instead of packing gland. (*Courtesy of Grove Regulator Co.*)

closes its supply port and relieves the pressure on the main-valve piston. Static pressure from the tank flows up port *C* under the piston and opens the valve so that liquid can flow back into the supply line.

The valve in Fig. 6–26 is suitable for either single- or double-pipe systems. Fluid from the pump enters at the left. If there is no pressure at the outlet side or in chamber *A*, pilot valve *B* is open, the main valve is lifted from its seat by inlet pressure, and flow is established. Liquid from the inlet passes through adjustable cock *C* to diaphragm chamber *A*. Cock *C* is adjusted so that its port area is less than that of pilot valve *B*. The final reduced pressure is transmitted through pipe *D* to the underside of control diaphragm *E*, which is opposed by adjustable spring *F*. As the desired reduced pressure reaches the value for which spring *F* is set, pilot valve *B* starts to close, and its port area becomes less than that of cock *C*. Inlet pressure therefore builds up in chamber *A* and closes the valve. Regulation is maintained by the varying ratio of pilot-seat area to cock area. On backflow through the valve, the pressure on the tank side is highest, and the pressure in chamber *G* holds the valve open.

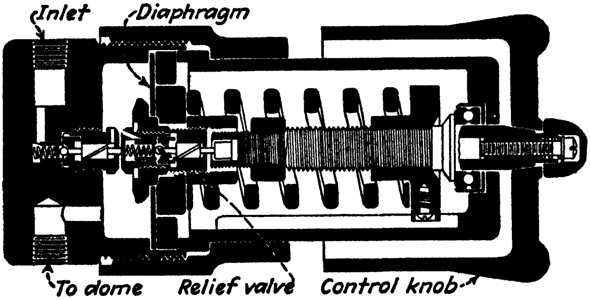


FIG. 6–23.—Automatic regulator to correct dome-loading pressure according to ambient temperature changes. (*Courtesy of Grove Regulator Co.*)

Regulating valves are made single- and double-seat or balanced

—two ports and an upper and lower disk of about the same area. This feature makes the valve easy to operate because flow enters between the two disks so that pressure thrust is practically nil. The inner valve is designed for quick or gradual opening (V port or parabolic). Valves that open gradually find greatest application where close throttling is desired or the rate of flow varies greatly.

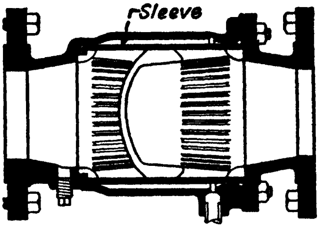


Fig. 6-24.—Hydraulic fluid forces flexible sleeve against ports to shut off flow. (*Courtesy of Grove Regulator Co.*)

If the double-seat valve uses metal-to-metal seats, it has one drawback caused by the difference in expansion between the valve plugs and the body. If the seats are ground at one temperature, the valve may leak at some other temperature. For dead-end service, especially where the temperature may vary, it is best to use a single-seat valve.

Always install a regulating valve with the inlet and outlet as marked on the body. Connection to the diaphragm should be made approximately 10 ft downstream from the valve or where turbulence is a minimum, but if necessary it can be made within 3 ft of the valve. Protect rubber diaphragms with a water seal when used on steam (Fig. 6-27). Best results are obtained on metal diaphragms if the condensate drains away from the diaphragm. Rubber diaphragms introduce other problems. If installed inverted on oil or chemical lines, the diaphragm is exposed to leakage through the stem packing. Heat from adjacent steam lines may cause early deterioration of the rubber. When making the diaphragm-control connection to the main line (point *A* in Fig. 6-27), drill and tap the line or weld on the correct-sized fitting. Never bush down a tee fitting to match the pressure tap. Numerous bushings screwed together in a tee form a pocket inside the main line. This pocket affects the flow thus causing a false pressure at that particular point.

A well-chosen regulating valve will not necessarily be of the same size as the connecting pipe. When this occurs, use bell or venturi reducers rather than pipe bushings (Fig. 6–28) to make connections. Install a strainer ahead of the valve to catch foreign material and protect the seats. To facilitate inspection, connect a throttling-globe valve as a by-pass in parallel with the regulating valve (Fig. 6–29). Impingement noises can be minimized by installing the valve in a straight run of pipe. Do not under any circumstances have an elbow or a pipe bend immediately downstream from the valve.

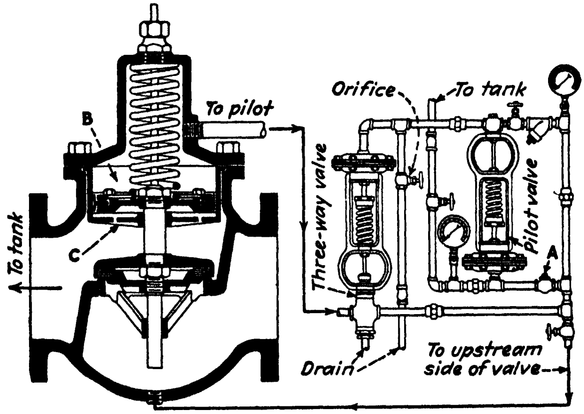


FIG. 6–25.—An altitude valve with external control pilots. (*Courtesy of Davis Regulator Co.*)

Safety and Relief Valves.—Safety and relief valves provide an emergency outlet for confined energy that is approaching the danger point. When specifying these devices, state the pressure at which they are to lift, the relieving capacity, and the temperature of the fluid handled. Temperature determines what kind of metal the device is made of and affects the specific volume of the gas and hence the valve capacity in pounds per hour. Valves rated on saturated steam have a lower capacity on superheated steam in pounds per hour.

Although safety valves are attached to pressure vessels to relieve dangerous overpressures, it is surprising how frequently they are selected haphazardly and ignored after installation. Even some design engineers are inclined to treat them rather lightly. In fact, we often see a drawing for installation of a pressure vessel that contains the note "Provide one ½-in. safety valve set at 100 psi," without stipulating the necessary relieving capacity.

The table of safety-valve capacity tests, published by the National Board of Boiler and Pressure Vessel Inspectors, indicates that safety valves of various makes and models, when set to open at a certain pressure, discharge widely different quantities of steam (Fig. 6–30). Keep in mind that they are 100-psi valves and would meet the requirements of the above note on a drawing. Thus, merely specifying the structural size does not suffice, and the discharge curves in Fig. 6–30 show why. Another important point to consider is the kind of fluid to be discharged, because the type affects the valve capacity (Fig. 6–31).

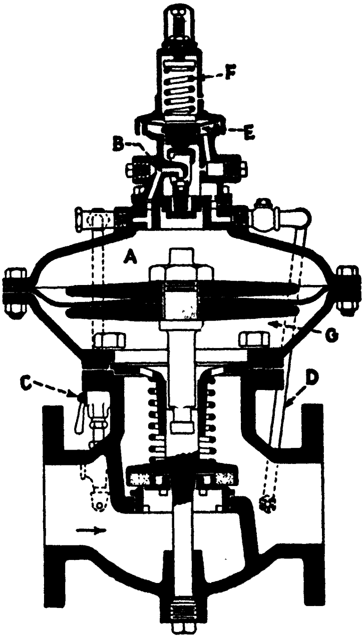


FIG. 6–26.—This altitude valve is a self-contained unit. (*Courtesy of Foster Engineering Co.*)

When used on steam, air, or gas service, a safety valve should be set to open at a pressure not in excess of the designed working pressure of the equipment it protects, or not more than 20 per cent above normal working pressure, whichever is less. Thus, a boiler designed for 100 psi, but now used only for heating a building that requires a maximum working pressure of 15 psi, should have a safety valve set at 18 psi, not 100.

According to the ASME Boiler Code, always install safety valves without stop valves on the inlet or discharge piping, because the

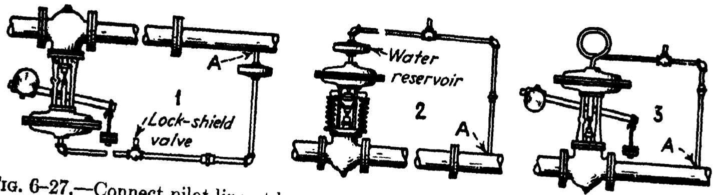


Fig. 6-27.—Connect pilot line at least 10 ft downstream, and use water reservoirs to prevent steam from reaching rubber diaphragms.

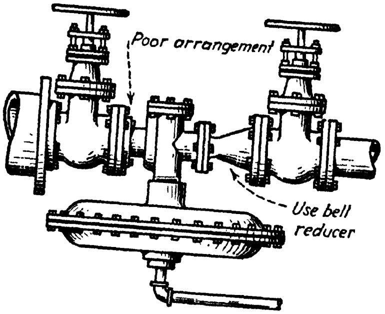


Fig. 6-28.—Use streamlined reducing fittings when attaching regulating valve.

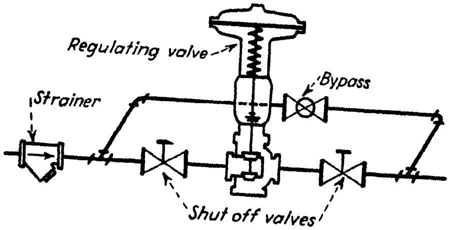


Fig. 6-29.—Protect the valve with a strainer and install a by-pass line.

accidental or intentional closing of such valves renders a safety valve useless. This code requirement is as it should be. However, in the chemical industry, a difficult situation frequently exists. In some processes it is impossible or impracticable to shut down operations, yet gummy residues precipitate inside the piping, including the safety valve, which may thus be rendered inoperative after a short period of time.

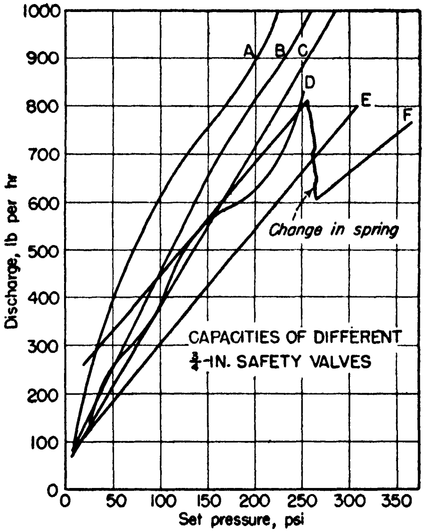


Fig. 6–30.—Discharge capacity of different types of valves varies over a wide range. (*Courtesy of Power.*)

In such plants, it is quite common to see stop valves placed between the pressure vessel and the safety valve. Chemical engineers recommend shutting the stop valve for ½ hr once a month, during which time the operator maintains extra vigilance at his pressure gages, to permit a mechanic to inspect, clean, and test the safety device. They believe this practice far safer than to prohibit installation of the stop valve and continue to operate for weeks or months with a safety valve so gummed up that it probably will not work. The stop valves, when installed, are always sealed open, and routine inspections are made to ascertain that the seals remain intact.

Safety valves are made in two general types: (1) relief valves used on liquids and (2) pop or pop-safety valves used on vapors or gas. The second are often called simply safety valves.

The relief valve (Fig. 6–32*A*) is a spring-loaded device designed and adjusted to open at predetermined hydraulic pressure acting on the underside of the disk. This pressure is resisted by the spring.

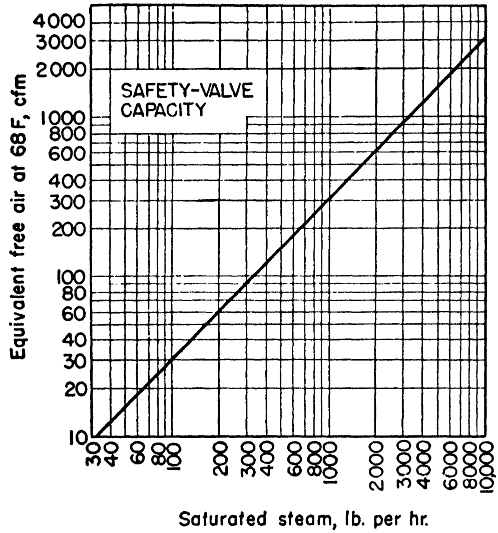


FIG. 6–31.—When transferring a safety valve from steam to air, or vice versa, check the relative discharge capacity. (*Courtesy of Power.*)

Only a relatively small quantity of liquid ordinarily passes through the relief valve before the pressure in the vessel falls and the valve reseats. Closing takes place almost exactly at the opening pressure. It requires a built-up or accumulated pressure of about 25 per cent above the opening pressure to lift the valve disk to maximum opening.

The pop valve (Fig. 6–32*B*) may have the same type of valve body; in fact, it may be impossible from an exterior inspection to distinguish between a relief valve and a pop valve of the same size. However, a setscrew in the rear of the valve body, which holds the blowdown ring in adjustment, usually identifies a pop or pop-safety valve. The fluid that passes the valve does not discharge freely as in a relief valve but is trapped in an annular space known as a *huddle* chamber.

In construction, the valve disk has a flange extending beyond the diameter of the seating contact; sometimes it is cupped downward slightly. A blowdown ring surrounding the valve seat fits quite closely around the valve-disk flange. Thus, when the fluid escapes through the valve seat, it enters the blowdown chamber and exerts a lifting pressure upon a much greater area of the valve disk This readily overbalances the spring and instantly opens the valve wide with a loud *pop*, hence the name. When vessel pressure falls to not more than 4 per cent, or 2 psi below the opening or *set* pressure, the valve snaps closed again. While blowing, a pop valve

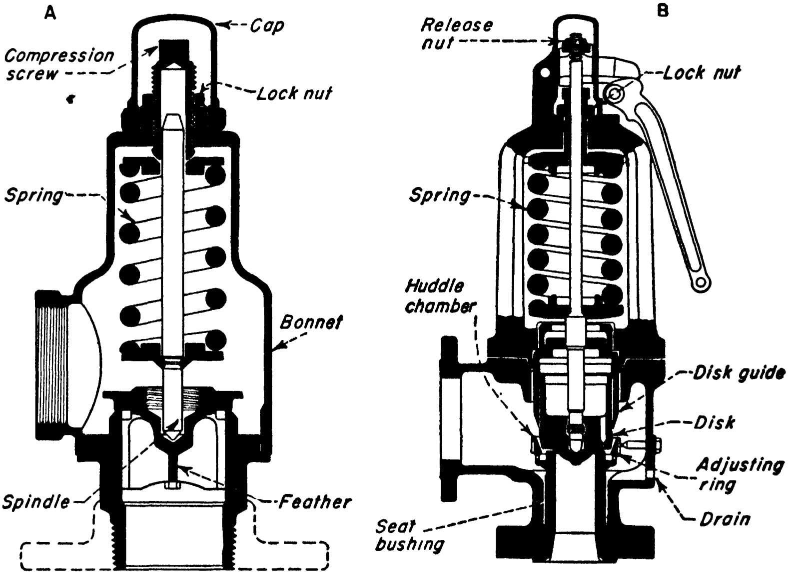


FIG. 6-32.—(*A*) Relief valves (used on liquids) close almost exactly at opening pressure. (*B*) Pop valves (used on vapor, gas) lower the pressure noticeably. (*Courtesy of Power.*)

should not allow a build-up of more than 5 per cent above set pressure.

A relief valve used for liquid service is usually located below the top of the vessel it protects. It may, however, be placed on the piping a short distance from the vessel; for longer distances, the connecting pipe must be larger than the nominal size of the safety device. If the vessel is tall and the relief valve is located considerably below the liquid level, the hydrostatic pressure (0.434 psi for each foot below the liquid surface) must be added to the vapor pressure in determining the set pressure.

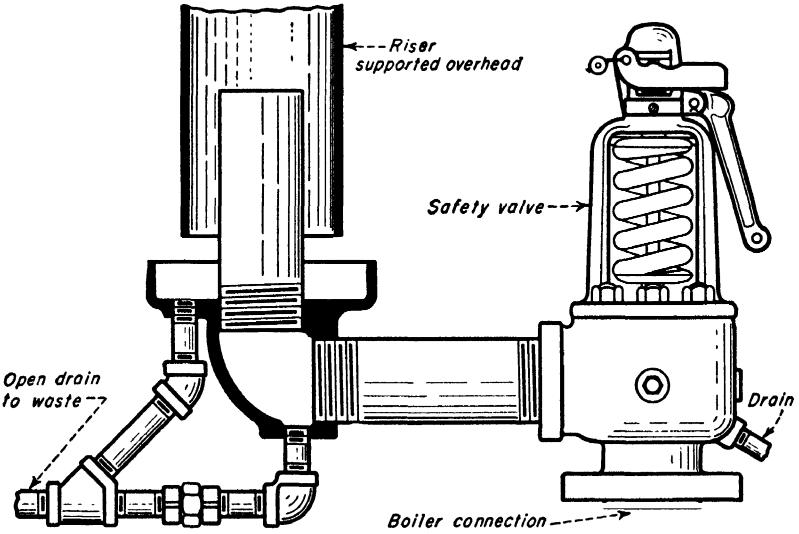


FIG. 6–33.—Safety-valve discharge piping must have suitable drain openings.

On the other hand, good practice dictates that a pop valve, for gas and vapor service, be mounted directly on the highest part of the vessel it protects. The discharge pipe for a safety valve should never under any circumstances be smaller than the valve. It should, where necessary, be protected from freezing. Pop-valve discharge lines that terminate outdoors usually have rain hoods over the outlets. When they are used in steam service, provide a drain hole in the elbow at the valve outlet (Fig. 6–33).

If the protected vessel is liable to be exposed to a serious fire that might envelop it, install a safety valve capable of relieving the greatly increased quantity of vapor that may thus be unexpectedly produced. Do not install bronze valves where the temperature is liable to go above 300 F.

Test the valves at least once a year, and more frequently if conditions warrant. This ordinarily makes it unnecessary (except as required by law) to raise the lifting lever periodically, on units so designed, for periodic testing is far more beneficial than periodic tripping. Maintain a record of each test. If the plant contains a large number of safety valves, a special bench, either fixed or portable, can be constructed for testing. Attach a steel plate about 13 by 13 by 1 in. to one end of the bench, and drill a $\frac{3}{8}$-in. hole through its center. Tap both sides of the hole for $\frac{1}{4}$-in. pipe fittings. Attach a compressed-air supply to the underside, and screw a fitting into the top side for testing pressure gages if desired. Provide four $\frac{7}{8}$-in. studs on a 10-in.-diameter circle around the air inlet for clamping down the safety valve.

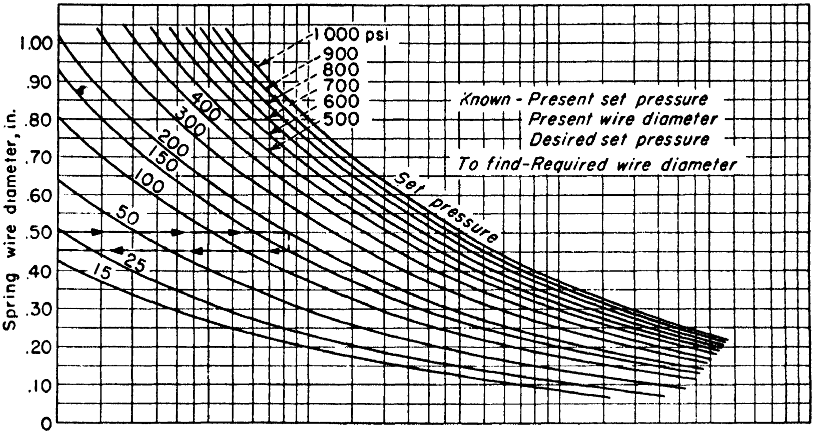


FIG. 6–34.—Follow course of arrows from initial spring-wire size to initial pressure line, then vertically to new pressure, and back horizontally to new wire size. (*Courtesy of Power.*)

Mount under the bench a 220 cu ft cylinder containing compressed air at about 2,200 psi. Connect it to the underside of the plate and to a header that serves several accurately calibrated test gages of different pressure ranges. Use copper tubing. The pressure of the plant compressed-air system usually is not high enough to test safety valves. Install needle valves at the cylinder outlet and under each test gage. Equip the valve at the cylinder with a long handle to facilitate accurate control. Suggested ranges of the test gages are 0 to 30 psi, 0 to 60 psi, 0 to 100 psi, 0 to 300 psi, and 0 to 500 psi, depending upon a plant's needs. A 3-in. vise is a useful tool to have on the bench.

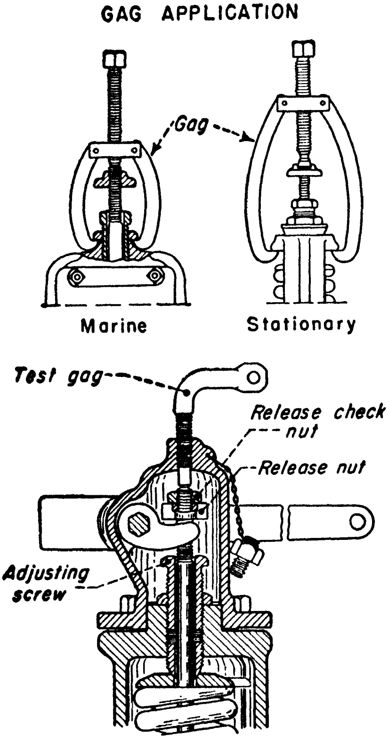


Fig. 6–35.—Adjustable gag holds safety valve closed during hydraulic test on vessel.

On the test plate, place a piece of live rubber about 9 in. in diameter and 3/16-in. thick, having a 1/2-in. hole in its center to fit over the air inlet. Place the safety valve on the gasket, if it is of the flanged type; otherwise screw it into a flange of the proper size, and place the latter on the gasket. By means of two or four finger clamps and the 7/8-in. studs, clamp the valve to the test plate.

Using the long-handle needle valve, admit air slowly, watch the pressure gage until the safety valve opens, and note the pressure. If a relief valve is being tested, the air will hiss as it leaks past the disk. A pop valve will open almost instantly with a loud pop.

Release the pressure, again raise it slowly to a few pounds per square inch below the set pressure, and make a leak test. Test for leakage by dipping a long-bristled varnish brush of appropriate width into a bucket of thick soap solution and drawing it slowly across the valve outlet so as to leave a film across the opening. If this bubble remains for a few seconds, it indicates a tight valve. Desirable leaktight pressures are given in Table 6–1. If the valve leaks, lift the disk from its seat, and wipe both surfaces to remove any adhering dirt particles. Continued leakage must be corrected by regrinding.

TABLE 6–1. LEAKTIGHT PRESSURE

Opening Pressure, Psi	Leaktight Pressure, Psi	
0–10	Opening pressure minus	1
10–25		2
25–50		3
50–100		5
100–175		7
175–250		10
250–350		15
350–500		20
500–1,000		25
1,000–2,000		35
2,000–5,000		100

Occasionally it may be necessary or desirable to change the operating pressure. This means changing the safety-valve setting. Most valve springs permit adjustment of about 10 per cent either way. If the contemplated change in operating pressure exceeds this amount, install a new spring. Of course, it is always safe to reduce the pressure, whereas the safety-valve body may not be able to withstand an increased pressure. Give this fact serious consideration before changing to a heavier spring.

In changing the spring size, the chart in Fig. 6–34 is useful. Since the diameter and over-all length of the spring are fixed by valve-body design, the change must be made in the diameter of the spring wire.

When changing the setting of a safety valve, it is extremely important that the stamping, either on the name plate or on the valve hex, be changed accordingly. Of equal importance, however, is determining whether the pressure vessel and connected piping will withstand the higher pressure. When making a hydrostatic

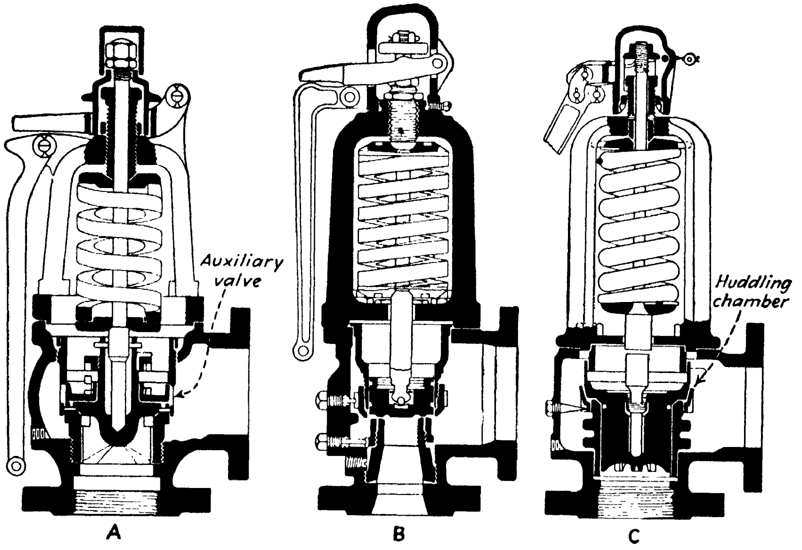


FIG. 6–36.—Boiler safety valves showing different popping actions.

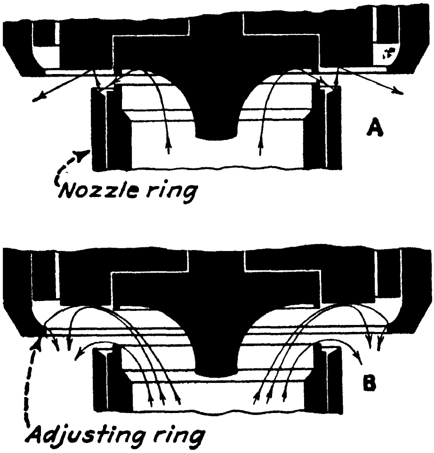


FIG. 6–37.—Steam pressure overcomes spring and lifts valve disk *A*. Then jet strikes nozzle ring and raises valve fully, *B*. (*Courtesy of Crosby Steam Gage & Valve Co.*)

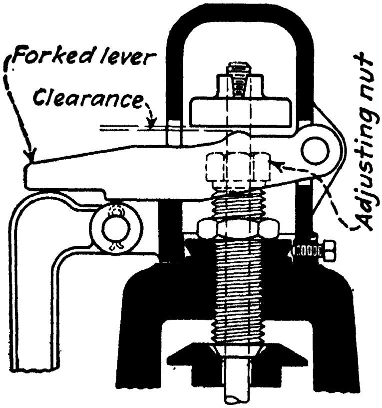


FIG. 6–38.—Adjust for correct popping pressure only when boiler is popping or pressure is considerably below the setting; otherwise the seat may be damaged.

test, hold the safety valve on its seat with a suitable gag (Fig. 6–35).

Only spring-loaded safety valves are permitted by boiler codes. They should pop clean without simmering and close without chatter or simmer within the required blowdown limits. To secure a clean pop and proper blowdown, the valve in Fig. 6–36*A* employs an auxiliary valve, the one in *B* the reaction energy in the steam velocity, and the one in *C* a huddling chamber. *B* has a flat seat as compared with a 45-deg seat in the others.

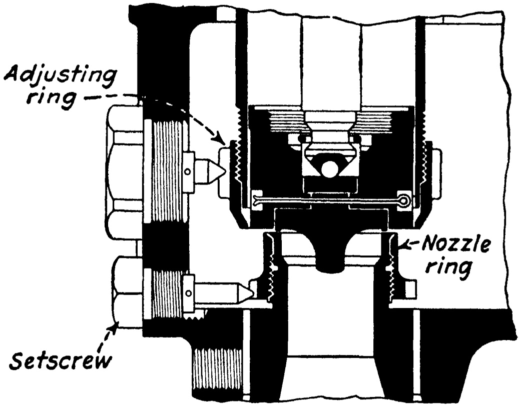


FIG. 6–39.—Set blowdown range by positioning the adjusting ring. Make increment settings with nozzle ring. (*Courtesy of Crosby Steam Gage & Valve Co.*)

For boiler service, the safety valve must discharge all the steam generated without allowing the pressure to rise more than 6 per cent above the maximum allowable working pressure. Blowdown—the difference between popping and closing pressures—is usually set at 4 per cent of the set pressure. Ordinary relief valves are not adjustable for blowdown; they trip at the set pressure and remain open until the pressure falls sufficiently to permit the opposing spring to force the disk to its seat.

The basic principle of blowdown adjustment of a safety valve centers on the kinetic energy of steam flow. When the force of the steam pressure acting under the disk overcomes the spring, the valve lifts slightly. The small jet of escaping steam is deflected from the nozzle ring (Fig. 6–37*A*) and assists in raising the valve farther. Downward deflection of steam from the adjusting ring in Fig. 6–37*B* then assists in holding the valve open until the blowdown point is reached.

To change the popping pressure, loosen the adjusting-sleeve lock nut (Fig. 6–38). Raise the boiler pressure until the valve pops; then turn the adjusting sleeve down to raise the popping pressure, or up to lower it. To prevent turning the disk on its seat, always make adjustments while the valve is popping or when the boiler pressure is at least 15 psi below the valve reseating pressure.

Adjust the blowdown range by raising or lowering the adjusting ring. In valves with an upper ring, lowering it increases blowdown; with only a nozzle ring, blowdown is increased by raising it. In valves having two rings, the upper one provides the major adjustment and the lower one the micrometer adjustments.

Consider the unit in Fig. 6–39. To change the adjusting-ring setting, remove the upper setscrew. Insert a screw driver, and engage it in the ring notches. Pushing the notches to the right raises the ring and shortens the blowdown. If the adjusting ring is raised too high in attempting to shorten the blowdown, it does not control the valve, and further raising has no effect. Opening lift may be very low (prolonged simmer), and closing may be indistinct and dragged out. Never move the ring more than 10 notches without retesting the valve.

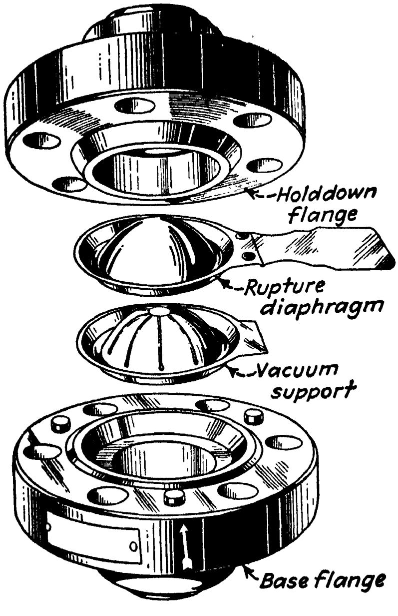


FIG. 6–40.—Rupture disks are designed to let go before the vessel fails. (*Courtesy of Black, Sivalls & Bryson.*)

Do not change the nozzle-ring position until all possible positions of the main ring have been tried. Never move the former more than five notches from its original position, and not more than two without retesting the valve. To adjust, remove the lower setscrew, and use a screw driver as before. Lowering the ring decreases the power of the low-lift pop action, and raising it increases the action.

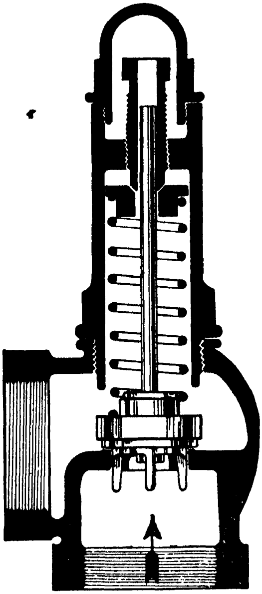


Fig. 6-41.—Relief valves protect vessels against overpressure and pumps against closed discharge.

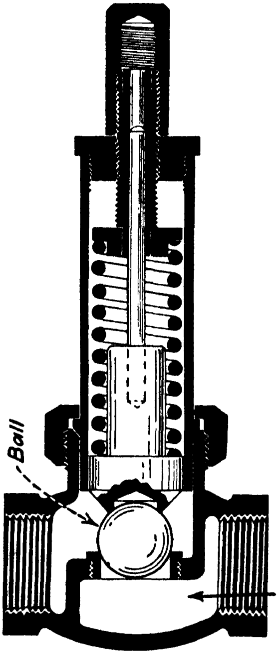


Fig. 6-42.—Relief valve with ball inner valve.

When adjusting for blowdown near the valve's minimum, raising the ring slightly corrects for simmering without sensibly affecting blowdown. On the other hand, slightly lowering the ring lessens blowdown slightly in a valve operating properly otherwise. When adjusting a valve, always keep a record of the number of notches each ring is moved. This assures returning to the original settings in case of error and prevents getting the valve badly out of adjustment.

Worn or damaged seats cause most troubles. Slight wear is corrected by lapping, but it must be done carefully to keep the seats true. On valves with flat level seats, lap the seat and disk on a cast-iron lapping block, which is tested frequently on a good surface plate. Use a fine-texture grinding compound, and apply it sparingly. Let the weight of the block do the work; do not press down. Never lap the disk against its seat, because this procedure may scratch the surfaces.

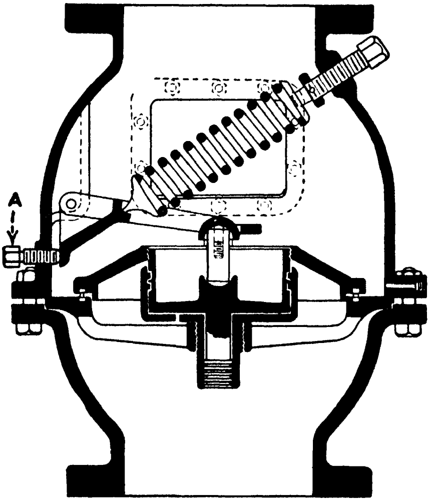


FIG. 6–43.—Back-pressure valve maintains exhaust pressure on process or heating lines.

Rupture disks, also known as *safety heads* (Fig. 6–40), are carefully designed "weak spots" that let go when the pressure or vacuum inside a closed vessel exceeds a certain safe maximum value. They should be considered as strictly an emergency relief—a safety measure of last resort, which blows out in time to save the vessel.

Where the safety valve is subject to excessive corrosion or gumming from material in the system, it is good practice to install a rupture disk in the pipe ahead of the valve, to keep the material from entering it. The rupture disk should be rated at or slightly below the valve-set pressure. Should pressure in the system exceed this amount, the diaphragm bursts. This allows the material to enter the valve, which then opens and relieves the pressure.

Recommended practice is to equip the vessel with a safety valve set slightly above the working pressure and large enough to hold the internal pressure from exceeding a certain value. Then back up the safety valve with a rupture disk calibrated to burst before the tank can.

TABLE 6–2. SAFETY-HEAD DIAPHRAGMS

Diaphragm material	Diameter, in.										
	½	¾	1	1½	2	3	4	6	8	10	12
	Practical minimum bursting pressure										
Aluminum or aluminum, lead lined*. .	150	150	110	100	60	40	35	25	20	19	18
Carbon steel*.	1,750	1,300	800	500	400	275	200	150	110	90	60
Copper or copper, lead dipped*. . . .	600	475	400	300	190	130	120	90	80	70	50
Gold or gold veneer.	600	475	400	300	190	130	120	90	Electroplated only		
Monel metal*.	800	750	600	450	400	300	275	220	200	150	125
Nickel.	1,200	1,100	850	600	375	250	150	140	250	225	200
Platinum.	200	180	135	100	75	80	70	60	Made specially		
Silver.	200	180	135	100	75	80	70	60	Electroplated only		
Stainless steel*	1,500	1,200	800	600	450	400	300	275	200	180	150

* Standard diaphragm materials.

Minimum bursting pressures in some large sizes (*e.g.*, nickel—6 in. minimum, 140 psi; 8 in. minimum, 250 psi) are higher than minimum pressures for small diameters of the same metal. This apparent discrepancy is explained by the fact that, in these particular cases, very thin metal is not obtainable in widths required for larger diaphragms.

Disks are made for a wide range in pressures (Table 6–2) to rupture within about 5 per cent of rated operating pressures. They are made of aluminum, copper, steel, stainless steel, monel, bronze, admiralty metal, nickel, Everdure, Herculoy, silver, gold, or platinum. In addition, the disk may be given a protective coating of lead, latex, bakelite, or neoprene.

Relief valves protect positive-displacement pumps against a closed discharge, tanks against rupture, and steam-using equipment against overpressure. Styles differ, but in general they use spring loading. The unit in Fig. 6–41 has an angle body and flat disk, whereas the one in Fig. 6–42 uses a ball inner valve.

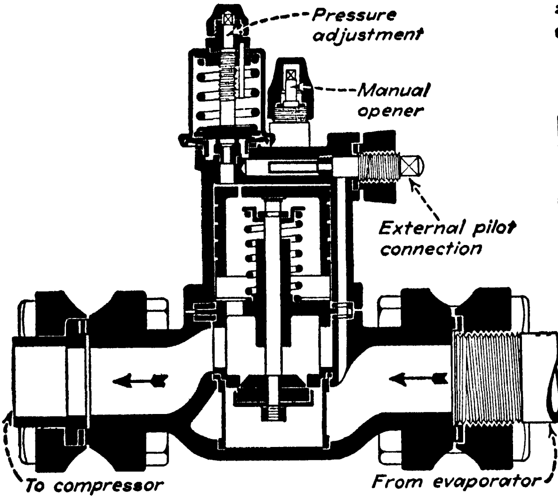


FIG. 6–44.—Valve holds predetermined back pressure on refrigerating evaporator. (*Courtesy of Alco Valve Co.*)

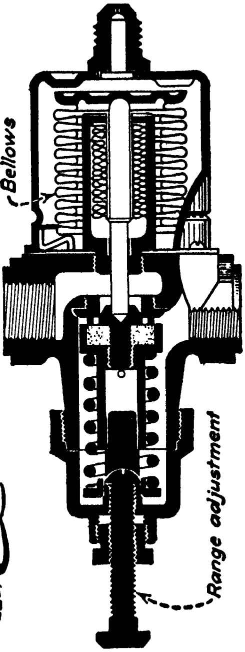


FIG. 6–45.—Head pressure from refrigerating compressor acts on the bellows to admit cooling water to condenser. (*Courtesy of Penn Electric Switch Co.*)

Another design (Fig. 6–43) serves to hold a back pressure on the machine exhaust and yet acts as a relief for pressures above its setting. These units can also be used as atmospheric valves on condensing turbines and engines. In operation, the spring and lever mechanism maintains a constant relation between spring force and back pressure regardless of the valve position (opening). To change from back pressure or vacuum operation to free exhaust, turn jackscrew *A* all the way in to hold the valve off its seat. The valve in Fig. 6–44 maintains a predetermined back pressure on refrigeration evaporators regardless of fluctuations in compressor suction pressure. In Fig. 6–45, the refrigerating compressor head pressure acts on the bellows to open the water supply to the condenser or cylinder jackets.

CHAPTER VII

TEMPERATURE VALVES

Wherever power-plant or process work calls for heat or cold, temperature-control valves find wide use. Although the conventional diaphragm valve may be hooked up with special automatic devices to control temperature, this chapter describes only those valves actuated by the effect of temperature changes on their self-operated mechanisms.

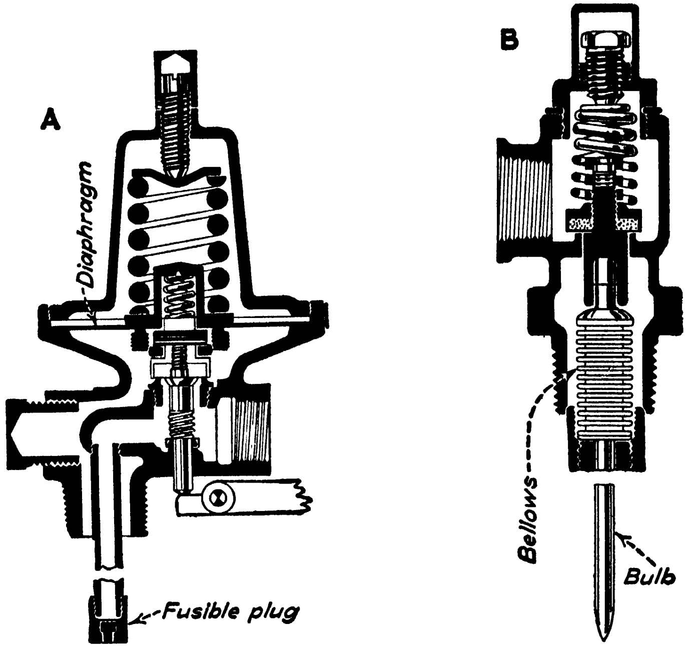


Fig. 7-1.—Temperature-relief valve *A* uses a fusible plug. Another design, *B*, has a sensitive bellows.

Operating motors are either bellows or metal-diaphragm units using liquid or gas expansion or vapor pressure for the actuating medium, as in pressure-operated thermometers. Most diaphragm valves have a sensitive bulb and capillary tube, whereas bellows units either have the operating fluid contained in the bellows or have a connected capillary and sensitive bulb.

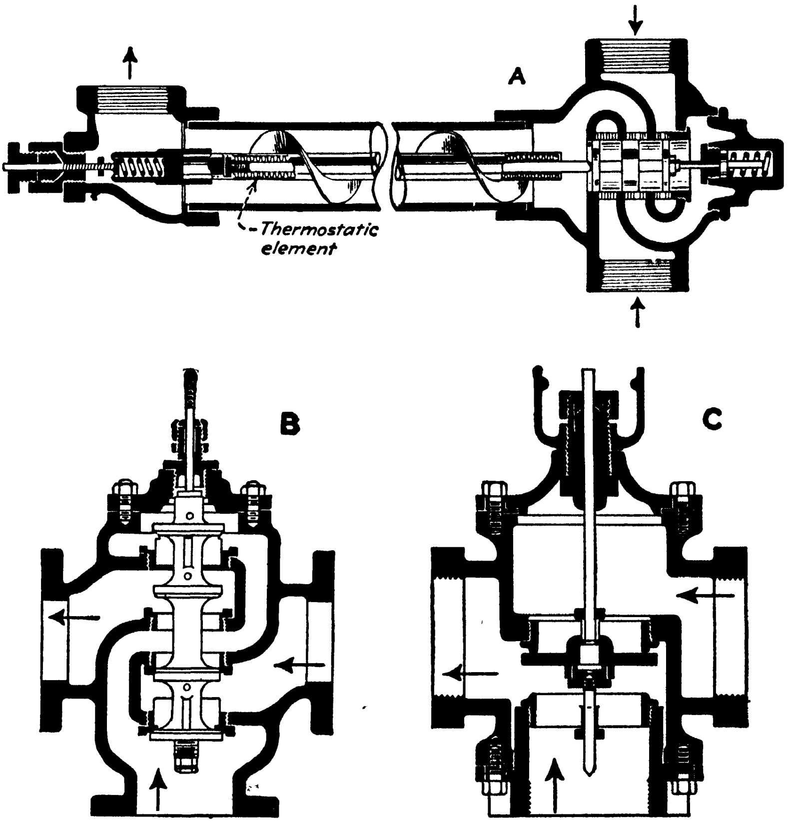


FIG. 7-2.—**Inside construction of three-way or mixing units: piston valve *A*, balanced poppet *B*, single disk *C*.**

Simplest of all are the temperature-relief valves (Fig. 7-1). Unit *A* uses an ordinary fusible plug for temperature relief and a spring-opposed diaphragm-operated valve for pressure relief. Either can function without interfering with the other. The only objection is that this device must be removed from the vessel to replace the fusible plug after it melts from high temperature. A different design, *B*, uses a bellows containing a volatile fluid as the actuating element. High temperature causes the bellows to expand and lift the valve off its seat; but, when the temperature returns to normal, the valve closes and is again ready for service.

These valves protect hot-water tanks where faulty heater operation may permit excessively high temperatures, which, upon tank rupture, cause serious explosions. They also serve on air-compressor discharge lines to release air and give warning of internal fire or cooling-water failure. The bellows is adjustable, and fusible plugs are available for a wide temperature range.

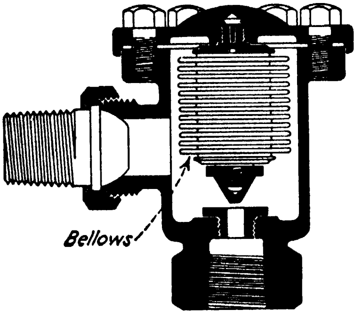


Fig. 7-3.—Select the thermostatic trap having the best filling agent.

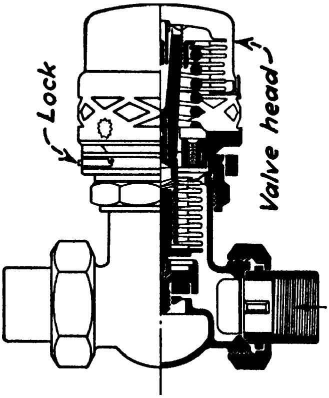


Fig. 7-4.—Thermostatic-bellows-operated valve serves heating radiators.

Three-way valves are designed to mix hot and cold liquids; to divert flow through either of two lines; or, in reverse, to maintain the desired temperature in compressor and internal-combustion-engine jackets, condensers, and unit coolers. Figure 7-2*A* uses a piston valve and passes the mixture around the operating bellows. Unit *B* has a balanced poppet and *C* a single-disk inner valve. Both may be operated by a bellows and bulb arrangement or a rubber diaphragm with external control. For mixing purposes, pressure in both supply lines should be about the same to keep the plug as nearly balanced as possible. In this capacity, a three-way valve is a throttling device balancing the flow from both sources to maintain uniform outflow.

When diverting flow, the valve disk closes one or the other port. To maintain engine-jacket or condenser temperature, the flow goes through the valve in reverse and acts to recirculate most of the hot water, adding only enough cool water to maintain the desired temperature in the cooled apparatus.

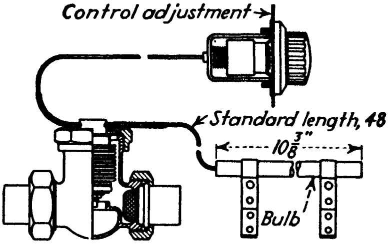


FIG. 7-5.—Concealed radiator uses both bulb and bellows.

The radiator trap in Fig. 7-3 uses water as the thermal-sensitive filling medium so that its reaction parallels all conditions of pressure and temperature in the steam system. Since the bellows functions inside its free length, the unit expands and closes the valve if rupture occurs. This causes the heating unit or radiator to cool, which gives immediate notice of trouble. For low-pressure service, the bellows is of brass or bronze; for pressures above 25 psi it is of nickel alloy.

The radiator-inlet valve in Fig. 7-4 uses a bellows seal instead of stem packing. The monel actuating bellows contains a sensitive expansible liquid, sealed in under vacuum, suitable for steam pressure up to 15 psi. These valves are factory-adjusted to control at any desired temperature between 60 and 75 F. The temperature is set by turning the valve head, which increases or decreases the spring tension on the bellows. The setting can be locked to prevent tampering. These units replace manually operated valves to give throttling control of steam in relation to room temperature acting on the sensitive bellows. The thermostatic element is insulated against heat transfer from steam piping.

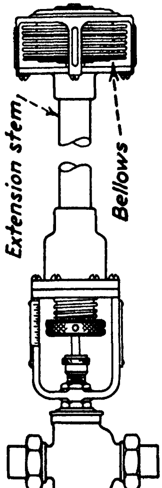


FIG. 7-6.—Actuating bellows can be mounted inside a tank or other vessel.

The arrangement is slightly different for a concealed radiator (Fig. 7-5). Here the control uses a bulb that mounts at the cool inlet grill, and a bellows control is located at the heated-air outlet. Both units connect to the bellows-operated valve through capillary tubing and act in unison to throttle the steam supply and maintain the desired temperature at the breathing line.

Bellows units are adaptable for mounting in the open atmosphere or for controlling the temperature of liquids in open tanks or vats. In chemicals likely to attack the bellows, or liquids carrying suspended matter, a protective cap covers the sensitive element. Figure 7-6 shows the extension stem that permits locating the sensitive element inside a tank. The thermostatic end is removable, and the only installation requirement is to drill a small hole in the tank.

An integral-bellows device (Fig. 7-7) acts as an expansion or throttling valve for refrigerating-system evaporators. Control is by exit-gas temperature. Suction gas, on its way to the compressor, passes over the actuating bellows. It contains a thermal-sensitive fluid that reacts and positions the valve to admit more or less refrigerant according to gas-temperature variations. Accurate control of gas superheat provides maximum coil capacity at all times and helps the evaporator to deliver more nearly its rated capacity. Figure 7-8 shows how evaporator capacity varies with gas superheat temperature.

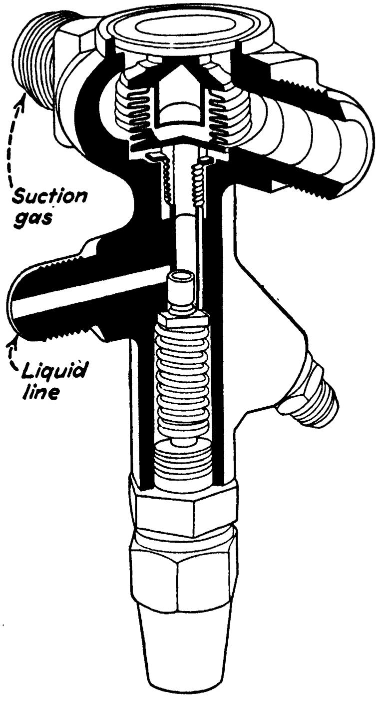


FIG. 7-7.—Bellows unit acted on by suction gas regulates liquid flow to refrigerating-system evaporator to hold superheat.

Bellows-operated valves complete with capillary and bulb (Fig.

7–9*A*) regulate temperature in refrigeration, air-conditioning, heating, and condensate-removal work. In the latter, the bulb is placed in the fluid being heated so that, when it reaches optimum temperature, the valve closes and the condensate backs up in the steam jacket (Fig. 7–10). Thus, by flooding or draining the coil or jacket, the effective heating surface is changed in direct relation to the load. The thermostatic element is liquid-filled, and the valve acts gradually to give true throttling control.

In vapor-pressure-actuated valves, the capillary tube extends to the bulb bottom (Fig. 7–9*B*) so that the tube and bellows always remain filled with liquid. If the vapor passes over to the bellows, it may act as a condenser and lower the pressure until enough liquid accumulates to fill it. In the meantime, the valve acts erratically.

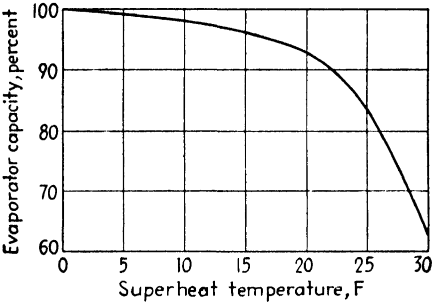


FIG. 7–8.—Evaporator capacity varies with gas superheat temperature.

The low-temperature refrigeration expansion valve (Fig. 7–11) uses a bellows and sensitive element *A* that connects to the evaporator inlet and another unit *B* (for superheat temperature) that connects to the evaporator outlet. The difference in temperature between these two elements governs the flow of refrigerant to keep the evaporator filled during compressor operation. Both power elements are charged with ethane, and their pressure changes about 1 psi for each degree of change in feeler-bulb temperature at 100 F. This allows the valve orifice to be relatively small with the needle working far off its seat, which minimizes plugging from moisture, dirt, or wax.

The design is such that the main-valve orifice opens only when the superheat feeler bulb is warmer than the evaporator bulb by an amount equal to the differential setting. Thus, on start-up, some means must be incorporated to create a temperature difference between the bulbs, or the valve will not open. This is accomplished by the by-pass valve, which remains open and feeds a constant supply of refrigerant. When the system is shut down, a solenoid-operated or other automatic valve must close the main liquid line.

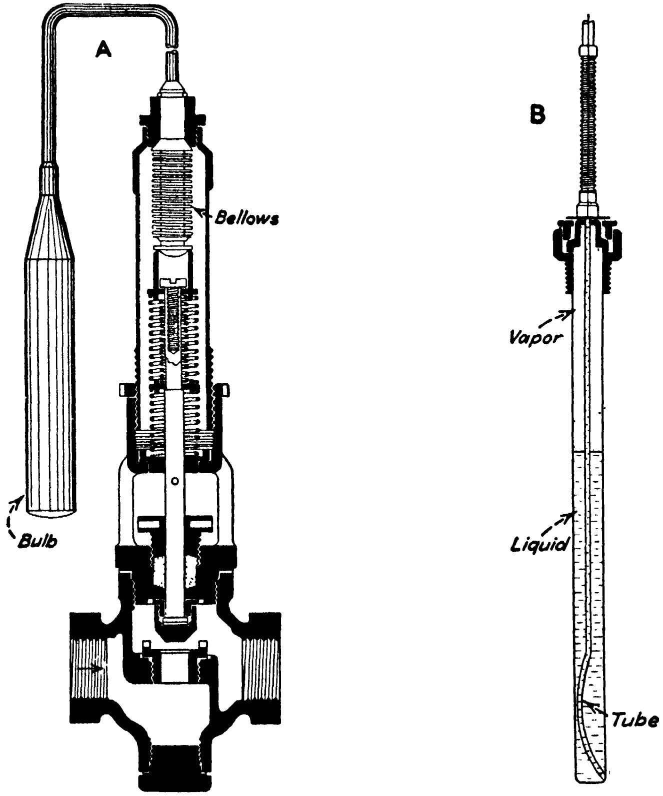


FIG. 7-9.—In vapor-pressure-actuated valves, capillary tube and bellows remain completely filled with the liquid.

The unit in Fig. 7-12A functions as the central control for a gas-fired hot-water heating system. The only electric circuit is that from the space thermostat to a thermal motor inside the mechanism. As long as the pilot flame burns, heat applied to sensitive

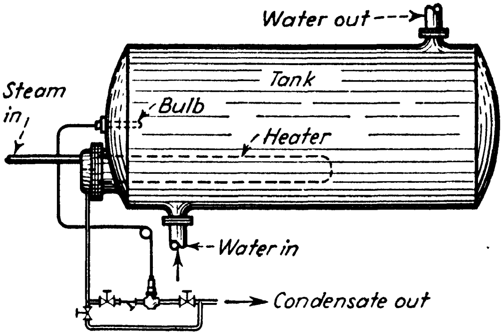


Fig. 7-10.—Temperature-control valve regulates condensate flow.

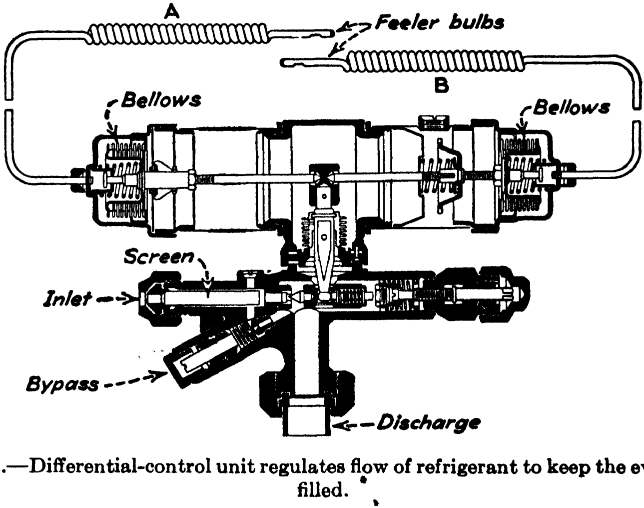


Fig. 7-11.—Differential-control unit regulates flow of refrigerant to keep the evaporator filled.

element *a* expands bellows *b* and raises bell crank *c*, which permits spring *d* to open the valve through snap-action mechanism *e*. Spring *f* closes the valve with a snap if the pilot flame fails. Bellows *g*, adjusted through knob *h*, closes the valve if the water temperature

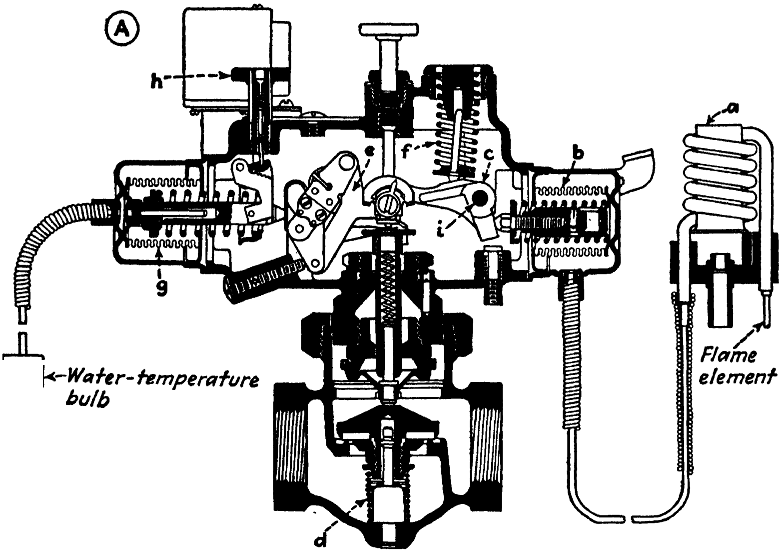


FIG. 7-12*A*.—Water bulb and flame-failure side of controller for gas-fired hot-water system.

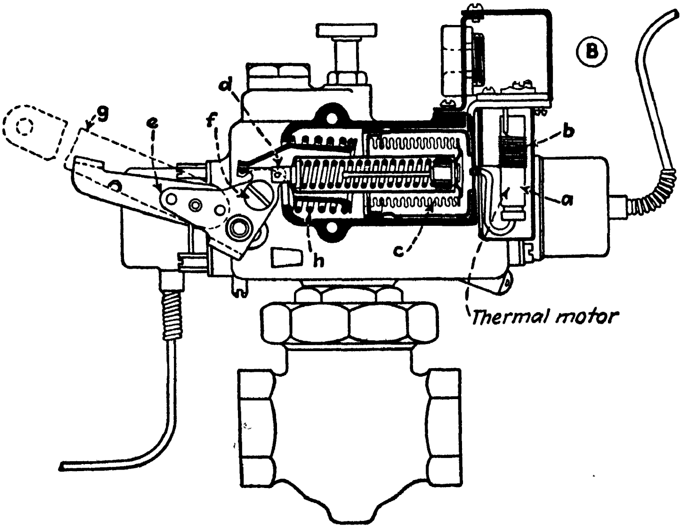


FIG. 7-12*B*.—Side view of Fig. 7-12*A* showing thermostat-controlled heater (thermal motor) on sensitive element.

exceeds the throttle setting. On a decrease in temperature, the bellows snaps the valve open.

Another view (Fig. 7–12*B*) shows the thermal motor *a* controlled by a room thermostat. Current passing through resistance *b* vaporizes liquid in the bulb, which increases the pressure on bellows *c* and forces stem *d* to the left. This rotates rocker *e* around shaft *f* to operate damper arm *g*. Another lever (not shown) connects to shaft *i* in Fig. 7–12*A* to operate snap-action mechanism *e* and open the valve. When the electric circuit opens or current fails, pressure on bellows *c* (Fig. 7–12*B*) falls, and spring *h* reverses the operation.

The action of the throttling element opens or closes the gas-supply valve gradually until it passes sufficient fuel to maintain the desired temperature under existing load. The throttling movement starts at the temperature as set with knob *h* in Fig. 7–12*A*.

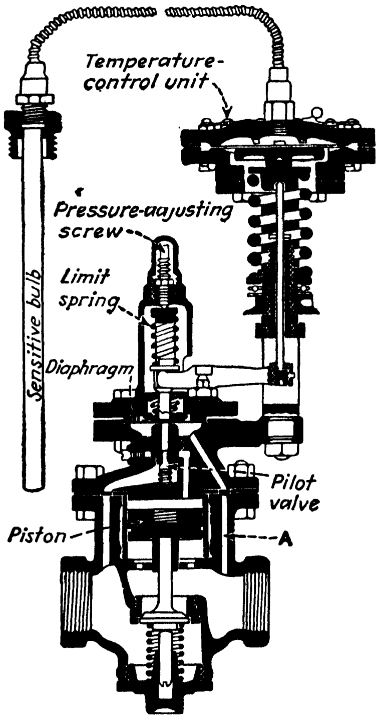


FIG. 7–13.—Combination pressure-reducing and temperature-control valve.

The combination unit in Fig. 7–13 puts the pressure-reducing and temperature-regulating functions into one body. The setting of the pressure-adjusting screw determines the maximum value of reduced pressure. The limit spring acting downward opens the pilot valve and admits high-pressure fluid to the top of the piston. When the piston opens the main valve, reduced pressure flows up passage *A* to the underside of the metal diaphragm. This balances the limit-spring loading to adjust the pilot valve. Temperature control comes from the upper diaphragm, which is actuated by vapor pressure generated in the sensitive bulb. The downward movement of the diaphragm (high temperature) acts through the stem and lever to oppose the pressure-reducing limit spring. This lightens the downward force on the pilot-valve diaphragm, and the reduced pressure then raises the diaphragm enough to reposition the pilot valve and throttle the main valve according to temperature requirements.

CHAPTER VIII

HYGROSTATS AND PSYCHROMETERS

In air-conditioning work, the amount of moisture in the air must be controlled within certain limits to maintain comfort conditions or an atmosphere suited for the process work involved. For instance, the humidity in printing plants must be held fairly uniform from day to day or the paper sheets will shrink and stretch as the outside weather changes, and faulty printing work will result. On the other hand, precision gages must be kept in a relatively dry atmosphere to prevent rusting and at a uniform temperature so that they remain accurate to within 0.0001 in.

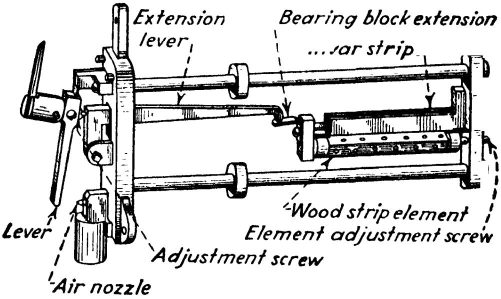


FIG. 8-1.—A wood-strip humidistat for pneumatic control. (*Courtesy of Brown Instrument Co.*)

Relative humidity is the commonly accepted measure of moisture in air for industrial work. It is often defined as (1) the ratio in percentage of the partial pressure of the moisture in the air to the saturation pressure, or vapor pressure of water at the same temperature; or (2) the actual weight of moisture in a given volume of air-vapor mixture divided by the maximum weight that the same volume can hold (at the same temperature) without causing condensation. The numerous instruments developed to register and control relative humidity may be divided into hygrostats (humidistats) and psychrometers.

Hygrostats operate on the principle that hygroscopic material expands and contracts with variations in its moisture content. This changes as the humidity of the surrounding air varies. Materials commonly used are wood, paper, animal membrane, and hair.

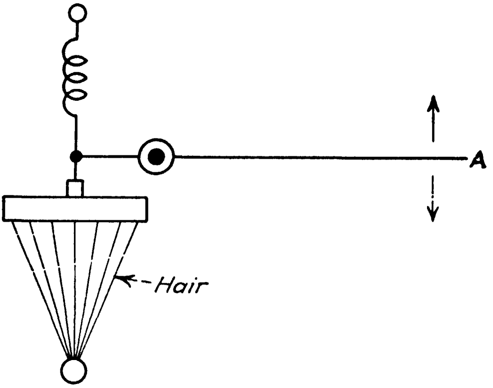


FIG. 8-2.—Strands of hair stretch or shrink to follow relative-humidity variations.

Figure 8-1 illustrates a hygrostat using wood as the sensitive element. Expansion and contraction of the wood, in response to relative-humidity changes, actuates the flapper in a one-pipe pneumatic control system. Another device uses a long strip of paper anchored at one end and folded back and forth across small rollers with the free end attached to an indicating pointer. A membrane unit can be used instead of paper.

The hair hygrostat in Figs. 8-2 and 8-3 moves a pointer, a pen, or electric contacts. Stretched taut by an opposing spring, the hair element contracts when the humidity falls and expands when it increases. The sensitive element must be mounted where air movement is unrestricted and where a minimum of dust will accumulate on the hair strands.

Range adjustment is made by the dial (Fig. 8-3), which is calibrated from 30 to 70 per cent relative humidity. The differential is factory-set at 2 to 3 per cent relative humidity throughout the full range. The electric-contact mechanism includes a permanent magnet that assures positive snap action to both positions. A simple tension release prevents the hair strands from being stretched by extremes of humidity or mechanical strains. The device can be used with solenoid-operated valves or dampers, relays, fans, heaters, compressors, pumps, and other humidifying and dehumdifying apparatus.

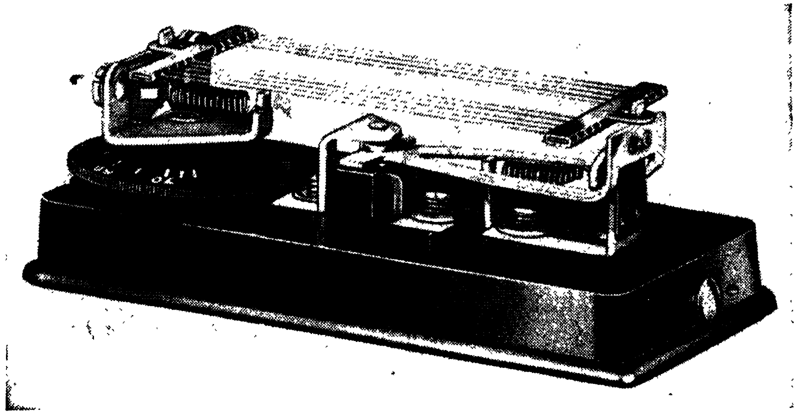


FIG. 8–3.—Hair instrument operates on a differential of 2 to 3 per cent. (*Courtesy of Penn Electric Switch Co.*)

Some manufacturers prefer a hair element on recorders because of its accurate response to changes in humidity over the entire range. Its response is slow but fast enough to record accurately all normal changes. Hair elements are not recommended for temperatures over 140 F, although one manufacturer produces a device that operates up to 200 F. A membrane element serves best on controllers because of its fast response to changes in humidity. Over a wide range, however, it is not so accurate in response as a hair element. Membranes should not be used at temperatures over 120 F. Care must be exercised to keep the sensitive elements of hygrostats clean and in a current of air to avoid introducing a lag in their response.

The moisture recorder in Fig. 11–16 detects and controls the moisture content of such hygroscopic materials as paper and textiles

TABLE 8-1.—RELATIVE HUMIDITY, TEMPERATURE 30 to 210 F*

Dry bulb, F	Difference between readings of wet and dry bulbs in degrees Fahrenheit																									Dry bulb, F
	2	4	6	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	45	50	55	60	70	
20	70	40	12																							20
30	78	56	36	16																						30
40	83	68	52	37	22	7																				40
50	87	74	61	49	38	27	16	5																		50
60	89	78	68	58	48	39	30	21	13	5																60
70	90	81	72	64	55	48	40	33	25	19	12	6														70
80	91	83	75	68	61	54	47	41	35	29	23	18	12	7	3											80
90	92	85	78	71	65	58	52	47	41	36	31	26	22	17	13	9	5	1								90
100	93	86	80	73	68	62	56	51	46	41	37	33	28	24	21	17	13	10	7	4						100
104	93	86	80	74	69	63	58	52	48	43	39	35	31	27	23											104
108	93	87	81	75	70	64	59	54	49	45	41	37	33	29	26	22	19									108
112	93	87	81	75	70	65	60	55	51	47	42	38	35	31	28	24	21	18	15							112
116	93	88	82	76	71	66	61	56	52	48	44	40	36	33	29	26	24	20	17	14						116
120	94	88	82	77	72	66	62	57	53	49	45	41	38	34	31	28	25	22	19	16	10					120
124	94	88	83	77	72	67	63	58	54	51	46	43	39	36	33	29	27	24	21	18	13					124
128	94	89	83	78	73	68	64	59	55	52	47	44	40	37	34	31	28	25	23	20	15					128
132	94	89	83	78	74	69	65	60	56	53	49	45	42	39	35	32	30	27	24	22	16	11				132
136	94	89	84	79	74	69	65	61	57	53	50	46	43	40	37	34	31	28	26	24	18	13				136
140	94	89	84	79	75	70	66	62	58	54	51	47	44	41	38	35	33	30	27	25	19	14	10			140
144	95	89	84	80	75	71	67	63	59	55	52	48	45	42	39	36	34	31	29	26	21	16	11			144
148	95	90	85	80	76	71	67	63	60	56	53	49	46	43	40	38	35	32	30	28	22	17	13			148
152	95	90	85	81	76	72	68	64	60	57	53	50	47	44	41	39	36	33	31	29	23	19	14	10		152
156	95	90	85	81	77	72	69	65	61	58	54	51	48	45	42	40	37	34	32	30	24	20	15	11		156
160	95	90	86	81	77	73	69	65	62	58	55	52	49	46	43	41	38	35	33	31	25	21	17	13		160
164	95	91	86	82	78	73	70	66	62	59	56	52	49	47	44	41	39	36	34	32	26	22	18	14		164
168	95	91	86	82	78	74	70	67	63	60	56	53	50	47	45	42	40	37	35	33	28	23	19	15		168
172	95	91	86	82	78	74	71	67	64	60	57	54	51	48	46	43	41	38	36	34	28	24	20	16		172
176	96	91	87	83	79	75	71	68	64	61	58	55	52	49	46	44	42	39	37	35	29	25	21	17	10	176
180	96	91	87	83	79	75	72	68	65	62	58	55	52	50	47	45	42	40	38	35	30	26	22	18	11	180
184	96	92	87	83	79	76	72	69	65	62	59	56	53	50	48	45	43	41	38	36	31	27	22	19	12	184
190	96	92	88	84	80	76	73	69	66	63	60	57	54	51	49	46	44	42	39	37	32	28	24	20	14	190
200	96	92	88	84	80	77	74	70	67	64	61	58	55	53	51	48	46	43	41	39	34	30	26	22	16	200
210	96	93	88	85	81	78	75	71	68	65	62	60	57	54	52	49	47	45	43	41	36	32	28	24	18	210

---

* *Courtesy of Foxboro Co.*

by measuring their electrical resistance. In the continuous drying and conditioning of such material, accurate measurement and automatic control of the final moisture content are both of great importance (Chap. XI).

Another intricate device that registers humidity uses a series of eight gold-wire coils for the full range of humidity. A special covering material applied over the coils absorbs moisture from the air. This lowers the resistance between the coil turns to unbalance an electric circuit and register the relative humidity.

The difference in thermal conductivity between air and water vapor can also be used to indicate humidity. The air whose humidity is to be measured is passed over a platinum wire that forms one arm of a bridge circuit. The opposite arm consists of another platinum wire sealed in a dry-air atmosphere. Both arms carry current and are maintained at an elevated temperature. Heat lost by the wires depends on the variation of the heat conductivity of the surrounding medium. Thus the wire temperatures and hence the degree to which the bridge is out of balance are determined by the moisture content of the air surrounding the measuring wire. The indicator can be calibrated to read directly in percentage relative humidity.

Psychrometers function by reason of a temperature drop caused by water evaporation. This drop varies directly with the rate of evaporation, which in turn is inversely proportional to the amount of water vapor already in the air.

If a thermometer bulb is fitted with a thin water-soaked cloth, for instance, and the air is passed rapidly over it, the temperature registered by the instrument will fall a definite amount depending on the amount of moisture in the air. This is known as the *wet-bulb temperature.* The reading of an ordinary thermometer (without the wet cloth) is called the *dry-bulb temperature.* Taking the dry-bulb reading and the difference between it and the wet-bulb reading, the relative humidity can be found in Table 8–1. Readings of relative humidity can easily be made with a sling psychrometer consisting of two pencil thermometers mounted on a stick that can be whirled rapidly. One bulb is kept moistened with a cotton-wick sleeve.

Figure 8-4 shows a simplified arrangement of a dry- and wet-bulb vapor thermometer interconnected to one pointer to indicate relative humidity directly. A small water spray keeps one bellows

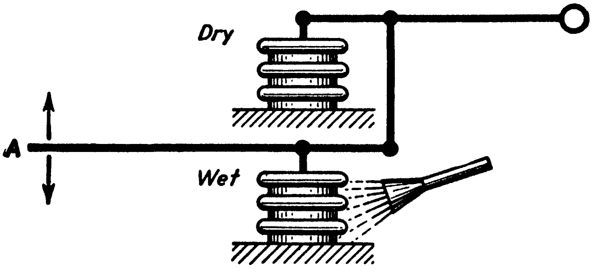


Fig. 8-4.—Dry- and wet-bulb thermometer registers relative humidity directly.

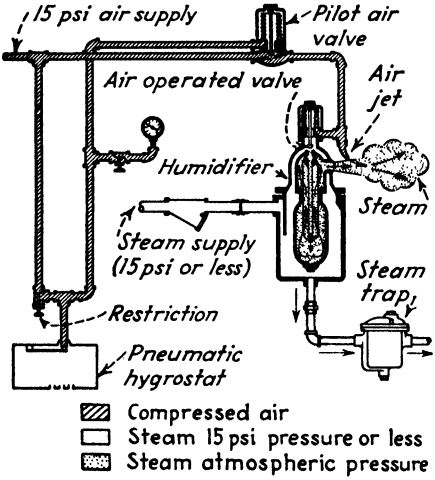


Fig. 8-5.—Device using steam to adjust humidity in a factory. (*Courtesy of Armstrong Machine Works.*)

wet, and air at a velocity of 15 to 17 ft per sec is blown over it by a small fan. Any of the thermometers described in Chap. IV can be adapted to read wet-bulb temperature. In most cases, the sensitive bulb is then made of silver-plated copper or stainless steel.

In practice, methods of keeping the bulb wet vary. Some instruments use cotton wicking that hangs in a water reservoir, and others use a porous alundum or ceramic sleeve that slips over the bulb. A small head of water from an inverted bottle or other water supply keeps the sleeve wet. The porous tubes serve best in kilns and driers or where deterioration and clogging of the cotton wick may result from chemical vapors or foreign matter. For best results, only distilled water should be used.

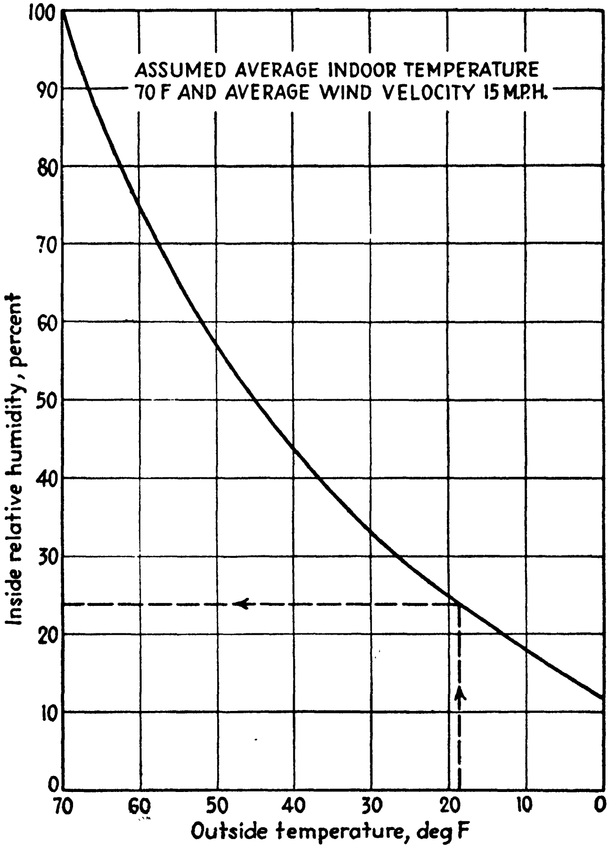


FIG. 8–6.—Relative humidities at which condensation appears on windows. (*Courtesy of Johnson Service Co.*)

Relative humidity can be controlled by any of the devices just described or by two-element thermometers. The dry-bulb thermometer then regulates the heat supply, and the wet-bulb unit controls the humidifying unit (Fig. 8–5) or acts as the temperature regulator on the spray water in the dehumidifier.

If a gradual readjustment of the humidity is desirable, a hygrostat can be readjusted by a remotely located thermostat. In winter, for instance, the humidity cannot be maintained so high as in the summer because wall and window temperatures often reach the dew point. This results in objectionable condensation (Fig. 8–6). A thermostat mounted near the windows can therefore serve to readjust the hygrostat so as to maintain the relative humidity at such a value that moisture will not condense on cold surfaces.

CHAPTER IX

CONTROL MECHANISMS

In previous chapters, we discussed primary elements (devices that detect pressure, liquid level, etc.) and regulating valves (mechanisms that control fluid flow). To complete the control system, these devices must be connected to a power source so that the primary element can either force a remote indicator to assume a definite position or force a valve to move and correct for any change in the controlled medium. To do this job satisfactorily, the complete system must be sensitive, powerful, stable, and rugged.

The primary element must detect and immediately respond to the slightest change in the varying medium, or it may go far beyond permissible limits. To illustrate lags that may upset controller sensitivity, assume, for simplicity, that a regulator is to maintain constant hydraulic pressure by controlling pump speed. Here there is (1) a short interval between the pressure change and the response of the regulator, (2) an interval between the response of the regulator and the time the pump changes speed, and (3) an interval between the pump-speed change and the pump-pressure response. The total of these time intervals equals the total time lag between detection of change and correction. If this period is too great, the pressure may deviate beyond the allowable limits and cause a swing, or hunting, that is difficult to correct.

A control system must have sufficient power not only to operate the valve or damper but also to overcome accidental friction or abnormal momentary pressure that may occur in any part of the system. Power in excess of normal requirements is essential because it assures absolutely positive operation.

More specifically, automatic controllers consist of four essential parts: (1) a primary sensitive element, (2) a measuring device that converts primary-element response into some sort of indication or recording, (3) the controller that converts primary-element position into power impulses, which operate the (4) final control element that varies the flow to meet primary-element detections. In some controllers, the first three elements are not always separate and distinct.

A liquid-level controller, for instance, might use a float that connects directly to the supply valve. Here the float detects the level (element 1) and by its vertical position indicates this level (element 2). At the same time, it acts as element 3 to position the supply valve.

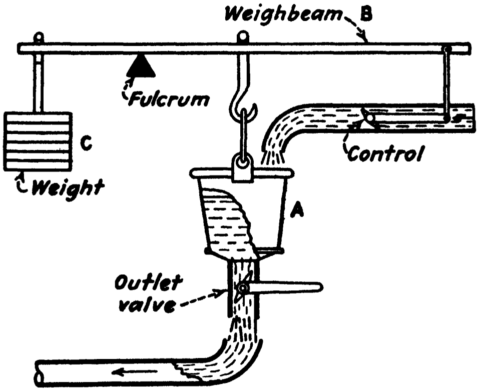


FIG. 9-1.—Weigh beam operates control valve to maintain constant level in bucket. (*Courtesy of Republic Flow Meters Co.*)

Control systems can be broadly classified according to the medium that furnishes power for operation as (1) hydraulic, (2) electric, (3) pneumatic, or (4) combined electric-pneumatic.

Hydraulic Control Systems.—To understand the operating principle better, study the simple hydraulic system in Fig. 9-1. The problem here is to keep a constant water level in the bucket. The weight of water in bucket *A* is balanced on weigh beam *B* by weight *C*. The weigh beam remains in a balanced position as long as the water weight remains constant; but, if the outlet-valve position changes, the water level varies. Thus the weigh beam moves to adjust the control valve. Starting with this impractical device, let us follow the steps necessary to convert it into an actual regulator.

In Fig. 9–2, the weight is replaced by a spring and the bucket by a diaphragm that supports the static weight of the liquid head. The action is the same as before except that the diaphragm pressure is balanced against the spring tension. When the pressure on the diaphragm increases or decreases, it moves the supply valve to restore the pressure to normal. The weigh beam is again balanced. The sensitivity of the weigh beam is low because it must move the heavy valve. Furthermore, for every supply-valve position, there is a different weigh-beam position and, consequently, a different tension on the spring. Therefore, the regulator does not control at an exactly constant pressure or level.

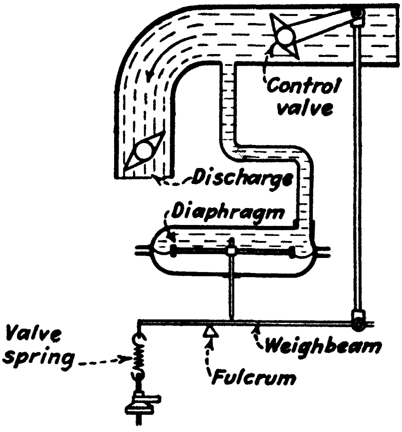


Fig. 9–2.—Diaphragm opposed by a spring replaces the bucket and weight.

Figure 9–3 shows a mechanism that amplifies the weigh-beam force without affecting its sensitivity. The amplifier is a differential piston operating in a cylinder supplied with oil or air under pressure. From the inlet, the operating fluid goes to a chamber above the piston and through a restricting orifice to a chamber under the piston. The latter chamber drains through a pilot valve. The only work imposed upon the weigh beam is that necessary to operate the pilot valve. With the latter throttled, pressure builds up under the piston and moves it upward. Free flow through the pilot valve reverses the operation; and, when the weigh beam is balanced (always in a horizontal position), the pilot valve is balanced, and the amplifying piston remains stationary. This regulator is practical only for systems having no time lag (interval between the main-valve movement and the change in the controlled medium).

In Fig. 9–4, a stabilizing device takes care of time lag by bal-

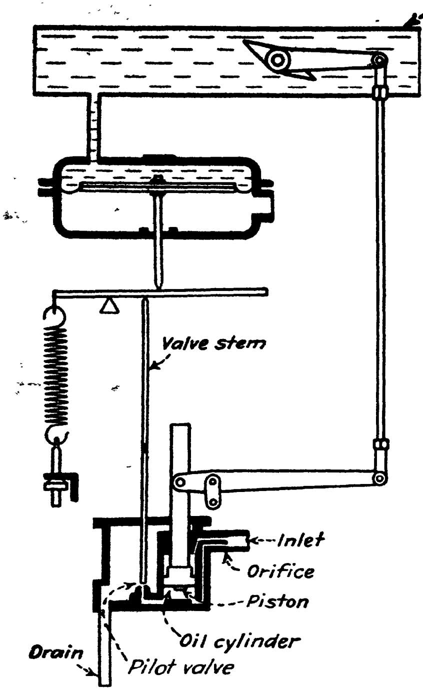


FIG. 9-3.—Weigh beam regulates oil pressure acting on an amplifier piston.

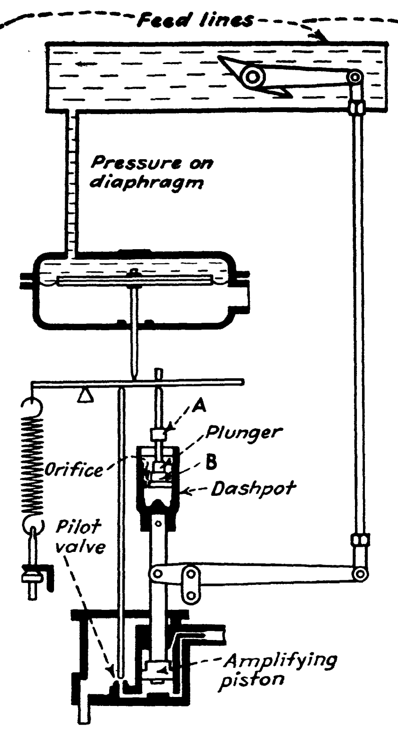


FIG. 9-4.—Oil dashpot on amplifier piston anticipates pressure's return to normal.

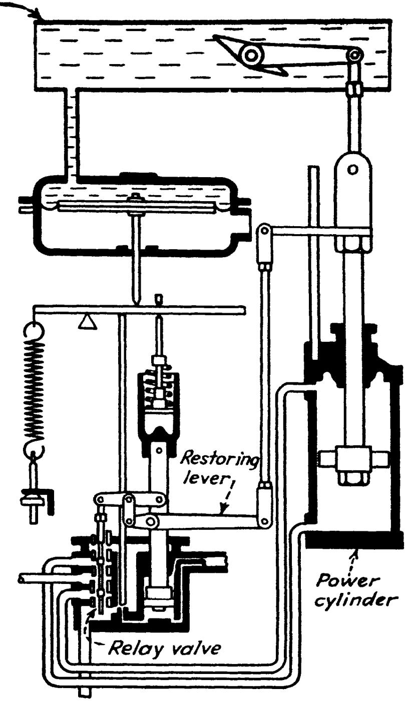


FIG. 9-5.—Adding a relay valve and power piston takes care of large-sized valves.
(*Courtesy of Republic Flow Meters Co.*)

ancing the weigh beam prematurely to prevent hunting. The device that accomplishes this is an oil-filled dashpot built into the amplifier piston. The dashpot plunger stem connects to the weigh beam, and the plunger is free to slide on the stem in opposition to a Picard spring anchored at *A* and *B*. The movement of the amplifier piston creates a differential pressure across the dashpot plunger and against the Picard spring (one that can be put under either tension or compression) that momentarily rebalances the weigh beam in anticipation of the return of the controlled pressure to normal. Differential pressure is gradually relieved by oil leakage through an orifice in the dashpot plunger, and its force acting on the weigh beam decreases; thus the dashpot prevents regulator hunting. To make the regulator applicable for any size of supply valve, a secondary, or relay, valve and power cylinder must be added.

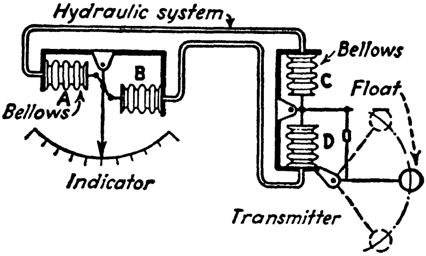


FIG. 9-6.—Sealed hydraulic system arranged to indicate liquid level. (*Courtesy of The Liquidometer Corp.*)

In Fig. 9-5, the amplifier piston, instead of moving the supply valve directly, now operates a relay valve. As the latter moves, it uncovers both power-cylinder ports; oil under pressure then flows to one side of the power piston and drains from the other side. Full oil pressure acting on one side of the piston makes it powerful enough to move heavy valves. Relay-valve ports are so located that, when the valve stem is in mid-stroke, both power-cylinder ports are closed, and the power piston is held stationary. Obviously, unless the relay valve returns to its mid-position at the instant the power piston reaches a new control position, the piston would continue and make a full stroke. A restoring lever that connects to the amplifier- and power-piston rods controls the relay-valve movement. The amplifier piston moves the relay valve from mid-position, whereas the power piston always moves it toward mid-position.

A balanced hydraulic-operated indicator (Fig. 9–6) uses two pairs of bellows fixed at their outer ends. Bellows *A* and *B* are linked together as are bellows *C* and *D*. Bellows *A* and *C* are connected by tubing, as are bellows *B* and *D*. The two circuits are filled with liquid. When the float moves down, tank-end bellows *D* compresses and displaces liquid, causing dial-end bellows *B* to expand. Simultaneously, tank-end bellows *C* expands, taking in liquid, causing dial-end bellows *A* to compress. When the float moves up, the action is reversed.

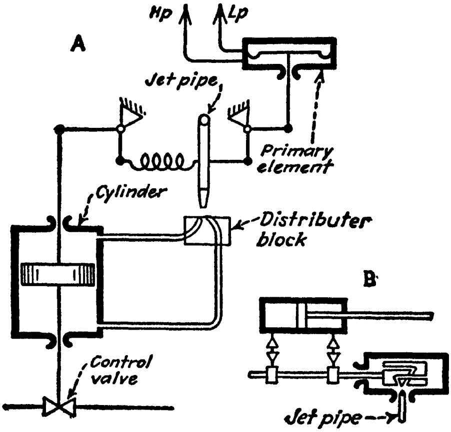


FIG. 9–7.—Diaphragm moves oil jet between two orifices leading to power cylinder. (*Courtesy of Askania Regulator Co.*)

The hydraulic system (Fig. 9–7) uses a jet-pipe relay consisting of a nozzle that swings freely on a pivot. Oil discharge is directed at two closely adjacent receiving orifices in the distributor block that connect directly to the power cylinder. The total movement of the jet pipe is less than one mechanical degree.

With the jet pipe in mid-position, equal pressures exist in each receiving orifice, and consequently pressure on each side of the power piston is equal. When the slightest movement of the jet pipe occurs, one orifice receives more oil, thereby creating a pressure differential across the piston. The piston then moves at a rate proportional to the jet-pipe deflection. An oil catcher maintains a small quantity of oil around the distributor block to prevent foaming.

The jet-pipe movement is controlled by a primary element and opposing spring that position it to move the valve-operating power piston *A*. On applications requiring large power pistons, the jet oil supply and pressure are not adequate. Here an auxiliary piston moves a relay valve to control the power piston. The jet then directs oil into two orifices in the relay-piston periphery that lead to opposite ends of the piston. The jet-pipe movement to the left forces the relay piston in the same direction until both orifices are again equidistant from the jet *B*.

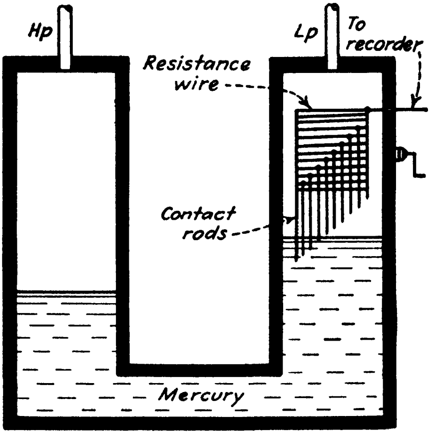


FIG. 9–8.—Mercury rise and fall varies resistance in external metering circuit.

Electric Controllers.—Electrical systems for indication and control include series resistance, resistance balance, reactance balance, self-synchronizing motors, periodic-impulse relays, and on-off switch operation.

The series-resistance unit in Fig. 3–25 operates a remote indicator by varying the resistance in the transmitting circuit. Static pressure inside the chamber acts against the bellows and forces the push rod up to move the wiper arm across the resistance commutator. Each copper disk in the commutator connects to a section of fixed resistance. The movement of the wiper arm thus varies the amount of resistance in the circuit in relation to the liquid level or pressure. Attaching a mercoid switch to the wiper arm makes the unit an on-and-off controller. A suppressed zero body can be used to measure the level above a fixed point in an overhead tank.

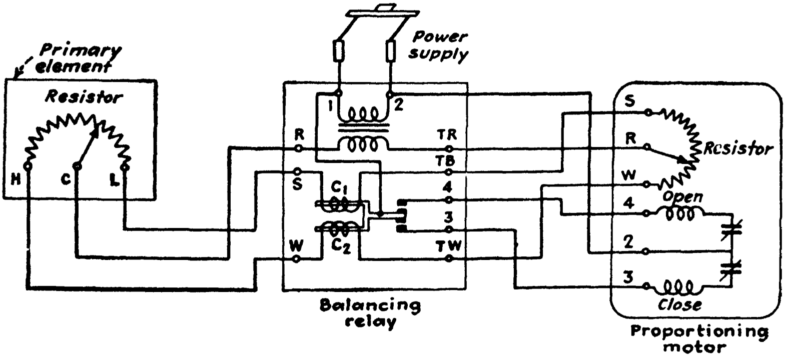


FIG. 9-9.—Two rheostats connected into a bridge circuit use a sensitive relay that detects unbalance in the bridge to control a reversing motor-operated valve. (*Courtesy of Brown Instrument Co.*)

The mercury-manometer transmitter (Fig. 9-8) contains a graduated resistance in the low-pressure chamber. The resistance wire is wound in a coil with a series of vertical rods that contact the rising mercury; or the entire resistance can be contained in concentric rings of individually laid rods, cut so that their lower ends form a true spiral contact. When the mercury makes or breaks contact with each rod, the current flow in the transmission circuit varies to correspond with the transmitter mercury position. The receiving instrument is similar to a simple wattmeter.

The resistance balance (potentiometer) system (Fig. 9-9) uses two rheostats connected into a bridge circuit with a sensitive relay that detects unbalance in the bridge. The relay controls a reversing motor that, as it operates the flow valve, restores the bridge to balance. The primary element operates a left-hand rheostat. The potentiometer circuit is universally used when precise and dependable temperature readings are required.

In another resistance balance unit (Fig. 9–10), a float and counterweight are used to move the transmitter slide wire shown in Fig. 9–11. Changes in the transmitter slide-wire resistance occur with changes in float position. Measurement is made by the electrical-balance method that employs a Wheatstone-bridge circuit to determine the transmitter slide-wire resistance in terms of known standard resistors. The balance-point detector is a galvanometer through which no current flows when the resistance of the transmitter and standard resistors balance. The receiver can be used with a pneumatic system for control purposes.

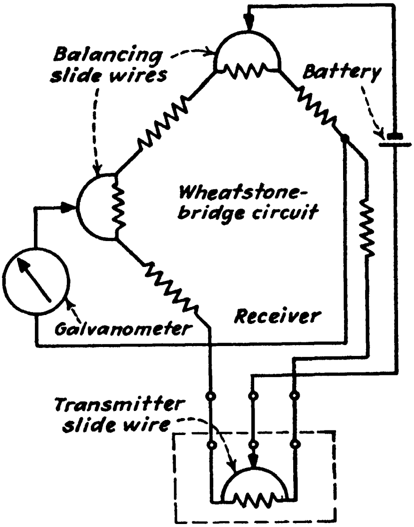


FIG. 9–10.—Electrical balance instrument employs Wheatstone-bridge circuit. (*Courtesy of Leeds & Northrup Co.*)

In one inductance bridge, two coils *A* and *B* are connected into an a-c circuit (Fig. 9–12). If an iron core is centered in each coil, current flows only in the two outside wires, because the two parallel circuits are then balanced. Moving the left-hand core above the mid-point of coil *A* (Fig. 9–13) increases inductance in the top half of the coil and reduces it in the lower half. Consequently, more current flows in the lower half than in the top. As a result, part of the current from the bottom half of coil *A* flows through the center wire to the top half of coil *B*.

The top half of coil *B* now has more current flowing in it than the bottom half and exerts a greater pull on the iron core. This lifts the core into the same position as that in coil *A*. After this occurs, the circuit is again in balance, and current flows in the two outside wires only. From the foregoing, it is evident that, as the core in coil *A* moves up or down, the core in coil *B* will take a similar position.

Figure 9–14 is another inductance-bridge arrangement with power connections made to the center of two coils *A* and *B* instead of to the ends. Moving the core up in coil *A* increases the inductance in its top half and reduces the inductance in its lower half; consequently, more current flows in the bottom half of both coils *A* and *B* than in the top half.

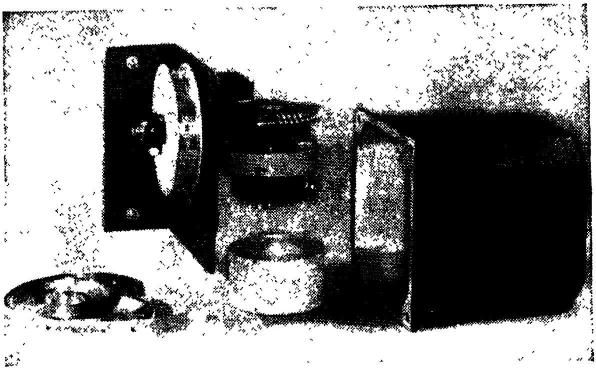


FIG. 9–11.—Transmitting slide wire and float tape used with the circuit of Fig. 9–10. (*Courtesy of Leeds & Northrup Co.*)

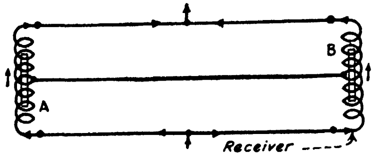


FIG. 9–12.—Moving core *A* from center unbalances inductances of the two halves of its coil.

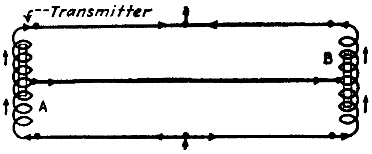


FIG. 9–13.—Current then flows through center wire to move right-hand core *B*.

Pull on the core in coil *B* is not depended upon to position the indicator or recorder. This is done by a continuously runing synchronous motor through a mechanism actuated by the sensitive galvanometer boom. Varying the level in the mercury chamber changes the position of the core in coil *A* and causes current to flow through the galvanometer circuit. Deflection of the galvanometer pointer operates a mechanism to position the receiving instrument. When these have been adjusted accurately, the core in coil *B*, connected to the receiver, will be in a position to balance the bridge, and the galvanometer boom will be at zero.

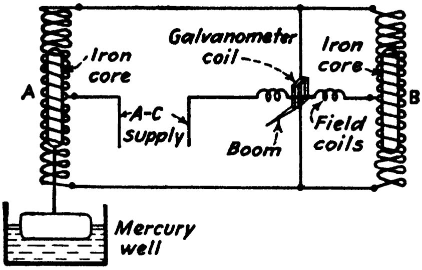


FIG. 9–14.—Galvanometer connected across bridge circuit controls a motor-operated indicator.

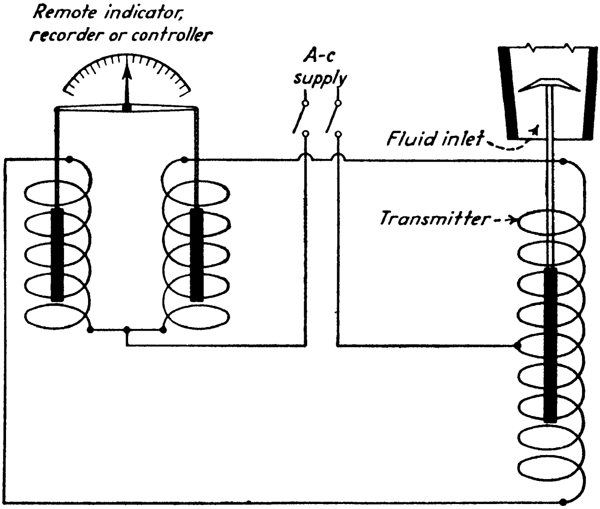


FIG. 9–15.—Inductance bridge uses a single transmitting coil and two individual receiving coils. (*Courtesy of Fischer & Porter Co.*)

The inductance bridge in Fig. 9–15 uses a single transmitter coil, but the receiver contains two separate coils mounted parallel with each other. Each of the latter has an actuating armature hanging from the centrally pivoted beam. Receiver armatures seek to maintain a position respective to their coils identical to that assumed by the transmitter armature inside its coil.

The system in Fig. 9–16 utilizes self-synchronizing motors that, when interconnected, operate so that one motor reproduces any motion imparted to the other. The three transmitter stator leads connect to the three receiver stator leads, and both rotor leads connect in parallel to the same a-c source. Connecting a-c power to the motors when the rotors are out of synchronism unbalances the stator voltages, and current flows in these windings. Thus a torque is produced in both rotors that tends to keep them in phase. Since the transmitter rotor is held mechanically by the measuring mechanism, the receiver rotor, being free to move, takes a position in synchronism with the transmitter.

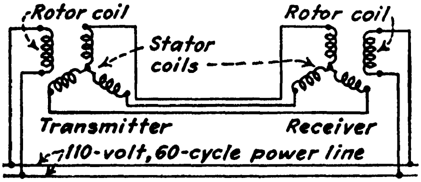


FIG. 9–16.—Self-synchronizing motors stay locked in step and move in unison.

The transmitter (Fig. 9–17) consists essentially of a mechanically operated measuring device; two contact arms, swinging about points *M* and *C*, and a constant-speed cam arrangement to swing the lower contact arm in a continuous reciprocating cycle. The upper arm is positioned by the measuring device through mechanical stop *HT*. The lower roller arms in both transmitter and receiver rise and fall with the rotating cams driven by telechron motors. An induction motor provided with shading coils *A* and *B* positions the receiver. Short-circuiting one coil produces rotation in one direction, whereas short-circuiting the other coil causes rotation in the opposite direction. If both coils are short-circuited or open-circuited simultaneously, no rotation results.

Contacts *L* in the transmitter short-circuit shading coil *A* when the mercury switch is closed, whereas contacts *O* in the receiver short-circuit coil *B*. If contacts *L* and *O* close simultaneously, no rotation of the reversing motor takes place. This means that the transmitter and receiver are in exact relation to each other.

If the level decreases (liquid-level control), contacts *L* will be closer together than contacts *O* and will close first, short-circuiting coil *A*, causing clockwise rotation of the reversing induction motor until the receiver reads the same as the transmitter. Likewise, if the level increases, the contacts *L* will be farther apart and contacts *O* will close first, short-circuiting coil *B*, causing rotation of the receiver motor in a counterclockwise direction to show a correspondingly higher level reading.

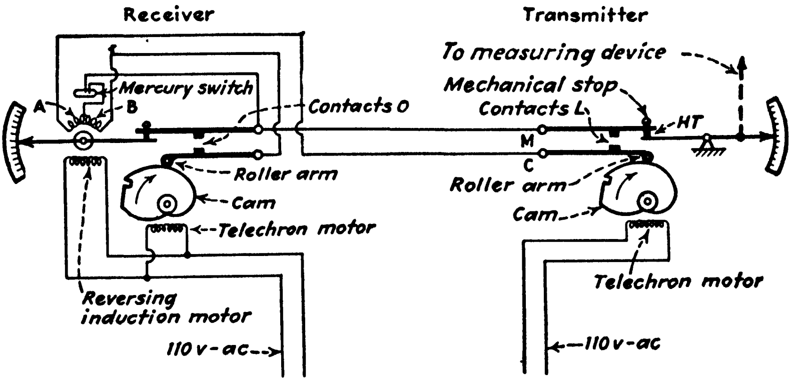


FIG. 9–17.—Two constant-speed cams close electrical contacts for a time period varying with the position of stop *HT*. (*Courtesy of Bailey Meter Co.*)

The mercury switch shown in the common line and installed in the receiver unit is closed during the metering operation but is open during the rest of the cycle. Opening of the mercury switch takes place before contacts *L* and *O* open on the down strokes of the roller arms, thereby preventing arcing or burning at these contacts. The complete circuit includes a phasing circuit as well as the metering circuit just described. The phasing circuit gives the cams the same angular position after a power interruption to either or both motors or when energizing the meters.

The mechanism in Fig. 9–18 consists of a constantly rotating spiral cam *A*, rocker plate *B*, and a magnet-operated mercury switch *C*. The pointer of the measuring element swings between the rocker plate and the spiral cam in such a way that, when the cam engages the end of the pointer, the rocker plate moves forward about axis *D* through a limited angle, thus actuating the magnet switch. A telechron motor rotates the cam continuously at a constant speed of one revolution in 15 sec so that it always engages the pointer at the same instant during each cycle. However, because of the shape of the cam's trailing edge, the total length of time that the pointer is engaged depends on its deflection from zero position. The duration of each impulse is proportional to the measured quantity, because the time the pointer is on the cam varies with the pointer position.

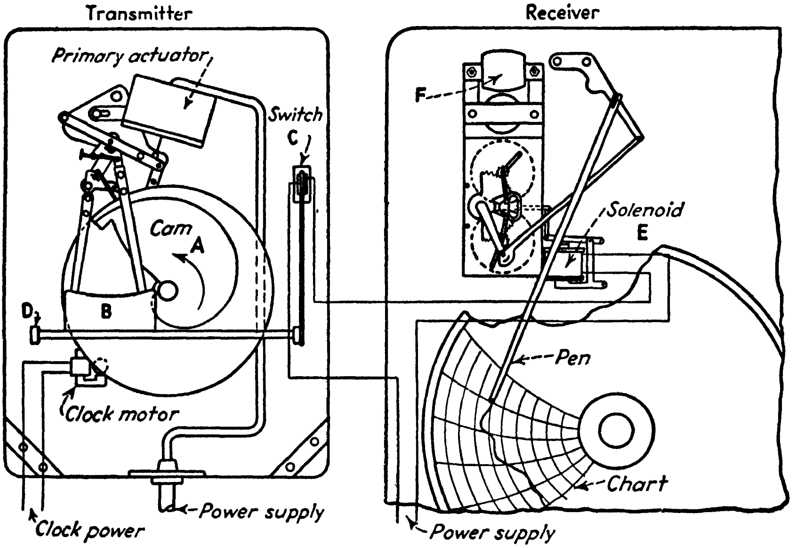


FIG. 9-18.—Constant-speed cam together with rocker plate initiates electrical impulses that position the receiver. (*Courtesy of The Bristol Co.*)

Since the mercury switch is actuated by the magnet and closes when the rider is off the cam, and since the cam rotates at a constant speed of one revolution in 15 sec, it follows that the total time that the rider is on and off the cam for each cycle is always 15 sec, regardless of the position of the pointer. The time that the rider is off the cam changes, however, as the pointer and rider are moved along the edge of the rocker plate by the measuring element. This governs the duration of each impulse transmitted to the receiver, making it proportional to the measured quantity.

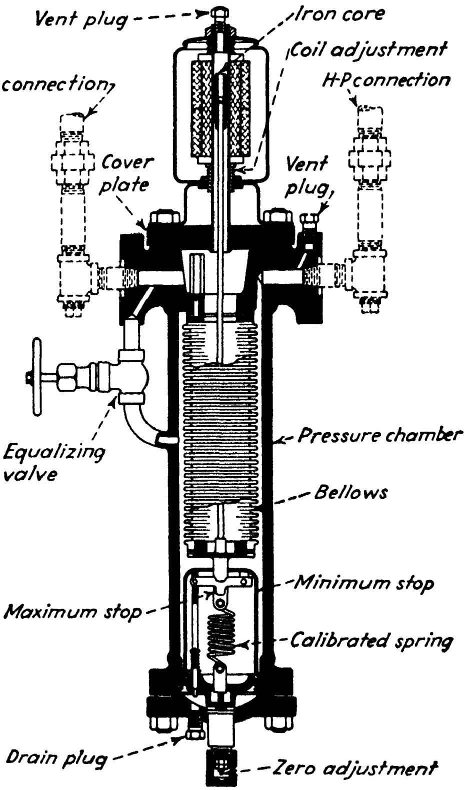


FIG. 9–19.—Differential-pressure unit uses a transformer coil to unbalance an electric circuit in an electronic amplifier. (*Courtesy of Bailey Meter Co.*)

The receiving instrument, through its solenoid *E*, functions to record the readings as measured by the transmitter and sent to it in the form of impulses over the two-wire transmission circuit. In it is mounted a recording chart operated by a telechron motor *F*. The pen is positioned on the chart in accord with the deflection of the pointer in the transmitter.

The differential-pressure transmitter in Fig. 9–19 consists of a pressure housing that contains a stainless-steel or bronze bellows and a calibrated balancing spring. The motion of the free end of the bellows is transmitted through a solid link to an iron core that moves inside a stainless-steel tube surrounded by a set of transformer coils. These consist of a single primary and two secondary windings.

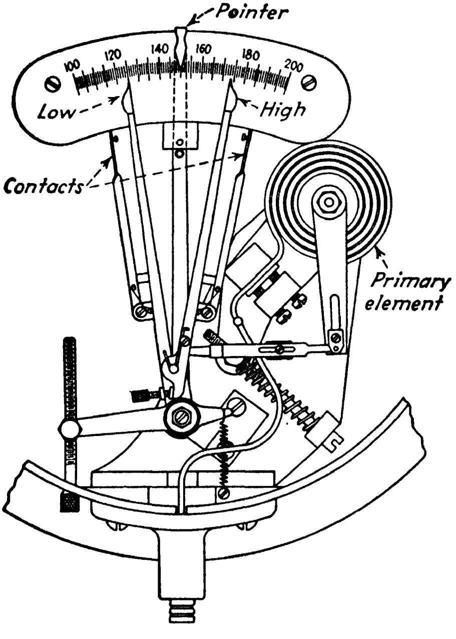


FIG. 9–20.—Typical mechanical thermometer equipped with control or alarm contacts. (*Courtesy of C. J. Tagliabue Mfg. Co.*)

The motion of the iron core varies the ratio of the voltages across the two output windings of the secondary. This variable voltage ratio is used to operate a receiver through an electronic amplifier.

Figure 9–20 illustrates a conventional mercury, vapor-tension, or gas-filled thermometer equipped with electrical contacts to control electric ovens, refrigerating units, etc., and to operate signals.

The pyrometer in Fig. 9–21 uses a potentiometer consisting of slide wire *A* and contact *B*. The voltage of the temperature-measuring thermocouple is automatically balanced by changing the position of contact *B* until the drop across the slide wire between points *O* and *B* is equal to the emf corresponding to the temperature of the thermocouple. Slide wire *A* is part of a bridge circuit consisting of resistances *C*, *D*, *E*, and *F*, the current through which is supplied by a dry battery. The slide-wire contact is operated by relays *G* and *H*.

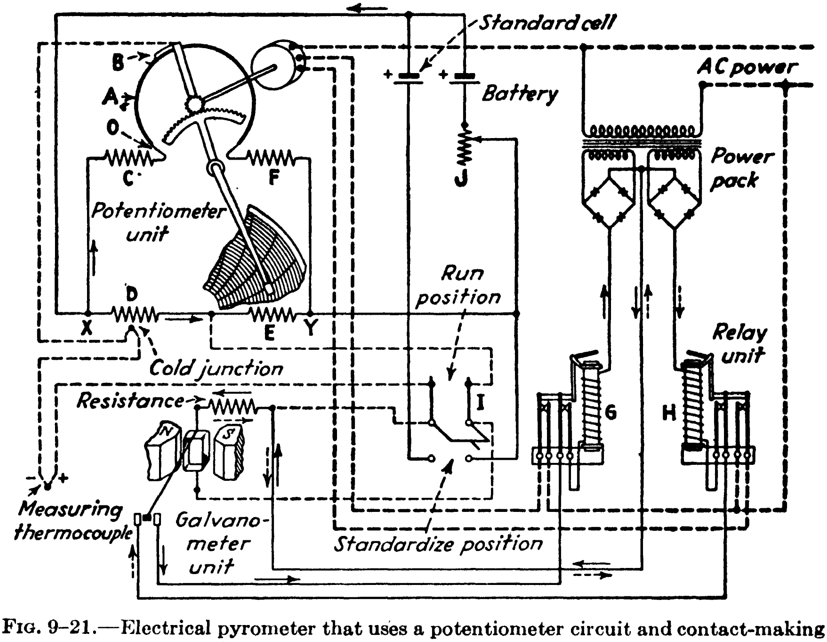


FIG. 9-21.—Electrical pyrometer that uses a potentiometer circuit and contact-making galvanometer. (*Courtesy of The Bristol Co.*)

Proper adjustment of the bridge current from the dry battery is obtained by closing switch *I* in the downward position to cut the galvanometer in series with the standard cell. Rheostat *J* is then adjusted until the galvanometer pointer is in the null or neutral position. This applies a voltage to points *X* and *Y* equal to that developed by the standard cell.

The cold junction of the thermocouple is located adjacent to and assumes the same temperature as resistance *D*. This resistance is designed with the proper temperature coefficient of resistivity to cause a change in the drop across its extremities equal, but of opposite polarity, to any change in the emf of the cold end of the couple, caused by temperature variations.

Pneumatic Controllers.—Pneumatic systems convert measuring-element positions into a proportionate air pressure and transmit it to receiving instruments or control valves through noncorrosive tubing. Installations for measuring purposes have been made up to 4,000 ft between the transmitter and receiver. In these instruments, the primary element moves a small pilot valve or flapper and nozzle mechanism to regulate the air flow.

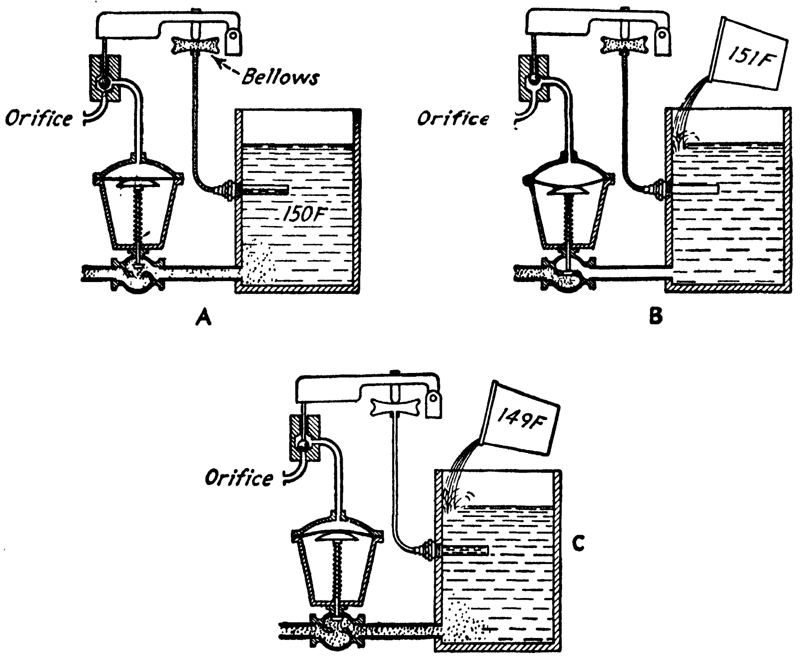


FIG. 9–22.—Pneumatic controller is at equilibrium in *A*. At *B*, temperature is too high and steam is shut off. Low temperature at *C* opens steam valve.

To understand how a simple pneumatic system operates, consider Fig. 9–22*A*, which shows a controller set to maintain the liquid temperature at 150 F. The system is at equilibrium, the liquid is at the correct temperature, and the bellows is expanded just enough to hold the air valve in an intermediate position. With this setting, sufficient air pressure is applied to the supply-valve diaphragm to hold it in a position for correct steam flow. Excess air leaks out around the valve stem.

In Fig. 9–22*B*, conditions have been upset by adding heat from an outside source. The temperature element senses the change and expands to move the air valve against its upper seat. This shuts off all air leakage and opens the inlet wide so that full air pressure is applied to the valve diaphragm. Closing tightly, it shuts off the steam.

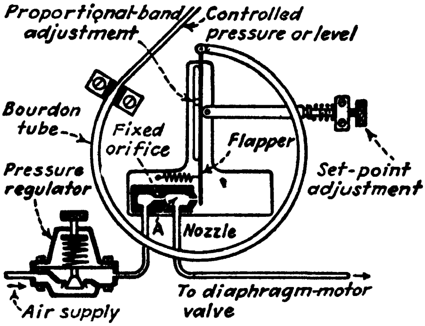


FIG. 9–23.—Pneumatic pilot flapper throttles nozzle discharge to adjust control pressure. (*Courtesy of Fisher Governor Co.*)

Suppose the normal condition was upset in the opposite direction (Fig. 9–22*C*) by introducing a cold substance. The action of the thermometer is to contract its bellows and move the air valve to its lower seat. Thus air to the supply-valve diaphragm is completely shut off, and the diaphragm chamber is vented through the annular space around the valve stem. The steam valve then opens wide.

In practical controllers, the air supply passes through a restricting orifice before reaching the throttling valve or nozzle. Figure 9–23 shows a typical pneumatic-controller arrangement. Air from a compressed-air source passes through a pressure-reducing valve set to maintain either 15 or 20 psi on the low-pressure side, depending on the type of controller. This reduced-pressure air then flows into the orifice chamber, through the orifice, and out the nozzle or to the diaphragm-motor valve.

With a constant pressure applied to the left side of the orifice, the pressure at *A* and on the valve diaphragm can be varied from zero to maximum by moving the flapper toward the nozzle. Because the fixed-orifice discharge is constant, the pressure in chamber *A* depends on the amount of air released out the nozzle.

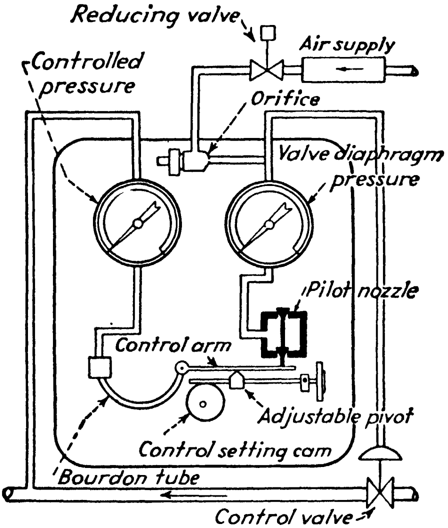


Fig. 9–24.—Simple pneumatic controller diagram illustrates sensitivity-adjusting pivot. (*Courtesy of Mason-Neilan Regulator Co.*)

Figure 9–24 illustrates a simple pneumatic controller complete with the diaphragm-operated control valve. Air at reduced pressure flows through the orifice on its way to the pilot nozzle and valve diaphragm. The bourdon-tube element moves the control arm to operate the pilot-nozzle valve. Because the free end of the bourdon tube always moves the same amount for a given internal-pressure change, the distance the pilot nozzle moves per increment of control-arm movement can be varied by sliding the adjustable pivot (fulcrum) to the left or right. For example, with the pivot at the extreme left, a slight movement of the bourdon tube moves the pilot nozzle from one extreme position to another and makes it an on-off controller. With the pivot at the right, conditions are reversed, and the free end of the bourdon tube must travel over a considerable range to move the pilot nozzle through its full travel. Thus the pressure applied to the valve diaphragm is more or less in proportion to the amount of controlled medium variation, and the valve is positioned accordingly. Figure 9–25 shows simple diagrams of two other controllers.

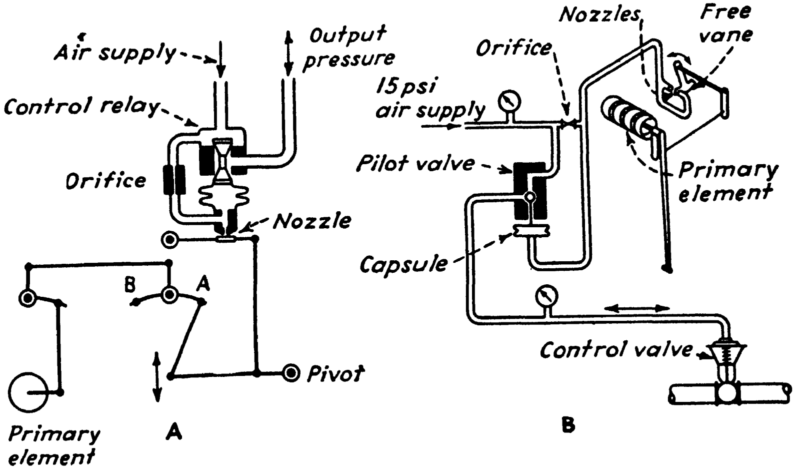


FIG. 9–25.—Both these on-off instruments utilize a bellows-actuated secondary valve to regulate air output. [*Courtesy of (A) The Foxboro Co., and (B) The Bristol Co.*]

Long piping between the primary element and the valve diaphragm (including the diaphragm chamber itself) has considerable capacity that introduces a time lag between forward and reverse action. To eliminate this effect, control systems include a booster valve (*A* in Fig. 9–26), which acts as a multiplying relay to fill and empty the line quickly.

A neoprene-faced spring-actuated flap controls both the supply port *B* and the exhaust port *C*. The nozzle pressure acts on the large bellows *D*, which operates the flap through tube *E*. Transmitted pressure from supply valve *B* acts on the inner side of small bellows *F* as well as on the valve diaphragm. Increased nozzle pressure moves the large bellows and the exhaust tube *E* down to force the flap away from port *B*. Decreased nozzle pressure permits the large bellows and exhaust tube to move up and allow the flap to close port *B*. When continued upward movement of the bellows moves the exhaust tube away from the flap, entrapped air in the diaphragm chamber flows up exhaust tube *E* and out between the two bellows to the atmosphere. When the transmitted pressure is correct, the pilot valve balances and the flap closes ports *B* and *C*.

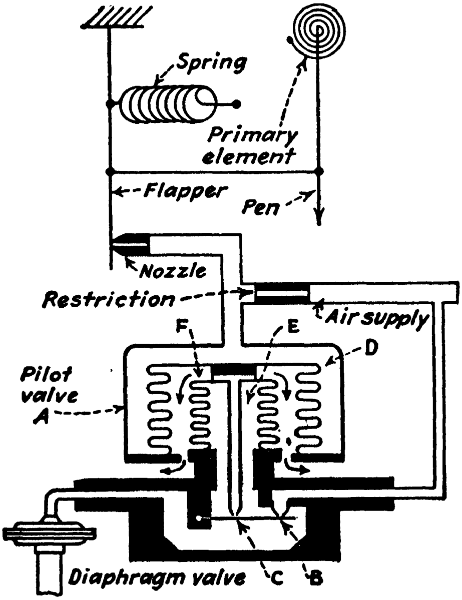


FIG. 9-26.—A multiplying relay or secondary valve fills and empties air-control lines quickly. (*Courtesy of Brown Instrument Co.*)

In the controller (Fig. 9-27), rising temperature moves the flapper toward the nozzle, obstructing the air bleed and building up pressure on the control-unit-assembly bellows. The downward movement of the bellows opens its valve and increases the air pressure acting on the main valve diaphragm. Conversely, when the temperature falls, the reverse action takes place. The diagram shows a direct-acting instrument—it delivers air as temperature rises. The unit can be changed to reverse acting—delivering air on falling temperature—by disconnecting link *C* and connecting it to the other end of crank *D*. Turning the adjusting knob moves

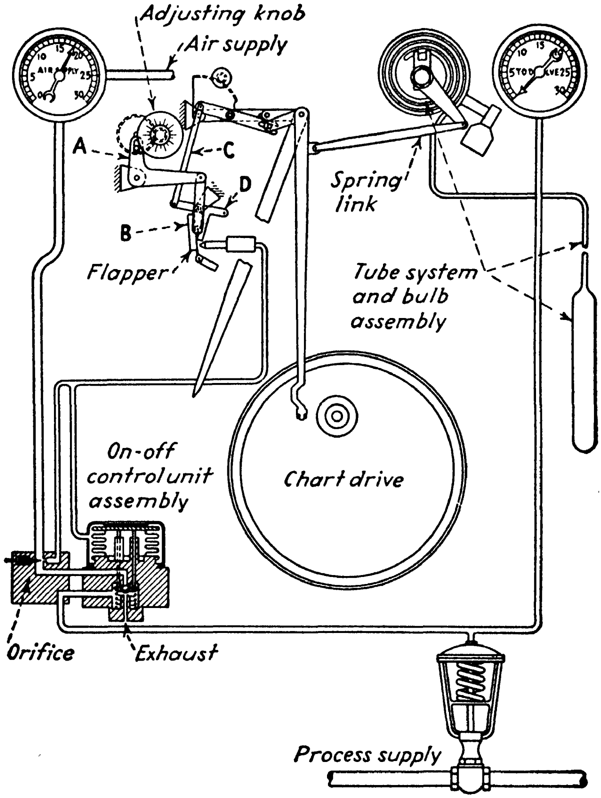


FIG. 9–27.—Instrument can be changed from direct to reverse acting by changing link *C* to point *D*. (*Courtesy of C. J. Tagliabue Mfg. Co.*)

roller *B* (located between the flapper and crank *D*) to increase or decrease the instrument sensitivity.

In air-conditioning work, it is sometimes advantageous to install a master thermostat that readjusts the various air-duct thermostats according to changes in outside temperature. In a pneumatic system, the master thermostat, instead of controlling the system directly, applies air pressure at *A* to an adjusting bellows in each of the submaster thermostats (Fig. 9–28). Expansion or contraction of this bellows resets its control valve for the new outside temperature, and the air-conditioning equipment operates according to the new setting.

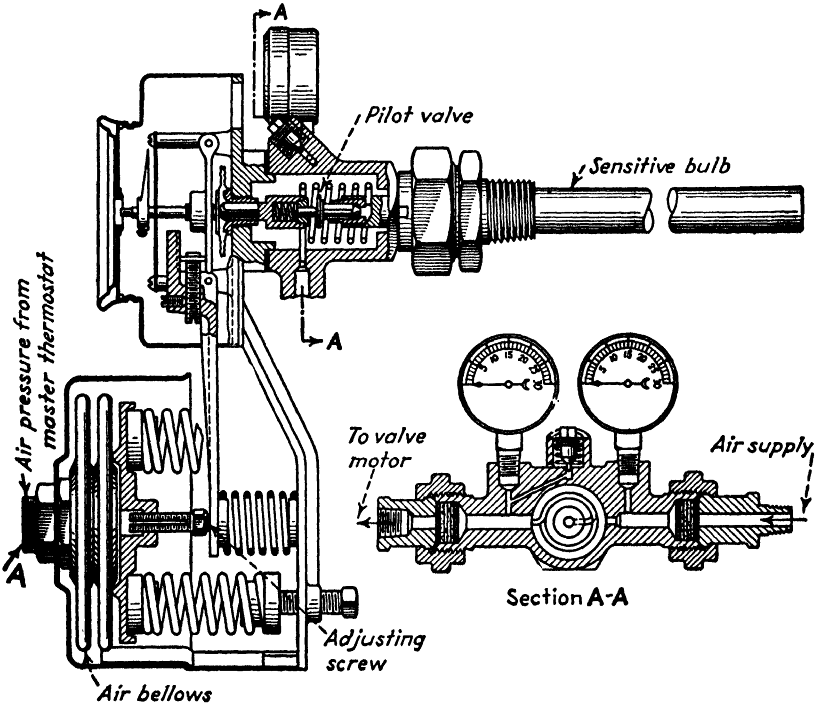


FIG. 9–28.—A pneumatic master thermostat readjusts any number of these submaster instruments, which in turn control local temperatures. (*Courtesy of The Powers Regulator Co.*)

In many processes, the air-pressure changes must be limited to small increments in order to give the control valve similar small movements. Because the nozzle-flapper normally moves only 0.002 in. to change the system air pressure from maximum to minimum, it is impracticable to get a small fraction of this amount by merely reducing flapper movement. The left-hand bellows (*A* in Fig. 9–29) gives these small increments by moving the flapper pivot.

An increase in air pressure going to the control-valve diaphragm also acts on the outside of bellows *A*. This force is transmitted through the liquid to the inside of the bellows, which moves the flapper away from its nozzle around fulcrum *X* just enough to stabilize the pressure and maintain the valve in its new position. This unit constitutes a simple throttling control in which the controlled medium would have to droop[1] (*offset*) below its control point to give full valve travel on large load changes.

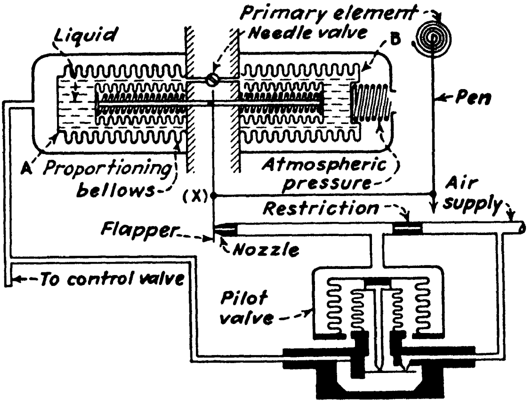


FIG. 9–29.—Pneumatic controller equipped with multiplying valve and proportioning bellows. (*Courtesy of Brown Instrument Co.*)

To hold the pressure, temperature, or whatever is being controlled constant (eliminate *offset*) it is necessary to increase the pressure on the control valve an additional small amount. The automatic reset bellows *B* does this job through the needle-valve connection between its liquid-filled bellows and that in *A*. Both of the small inner bellows, being spring-loaded and connected through the common push rod, will return to their normal position when the liquid pressure in the left-hand unit equalizes with that in the right-hand unit through the adjustable needle valve. As the pressures equalize, the push rod moves to the left, and the flapper begins to cover its nozzle and increase the air pressure enough to bring the medium back to its control (*set point*). Moving fulcrum

---

[1] Standard terms will be given in italics. See Appendix.

X vertically (like the pivot in Fig. 9–24) adjusts the throttling range (*proportional band*). The position of the needle valve determines the rate of return to the set point.

On applications requiring a wide proportional-band adjustment, it is essential that the control valve respond to controller pressure changes that may be so small that they do not provide sufficient power to position the valve. Under such conditions, best results are obtained with a valve positioner (see Chap. V) because valve-stem-packing friction is difficult to control, and turbulence effect inside the valve varies with flow.

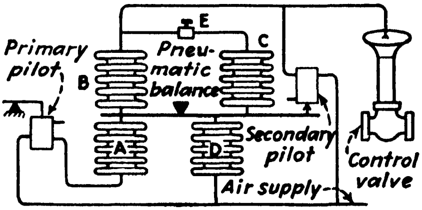


FIG. 9–30.—Electrical balance (Fig. 9–10) controls the primary pilot. From there on, operation is pneumatic. (*Courtesy of Leeds & Northrup Co.*)

Combination Electric-pneumatic Controllers.—The device in Fig. 9–30 forms one part of a combination electric-pneumatic controller, the electric circuit being shown in Fig. 9–10.

Turning as a unit with the measuring slide wire of an electrical-balance detector, a cam acts through levers to move the primary-pilot-valve stem. The pilot-valve outlet pressure corresponds to the actual liquid level, temperature, pressure, etc. Complete control includes a pneumatic balance that corrects automatically for offset. Bellows A is loaded by primary-pilot-valve outlet pressure that, when the variable medium is at the set point, is one-half the supply pressure. Bellows D, though having only one-half the lever arm of A, receives full supply loading pressure.

Bellows B and C have equal lever arms, and each receives loading pressure from the secondary pilot valve when the balance is in equilibrium. Any change in primary-pilot-valve outlet pressure acts on bellows A to move the balance arm and cause a like change in secondary-pilot-valve outlet pressure. This changed secondary pressure acts on the valve diaphragm and also on bellows *B* to offset the pressure change in *A* and bring the balance back to equilibrium momentarily.

The pressures in *B* and *C* tend to equalize at once through needle valve *E*. This causes bellows *C* to move the balance arm and valve

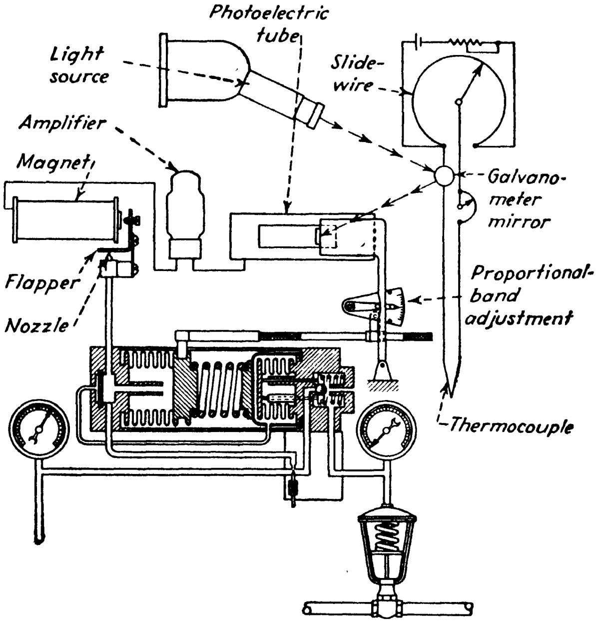


FIG. 9-31.—Combination electrical and pneumatic pyrometer uses a photoelectric tube and electronic amplifier. (*Courtesy of C. J. Tagliabue Mfg. Co.*)

stem still farther in the same direction. Thus a further change in secondary-pilot-valve outlet pressure goes both to the valve diaphragm and bellows *B*. The balance thus assumes equilibrium at a new arm position with pressures in *B* and *C* equal and that in *A* again one-half the pressure in *D*. The secondary pilot valve holds a new position corresponding to the new load, and the variable medium is again at the set point.

The combination electric and pneumatic pyrometer in Fig. 9-31 uses a light beam and photoelectric cell to operate a magnetic air valve. From this point on, the operation is all pneumatic. The action of the flapper controls the air pressure acting inside the left bellows chamber. The movement of the bellows to right or left operates the auxiliary valve to adjust the air pressure applied to the main valve diaphragm. The controllers discussed so far are known as on-off (*two-position*) instruments recommended where process time lag is small and supply capacity large. Chapter X deals with the more complicated control problems.

CHAPTER X

CONTROL SYSTEMS

Fundamentally all automatic-control problems are somewhat similar. It usually requires only an exchange of sensitive elements to adapt a controller for either pressure, liquid level, or temperature work. Every system has at least two parts—a sensitive (primary) element and a power unit. The first sees or feels something, and the second says or does something about it. Frequently both are combined in one unit. For example, a metal bellows containing a volatile fluid expands when its temperature rises, and operates the stem of a valve to shut off, or throttle, the steam supply. Although such units are sturdy and dependable, many processes have inherent lags that require devices to provide finer control adjustments.

A thermal (heat-transfer) system offers a better example of system lags for study and explanation than do pressure or liquid-level instruments; because, in temperature control, lags and capacities are more easily visualized. The principles derived here can be extended by analogy to other systems (pressure, liquid level, etc.) where they apply to valves, dampers, or rheostats.

The temperature-control problem in a continuous process is one of matching the rate of heat supply to the rate of heat demand in the face of interfering factors such as storage-capacity, transfer, distance-velocity, and controller lags. Storage capacity, as either fluid volume or heat content, acts as a balance wheel tending to keep the system in equilibrium. The storage capacity can be either the liquid that is being heated or the apparatus through which heat transfer takes place, but the effect of one is different from that of the other.

Stored heat in a large tank of liquid may permit wide variation in rate of throughflow or heating without seriously affecting the liquid temperature. On the other hand, the brickwork of a heat-treating furnace has high heat-storage capacity. If the heat input is higher than the conductivity rate of the metal being treated, the walls absorb heat faster than the metal, and the controlled temperature overshoots. Thus a high demand-side storage-capacity lag helps to maintain a uniform temperature after the process is in operation, but a high supply-side storage-capacity lag makes it difficult to establish proper balance between supply and demand without overshooting.

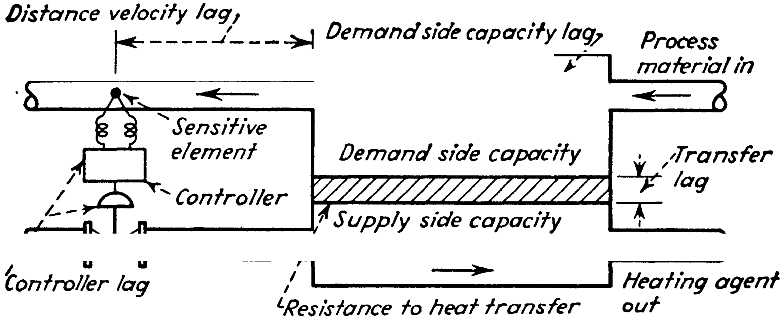


FIG. 10–1.—Simple heat exchanger illustrates various lags encountered in a temperature-control problem.

The simplest way of analyzing a temperature-control application is to set up a simple process as if it were a heat exchanger (Fig. 10–1). This is similar to the method used by E. D. Haigler.[1]

Heat requirements of the material being processed constitute the demand, and the heat available in the heating agent represents the supply. Here numerous factors can upset the desired balance of energy in the process. For example, the pressure of the steam or voltage of the electrical energy used for heating may change. On the demand side, the flow of process material may vary; its entering temperature may change; radiation to surroundings may increase or decrease because of a change in ambient temperature; or a reaction within the process may upset the heat balance.

Furthermore, still other factors can affect the rate of heat transfer without a change in demand. Except where the heating is the

---

[1] *ASME Transactions*, November, 1938, pp. 633–640.

result of direct contact with the hot substance, or with radiation from it, heat must pass through a barrier separating the demand and supply sides. The thermal resistance of this heat-transfer barrier determines the *temperature potential* (difference between the two temperatures) necessary to force a given quantity of heat through a unit area of the barrier in a given time. Hence, changes in the barrier surface, as by scaling or corrosion, can affect the thermal potential needed. In a vertical barrier, changes in liquid level on either side affect the area available for heat transfer.

Thus it must be possible to detect any unbalance between heat demand and supply and then change the thermal potential in such a way that the new required rate of heat transfer is obtained without overshooting. Unfortunately, most processes have inherent lags (Fig. 10–1) that (1) delay the discovery of a disturbance, (2) retard the recognition of its magnitude, and (3) retard the establishment of a new thermal potential. Furthermore, controllers themselves require more or less time to detect changes and make the necessary corrections.

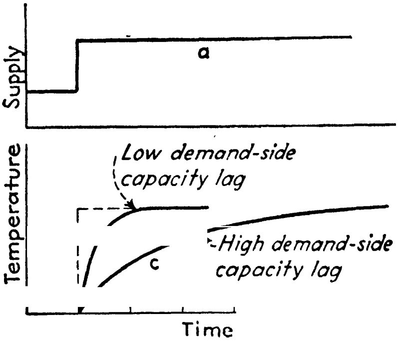


FIG. 10–2.—When heat input to Fig. 10–3 increases, water temperature rises as at *a* or *b*.

Starting with the demand-side capacity lag, which results from the heat storage and consequent *thermal inertia* of the demand side, this factor is usually an advantage rather than a disadvantage. The high demand-side capacity tends to stabilize the process temperature and prevent rapid departures from the set point. It is disadvantageous only when prompt response to a change in set point is desired.

Figure 10–2 shows the situation in an uncontrolled process, with demand-side capacity lag only, when a sudden supply change occurs. Curve *a* shows the change in supply, and *b* and *c* show the resulting change in temperature. A process with low demand-side capacity becomes stabilized quickly at the new temperature as in *b*, whereas one with high demand-side capacity responds slowly as in *c*. At a given rate of outflow, a receiver containing a large volume of process material thus tends to be more stable than one containing a small volume. A jacketed vessel, for example, is more stable than a shell-and-tube heat exchanger, which in turn is more stable than a double concentric-pipe heater.

Actually the rate of response depends upon both volume (energy storage capacity) and the rate of flow of material or energy through the receiver. Although high demand-side capacity is usually a favorable factor in control, this presupposes the ability to attain uniform conditions rapidly throughout the bulk of the material, as by thorough mixing.

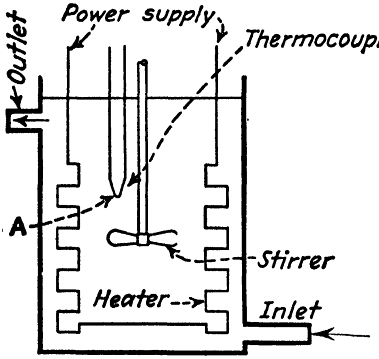


FIG. 10–3.—Large heat volume stored in the demand side permits variation in throughflow or heating.

Figure 10–3 shows the effect of demand-side capacity lag. The electric heater has a large surface and small cross-sectional area, and the stirrer keeps both heater and water at substantially the same temperature. A change in power supply to the heater is evidenced on the thermocouple by a changed water-heating rate. Assume the rate of flow and temperature of inflow fixed, the electrical-heat supply constant, and the temperature at *A* perfectly steady. Now let a sudden change in the heat supply take place. The water temperature does not rise to a new steady value immediately, but follows curve *b* or *c* in Fig. 10–2, the initial rate of increase, depending upon the water's heat capacity. On the other hand, the temperature of outflowing water will not change rapidly because of the heat capacity of the stored water.

Where the thermal inertia of the demand side is ordinarily favorable, the reverse is true of the supply-side capacity and its thermal inertia. The supply-side capacity can be considered the sum of all conditions on the heat-supply side that tend to stabilize the available rate of heat transfer and make a change in transfer difficult or time-consuming.

Any barrier between the supply and demand sides adds thermal capacity as well as thermal resistance and, if its capacity is high, acts as a thermal flywheel that must gain or lose energy before a new desired rate of heat transfer can be achieved. Thus the supply-side capacity and the thermal resistance of a heat-transfer barrier result in what is known as *transfer lag*—the retardation in establishing a new heat-transfer rate following a change in supply potential.

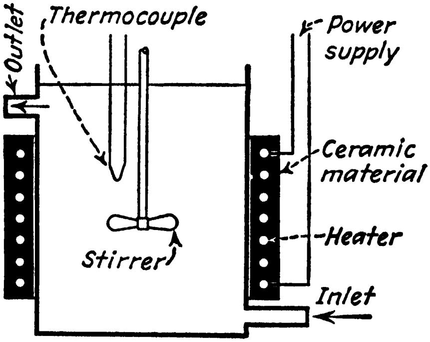


FIG. 10–4.—Thermal resistance between the fluid and heat source causes lag.

When thermal resistance prevents free heat flow from the point of storage to that of measurement, it introduces transfer lag (Fig. 10–4). Here the heater is embedded in ceramic material outside the tank. Assuming that each element has both thermal resistance and thermal capacity, a new rate of heat supply can be established instantly to the heater, but considerable time elapses before it is fully effective within the tank. As a result, a sudden change in supply produces a temperature response, as indicated by the solid line

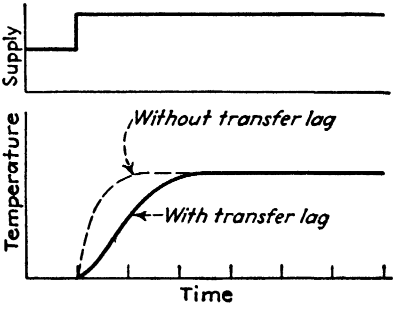


FIG. 10–5.—A sudden increase in heat in Fig. 10–4 raises the temperature along the solid line.

(Fig. 10–5), in contrast to the dotted line obtained when transfer lag is negligible.

Under steady conditions, with heat supply and demand equal, the rate at which heat enters the bath depends only upon the temperature drop across the thermal resistance separating the supply and demand sides and is entirely independent of thermal capacity on the supply side. Increasing the supply-side heat capacity delays the feel of an increased rate of heating, because it requires a longer time to build up the necessary temperature head.

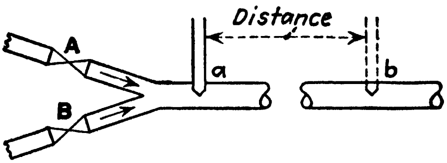


Fig. 10–6.—Distance lag appears if sensitive element is too far from heat source.

The distance-velocity lag is the delay in detecting a change in the measured value because the material must be transported some distance to the sensitive element. Moving the thermocouple (Fig. 10–6) from *a* to *b* introduces a distance-velocity lag. A changed setting of valve *A* or *B* is not evidenced at *b* until the new mixture covers the distance from *a* to *b*.

As previously stated, a change in heat supply, as a result of process lags (demand-side storage, transfer, and distance-velocity lags) produces a temperature response similar to the full-line curve in Fig. 10–5. At first, the temperature does not change; then it increases slowly, following which the rate of change reaches a maximum value (Fig. 10–7). The period when no substantial temperature change occurs is called the *process dead time.* The cumulative effect is such that, even though the sensitive element signals the temperature departure, deviation continues unchecked for a short interval.

Another series of lags referred to collectively as *controller lag* includes the time required for (1) the primary element to achieve equilibrium with the new temperature, (2) the measuring element to become stable at its new position, (3) the control element to detect the deviation and send out the necessary correcting impulse, and (4) the valve or other final control element to reach a position in accord with the controller impulse. Whether these lags have an appreciable effect depends on each particular application.

The controller's job is to measure changes in the controlled variable (temperature or pressure) and make necessary corrections to return the condition to normal. Devices are most generally classified as

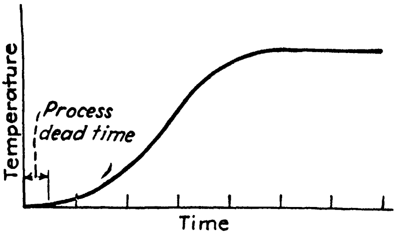


FIG. 10–7.—Cumulative effect of process lags introduces process dead-time period.

1. Two-position.
2. Proportional-position.
3. Floating.
   - *a.* Single-speed.
   - *b.* Two-speed.
   - *c.* Proportional-speed.
4. Proportional-speed floating.
   - *a.* With other compensation.

Two-position Controller.—The two-position controller moves the valve to either of two positions. In Fig. 10–8, curves *a*, *b*, and *c* show demand, temperature, and supply (valve position), respectively, plotted against time. From 0 to time 2, conditions are steady with supply and demand in balance. The valve is periodically shifted between its maximum and minimum positions as the temperature falls to *L* and rises to *H*. The interval between *L* and *H* is the proportional band.

In the cycle *d* to *e*, the valve takes its maximum open position as the falling temperature reaches value *d*. The controlled temperature does not begin to rise immediately but continues to drop to point *f*. It then rises to value *g* (the valve moves to the minimum position) and continues to rise, as a result of energy stored in the heater, to maximum value *h*. At *e*, a new and similar cycle begins.

At time 2, demand suddenly increases. After a transient state extending from time 2 to about 3.5, a new, steady state is reached with supply and demand balanced. At time 6.6, demand reduces to its original value. The result is similar to that obtained when demand increases, with an overshoot in the opposite direction. These curves are typical of two-position control in the presence of a large transfer lag. When demand-side storage lag is the only important retarding effect, the direction of temperature change reverses immediately upon a change in valve position. When the demand is steady, the temperature curve is confined between limits *H* and *L*, and appears sawtoothed.

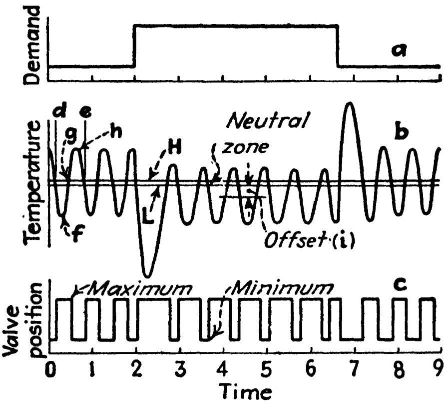


Fig. 10–8.—Droop (*offset*) is the amount the temperature drifts away from the control point on load changes.

Time space 3 to 6.5 (on the temperature curve) illustrates offset. In this area, high and low temperatures no longer average out at the set point as from time 0 to 2, but show a lower value *i*. The magnitude of this offset depends on the system demand, and its direction is opposite to the change in demand. The offset disappears at time 7 when the demand falls off.

Two-position controllers recognize neither rate nor magnitude of load changes. They include electric, pneumatic, and hydraulic devices in which high and low contacts, or pilot-valve positions above and below the set point, open or close a valve or move it to two positions not fully open or closed. Sometimes the controller valve is fitted with a by-pass to supply minimum requirements instead of a stop to prevent the valve from closing tight.

With these controllers, the rate at which the control valve opens or closes is not adjustable; therefore, the rate of supply through them is fixed and the valve cannot be synchronized with the rate at which the system comes to equilibrium.

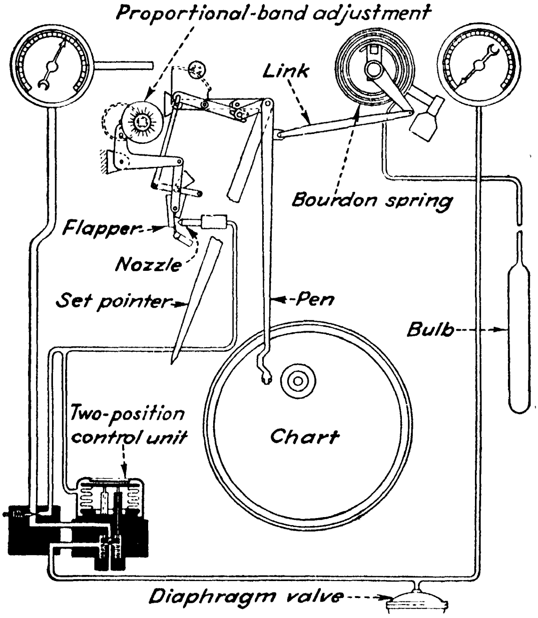


Fig. 10–9.—An on-off controller has no throttling action; controlled valve is always at one or the other extreme position. (*Courtesy of C. J. Tagliabue Mfg. Co.*)

In the pneumatic unit (Fig. 10–9), the pen, in moving toward the set pointer, carries a flapper strip toward the orifice. This obstructs the air discharge and builds up pressure on the control-unit bellows, which then opens its valve and admits pressure to close the main valve. Conversely, when the pen moves below the set pointer, reverse operation occurs, and the main valve opens. The graph in Fig. 10–10 shows how a two-position controller functions. The relation between the high and low points *a* and *b* of the instrument pen remains fixed, but the amount of valve opening can be varied by adjusting the limit stops. Narrowing the valve-opening range tends to straighten out the control line for a particular load; but, if the range is too narrow, it must be readjusted manually for each new load change.

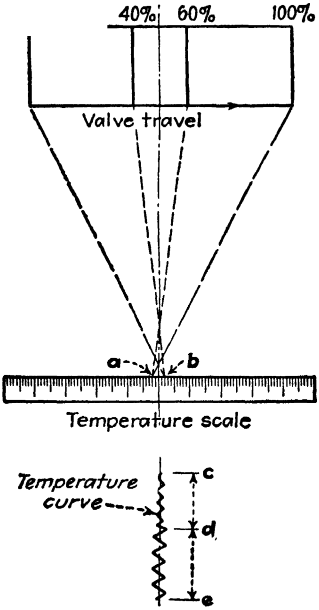


FIG. 10–10.—Control range *a* to *b* can operate valve over any percentage of its travel.

Temperature curve *c* to *d* is obtained when valve travel is limited between 40 and 60 per cent positions; curve *d* to *e* is obtained when the valve moves through its full travel.

Proportional Controllers.—Continuous cycling of the control valve is undesirable in some processes. These require a *proportional-positioning* system (sometimes called proportioning or throttling) that selects a definite and different valve position for every temperature value within its working range. (Figure 10–11 shows a simple flow controller.) To accomplish this, the controller has an adjustable mechanical device for proportioning the amount of valve movement to the temperature, called the *proportional band.* Making the band small (points *a* and *b* in Fig. 10–10) turns the controller into practically a two-position unit; it can be widened so that full-scale pen movement is required to full-stroke the valve. The proportional band is defined in percentage of full-scale movement of the controller pen. For example, if the instrument scale reads from 100 to 200 F and it is adjusted to full-stroke the valve on a 10 F temperature change, the proportional band is 10 per cent.

Consider the graph in Fig. 10–12, which is similar to that in Fig. 10–10 except that the valve has full travel and the distance between high and low range points *a* and *b* is varied. Assume the unit is controlling the temperature of flowing water at 150 F by throttling steam to a heating coil. With the pen at the set point (150 F), the valve is open 50 per cent and passing 10,000 Btu per min to hold this temperature. The flow of water suddenly doubles, which means that twice as much heat must be put into it. To obtain this, the valve must open to 75 per cent position, but to get this opening the pen has to move 2½ F from the set point (drop to 147.5 F).

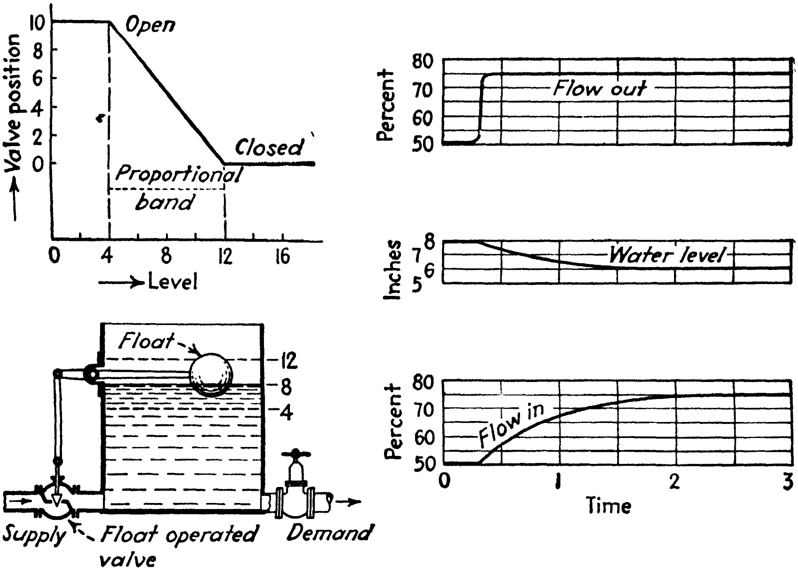


FIG. 10–11.—In a surge tank, proportioning control comes into play as the level falls and increases the inflow to maintain constant outflow. (*Courtesy of Bailey Meter Co.*)

The system then comes to equilibrium at the new temperature. The drop in temperature (2½ F) is the instrument offset on this demand increase. The broader the proportioning band the greater the offset for any given load change.

The temperature did not overshoot, showing that the system absorbed the additional heat at the same rate at which the instrument pen detected the load change. If the pen overshoots the new set point before reaching equilibrium, it indicates that the rate of heat input should be slowed down by widening the proportioning band. This in turn gives greater offset. Generally speaking, a high percentage of unfavorable factors, such as limited storage capacity, long process lags, high thermal potentials, and oversized control valves, requires a broader band.

The diagram in Fig. 10–13 illustrates the operation of a proportioning controller. Assume the system in balance with 9-psi air pressure applied to the control-valve line. The same pressure exists in the proportioning bellows because it feeds from the same line.

If the temperature falls, pen *A* moves reversing bell crank *B* to raise the left-hand end of the proportioning arc lever *C*. This moves flapper *D* nearer nozzle *E* to throttle the free-air discharge and build up pressure in relay bellows *F*. The bellows expands and opens the relay valve to pass more air into the control-valve and proportioning bellows, line *G*. Increased pressure in proportioning bellows *H* causes it to expand and lower arc-lever bearing *I*, which moves flapper *D* away from the nozzle. Spring-loading bellows *H* makes the output air-pressure change strictly proportional to the measuring-pen reflection.

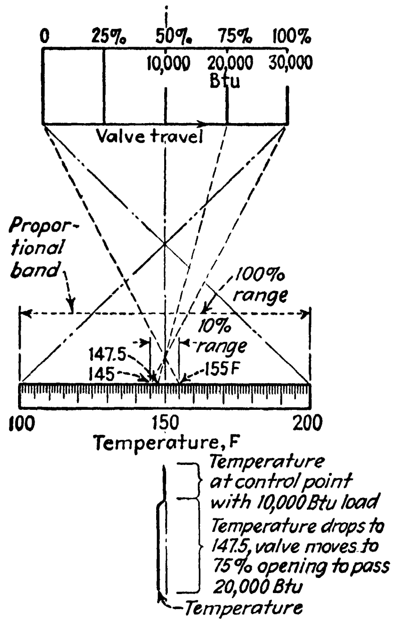


FIG. 10–12.—Effect of giving valve full travel and varying the instrument control range. (*Courtesy of Brown Instrument Co.*)

Figure 10–14 illustrates the complete control circuit for a resistance-thermometer controller using a balancing bridge and electron-tube power pack, which furnishes power to control the valve-operating motor. Unbalance of the measuring bridge causes current to flow through the correct plate circuit of the double-triode tube and into the corresponding reactor to energize the motor winding. The motor then positions the control valve and rotates the slide wire to balance the bridge circuit.

Floating Controllers.—With floating controllers, the position of the final control valve bears no fixed relation to the temperature

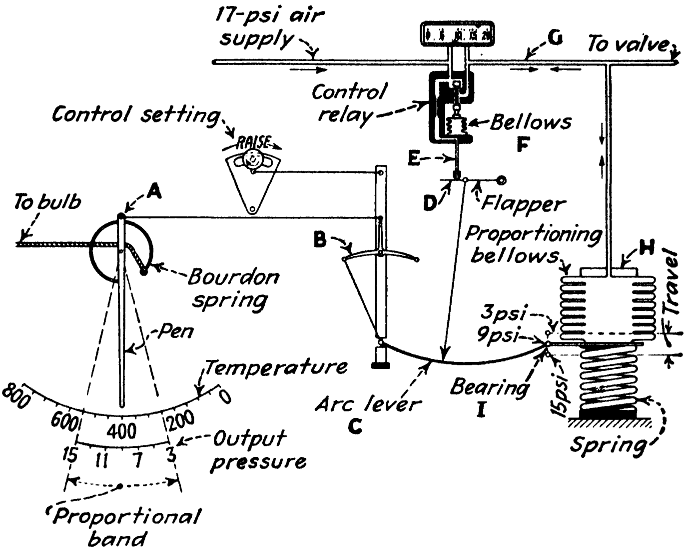


FIG. 10-13.—Proportioning controller selects a definite valve position for each load. (*Courtesy of the Foxboro Co.*)

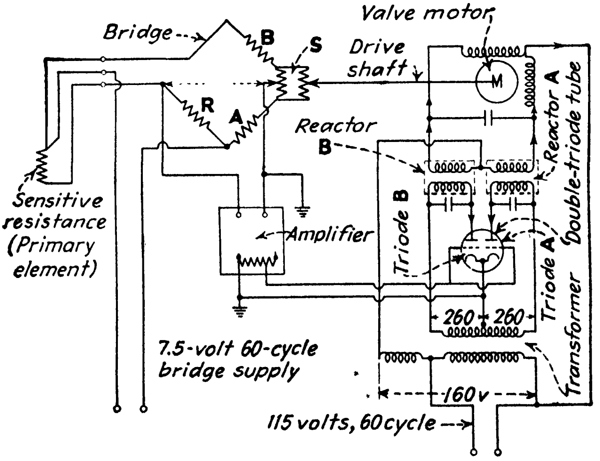


FIG. 10-14.—Resistance thermometer uses a balancing bridge and electronic-tube power circuit. (*Courtesy of Bailey Meter Co.*)

(or other variable) but is changed continuously or floated in the proper direction whenever the temperature deviates from the set point (Fig. 10–15). Ordinarily the valve moves slowly and does not reach its limit of travel before being arrested when the temperature comes back to normal. The arrested position has no relation to the temperature, and the instrument therefore has no offset characteristic. To avoid cycling, constant-valve-speed instruments have a dead zone (*neutral zone*) in which no action takes place so long as the temperature remains within this narrow range.

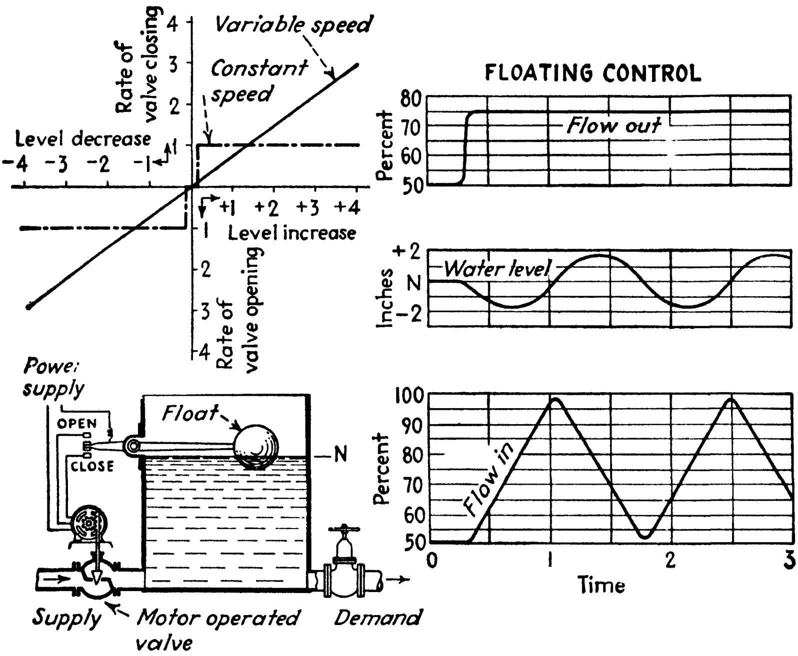


FIG. 10–15.—Floating control on a surge tank maintains constant outflow by floating the control valve from open to closed.

Consider the proportioning controller (Fig. 10–16) in service and maintaining the desired pressure. Power piston *A* is at rest. A slight change in pressure moves control diaphragm *B*, and this movement is transmitted to four-way valve *C* through lever *D*. The four-way valve then admits gas or hydraulic pressure to one side of the power piston and exhausts the opposite side so that the piston rod moves the valve or damper in the right direction.

At the same time, this movement is transmitted through levers *F*, shaft *G*, and roller chain *H* to compensating spring *E*. Its tension is thus changed to alter the effective spring loading of control diaphragm *B* and bring piston *A* to rest at the correct position. The proportional band is adjusted by changing the length of chain-lever arm to shaft *G*. This controller utilizes control-diaphragm travel to proportion the power-piston movement through shaft *G* and the chain lever in such a way that it compensates for normal valve and damper characteristics. Thus a given percentage change in the pressure under control always results in an equal percentage change in the output of the apparatus under control.

The angular relationship between levers *F*, shaft *G*, and the chain lever is such that, when the piston works near the top of its stroke, a given movement changes the tension of spring *H* at a greater rate than the same movement near the lower part of its stroke. The piston movement is therefore retarded most near the top (closing) and accelerated most near the bottom (open) to correct for dampers or valves that pass almost 50 per cent of their capacity when 25 per cent open, and 75 per cent when half open.

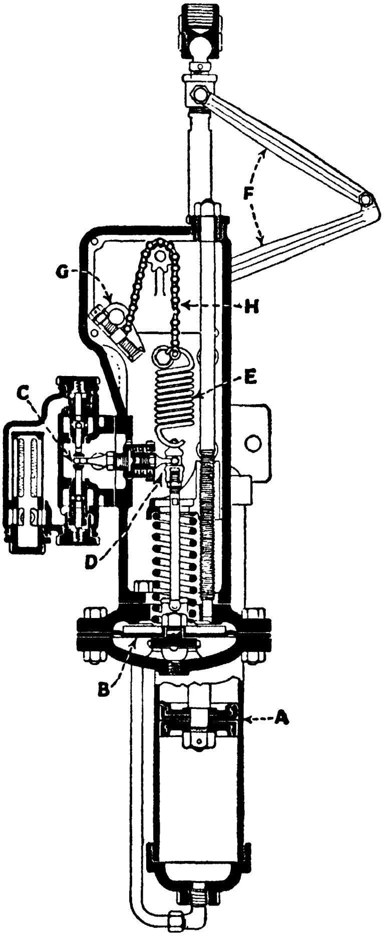


FIG. 10-16.—Controller arranged to correct for nonequal percentage flow of some valves and dampers. (*Courtesy of A. W. Cash Co.*)

Single-, two-, and proportional-speed floating controllers find wide use, although, for temperature-control work, they are generally combined with the proportional-position units described previously. Single-speed controllers move the valve continuously at constant speed as long as the temperature remains outside the neutral zone. The valve closes slowly as long as the high contact is made and, if the temperature reverses, remains in the position where contacts opened until the temperature passes through the neutral zone and operates the low contact.

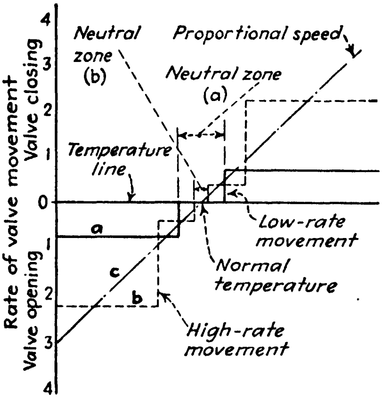


Fig. 10–17.—Dead zone is the temperature variation in which no controller action takes place.

The two-speed controller employs two valve speeds, low if the deviation is slight, high if the deviation is large. In the proportional-speed instrument, the valve speed is proportional to the pen deviation, and the controller has little or no neutral zone. Curves *a*, *b*, and *c* in Fig. 10–17 show how the rate of valve motion varies with temperature deviation in single-speed, two-speed, and proportonal-speed floating controllers, respectively. Curves *a* and *b* indicate an appreciable neutral-zone area, and curve *c* shows none. Controllers of this type are extremely important as an added part to proportional-position instruments, where they reset the controller to the set point after each demand change and thus eliminate offset.

The electrical controller in Fig. 10–18 is used with any instrument having a high and low contact with a neutral position between them. The wiring diagram shows this unit connected to a pyrometer. Starting with a cold furnace, contact *L* closes and energizes reversing motor *A*, which drives switch block *B* slowly across to limit switch *C*. Power motor *D* opens the control valve full and moves rider *E* rapidly up to limit switch *F*. Rider *E* always travels faster than switch block *B*.

When the furnace temperature reaches the set point, contact *H* energizes relay *G*, which in turn starts power motor *D* in the closing direction and causes reversing motor *A* to move switch block *B* to the left. As motor *D* closes the fuel valve, it moves rider *E* across to limit switch *I*. If the temperature falls, contact *H* opens. This deenergizes relay *G*, and its upper contacts close to operate motor *D* in the opening direction. *D* then opens the fuel valve until rider *E* moves to the right, touches switch block *B*, and opens the switch mounted thereon. The switch breaks the motor field circuit. If the temperature again rises, the main and reversing motors operate to move *B* and *E* to the left. *E* travels faster and goes to the limit switch; *B* moves slowly and stops when the *H* contact opens.

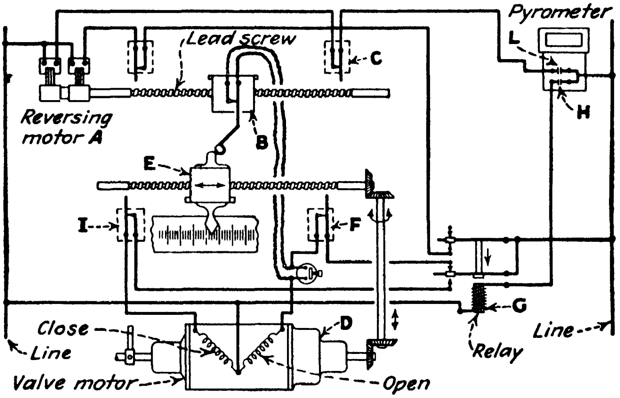


FIG. 10-18.—The floating switch block *B* determines the final position of the main control valve by stopping operating motor *D*. (*Courtesy of Wheelco Instruments Co.*)

Putting proportional-position (the valve opening determined by the temperature or other variable) and proportional-speed-floating control (the valve moves at a rate proportional to the magnitude of the temperature change) into the same controller produces a system that has the stability of a proportional controller and the constant set point of the floating unit. This controller moves the main valve at a rate whose sum is that of the rate proportional to the deviation and the rate of change of this deviation. The stabilizing influence of the proportional section eliminates the need of a neutral zone in the floating mechanism, the latter correcting for the offset characteristic of the former. In other words, after a change, the temperature (or other variable) comes back to the set point.

Figure 10–19 explains the action of such a unit if we assume that a sudden change in demand occurs at time 1 on curve *a*. As the temperature falls, curve *b*, both types of control response go into action. The proportional unit, curve *c*, changes the valve opening as the temperature varies but has no permanent effect in determining the final valve position. Floating control action, curve *d*, operates the valve at a rate proportional to temperature deviation, eventually leaving the valve positioned for the new rate. Curve *e* shows the resulting valve movement.

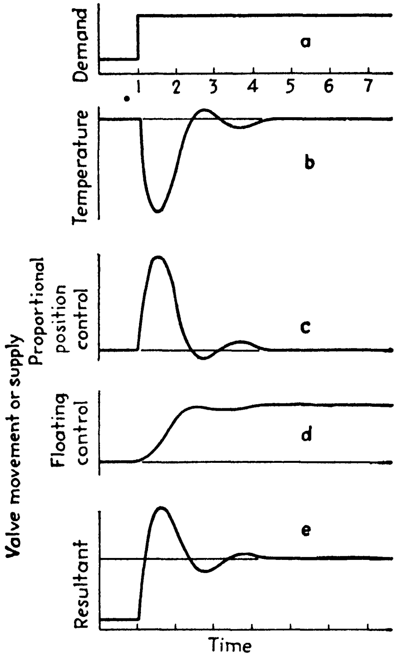


FIG. 10–19.—Resulting valve movement when controlled by a combined proportioning-floating controller.

Adjustments consist in varying the intensities of the proportional-position and floating action. In some instruments, changing the proportional-band width affects the floating speed. On others, the floating speed has an independent adjustment. Curves *a* and *b* in Fig. 10–20 show the effect of decreasing the proportional-band width; too great a reduction produces sustained oscillations. Curves *a*, *b*, and *c* in Fig. 10–21 show the results of increasing the floating speed. These devices are classified as proportioning controllers with automatic reset—they eliminate offset and return the temperature to its set point after a load change. Reset response acts only when the instrument pen is away from the set pointer.

The controller in Fig. 10–22 incorporates an automatic reset. Large bellows *A* is the proportioning unit; bellows *B* and both inner ones constitute the reset feature. The inner bellows are connected rigidly together through a rod that serves as the flapper pivot. The space between the bellows on either side is liquid-filled, and the chambers connect through the needle-valve passage. The outside of bellows *B* opens to the atmosphere.

Assume that the pen has moved from the set point and shifted the flapper away from the nozzle, thus lowering the pressure in the main control-valve line. This reduced pressure on bellows *A* permits it to move to the left, reducing the liquid pressure on that side. The liquid pressure on the right side momentarily remains the same. Since bellows *B* is free to move and maintain pressure on the liquid inside it, the unbalanced pressure forces liquid through the needle valve and into the left-hand bellows. The result is to move the flapper back to the nozzle. Automatic-reset action takes place as soon as the inside bellows has moved away from the set-point position because the left-side liquid is at a lower pressure.

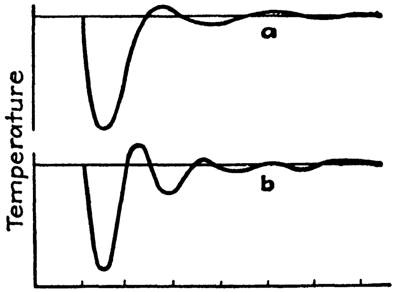


FIG. 10–20.—Reducing the proportional or throttling-range width causes oscillation.

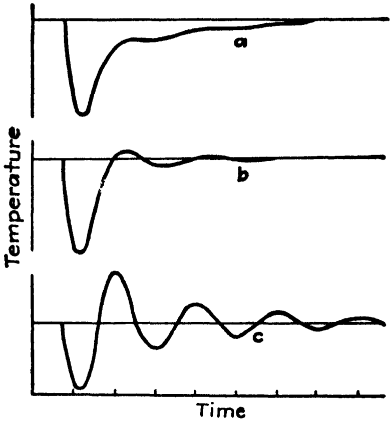


FIG. 10–21.—Increasing floating speed (*a* to *c*) too much causes the controller to cycle, or hunt.

As liquid flows from the high- to the low-pressure side, the pressures begin to equalize, and the inside-bellows system moves to the right toward its balanced position. This restoration moves the flapper to uncover the nozzle and change the control-valve pressure until the entire system comes back to balance and the pen coincides with the set pointer. Automatic-reset action can be controlled by adjusting the needle valve. The width of the proportional band is adjusted by moving fulcrum point *X* vertically up or down.

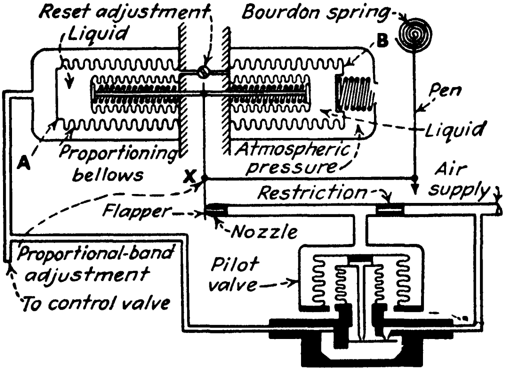


FIG. 10-22.—Automatic-reset attachment eliminates controller droop (*offset*) and holds the temperature constant. (*Courtesy of Brown Instrument Co.*)

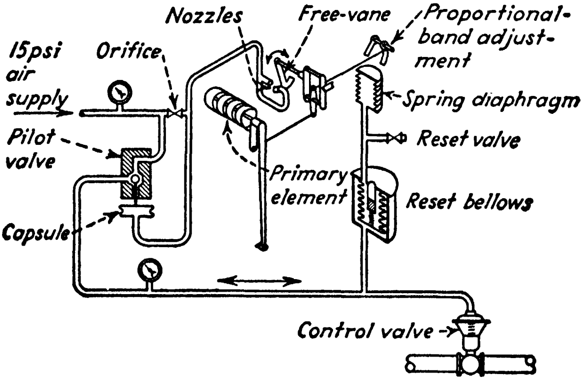


FIG. 10-23.—The free-vane controller floats the flapper or throttling vane between two nozzles to reduce the load on sensitive element. (*Courtesy of The Bristol Co.*)

An automatic-reset feature in the free-vane controller is illustrated in Fig. 10-23. The output pressure from the pilot valve goes to the outside of the reset bellows, which, at equilibrium, has atmospheric pressure on its inside. The inside of the bellows connects to the spring diaphragm (restoring bellows) by a tube that carries the reset needle valve. The restoring bellows operates through a linkage to reduce the controller sensitivity (*proportional*

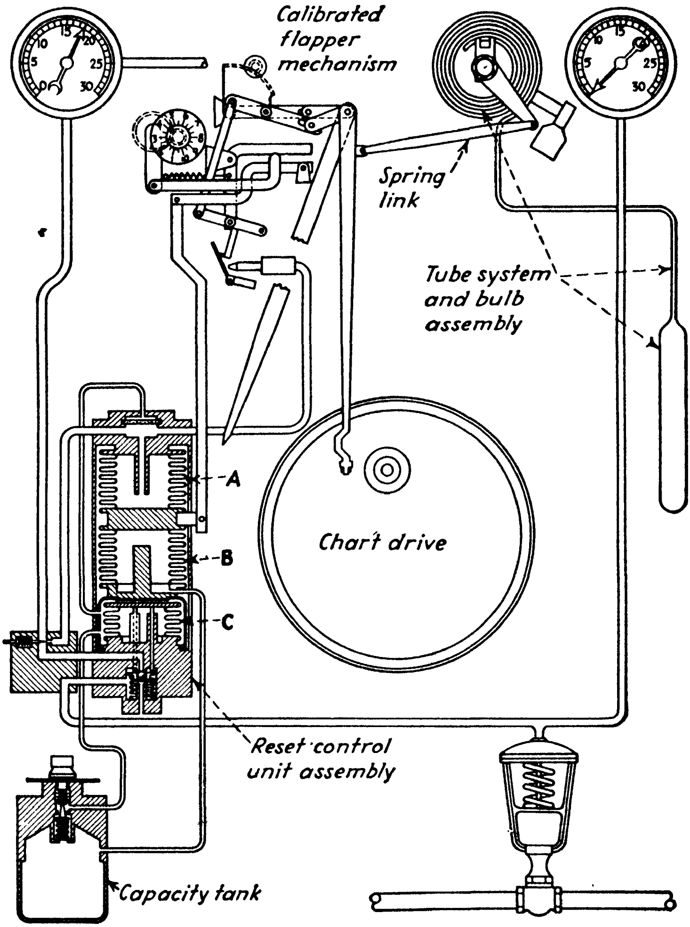


FIG. 10-24.—Proportioning controller equipped with automatic reset. (*Courtesy of C. J. Tagliabue Mfg. Co.*)

band). Pressure in the line between the two bellows vents to the atmosphere through the needle valve.

Suppose the free vane moves between the two nozzles to throttle the discharge and reduce the pilot-valve output pressure. This reduced pressure acting outside the reset bellows allows it to expand, thus reducing the pressure inside the reset bellows, connecting tube, and spring diaphragm, bringing it below atmospheric pressure. The spring diaphragm contracts and cancels a part of the free-vane movement through the connected linkage. This is the proportional-band reducing action. Since the needle valve is partly open the internal pressure in the bellows and spring diaphragm approaches atmospheric. Thus the original proportional-band reduction is canceled and the automatic reset obtained.

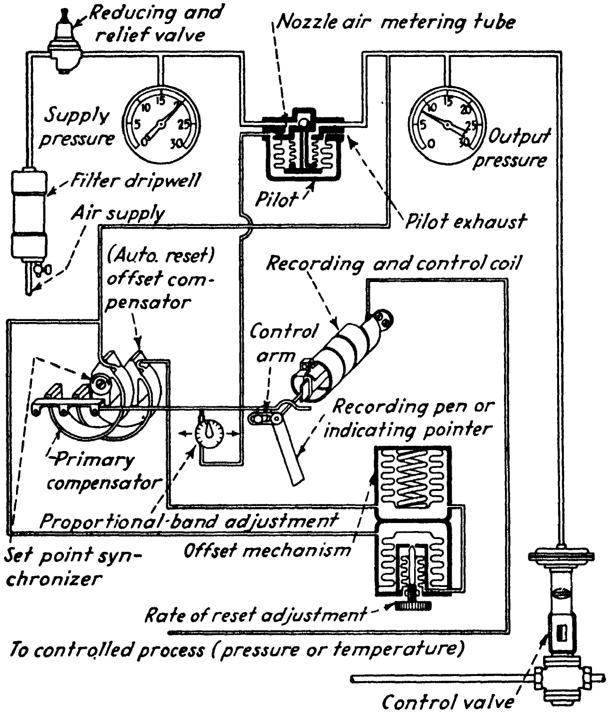


FIG. 10–25.—Automatic reset feature incorporating a bourbon tube. (*Courtesy of Mason-Neilan Regulator Co.*)

The device in Fig. 10–24 uses a capacity tank and bellows to incorporate an automatic-reset action. In operation, the pen arm, receiving an impulse from the primary element, starts to move the flapper closer to the nozzle and obstruct the air discharge. Pressure builds up in upper bellows *A* and air-valve bellows *B* and up to the adjustable needle valve in the capacity tank. Although restricted at this point, the air continues to flow in a small degree into the capacity tank and the middle bellows *C*.

As the upper bellows starts to expand, because of the unbalanced pressure between it and the lower bellows, the link fastened between them moves down, slowly opening the nozzle and keeping the flapper in a throttling position. During this action, bellows *B* is adjusting its valve to regulate the air flow to the diaphragm valve.

When air passes through the needle valve into the capacity tank and bellows *C*, a pressure builds up in the latter equal to that in bellows *A*. The lever between them then returns to its original position, and the mechanism is in balance. The instrument in Fig. 10–25 uses a bourdon tube for automatic reset.

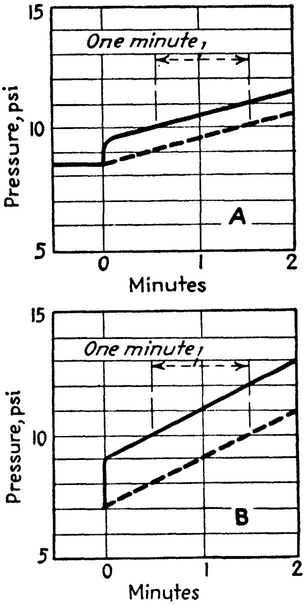


FIG. 10–26.—Rate-response effect causes the controller to increase its output pressure on sudden load changes.

The preact attachment (*rate action*), when applied to proportional or proportional-plus-floating control, adjusts the valve more rapidly and moves it farther in response to sudden load changes than would otherwise occur. Since this response produces an additional output-pressure change per rate of pen movement, it has a unit of preact time in minutes.

Assume the controller pen moving away from the set point at such a rate that the proportional-response output pressure changes

1 psi per min (dotted line in Fig. 10-26*A*). Adding 1 min preact time causes the output to follow the solid line 1 psi higher. Without altering the setting, a pen velocity twice as rapid gives 2 psi additional pressure (Fig. 10–26*B*). The time by which the solid line leads the dotted line is the preact time. A preact attachment does not replace an automatic reset, since it ceases to act when the pen becomes stationary.

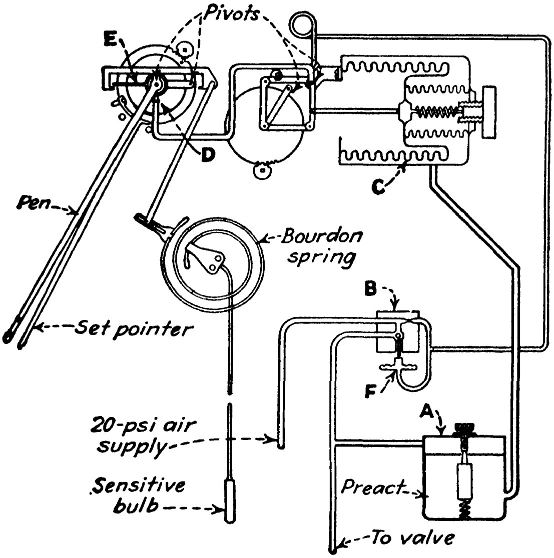


FIG. 10–27.—Preact device (rate-response effect) retards air flow to and from sensitivity bellows *C*. (*Courtesy of Taylor Instrument Cos.*)

The action is achieved by interposing a damping device *A* between relay valve *B* and follow-up bellows *C*, which retards air flow to and from the proportional-band-adjusting assembly (Fig. 10–27). Assume that the control is at equilibrium with an output pressure of 6 psi applied to the diaphragm valve, the proportional band is set for 4 psi per in. pen travel, and the preact setting is 1 min. The pressure outside bellows *C* is the same as that on the valve diaphragm, 6 psi.

Suppose the temperature begins to increase at a constant rate of

1 in. pen travel per minute. Without a preact device, the output pressure would rise at the rate of 4 psi per min, and 1 min later it would be a total of 10 psi $(6 + 4)$. However, the preact device throttles the air flow to bellows C, which retards the follow-up motion of nozzle D and maintains slightly more clearance between the nozzle and the rising circular baffle E. This causes the pressure rise to lag in capsular chamber F; its ball valve remains lower and the output pressure to the valve higher than dictated by proportional response alone, and valve-closing action is therefore accelerated.

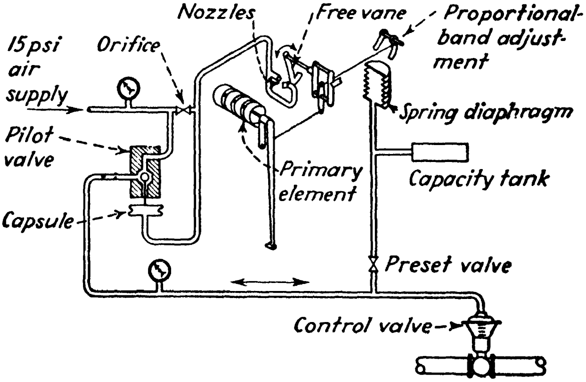


FIG. 10-28.—Rate-response effect uses a capacity tank and separate needle valve (preset). (*Courtesy of The Bristol Co.*)

Additional output pressure is the product of preact time and the rate of proportional-response pressure change, $4 \times 1 = 4$ psi. Therefore, at the end of 1 min, the output pressure is 14 instead of 10 psi. The pressure soon equalizes, and the controller comes to rest. The free-vane instrument in Fig. 10–28 uses a preset valve and capacity tank to obtain the same results as in the device just described.

The pneumatic controller in Fig. 10–29 employs a circular baffle to throttle the nozzle discharge. The complete instrument includes preact and automatic-reset devices. To follow its operation, assume that the temperature is starting to deviate downward from its control (*set*) point.

Primary-element arm *A* rotates clockwise and moves hook arm *B* counterclockwise about its pivot *C* to lower circular baffle extension *D* and cut off air discharge from nozzle *E*. Therefore pressure builds up in capsular chamber *F* and raises ball valve *G* against its upper seat to shut off air pressure to valve diaphragm chamber *H*.

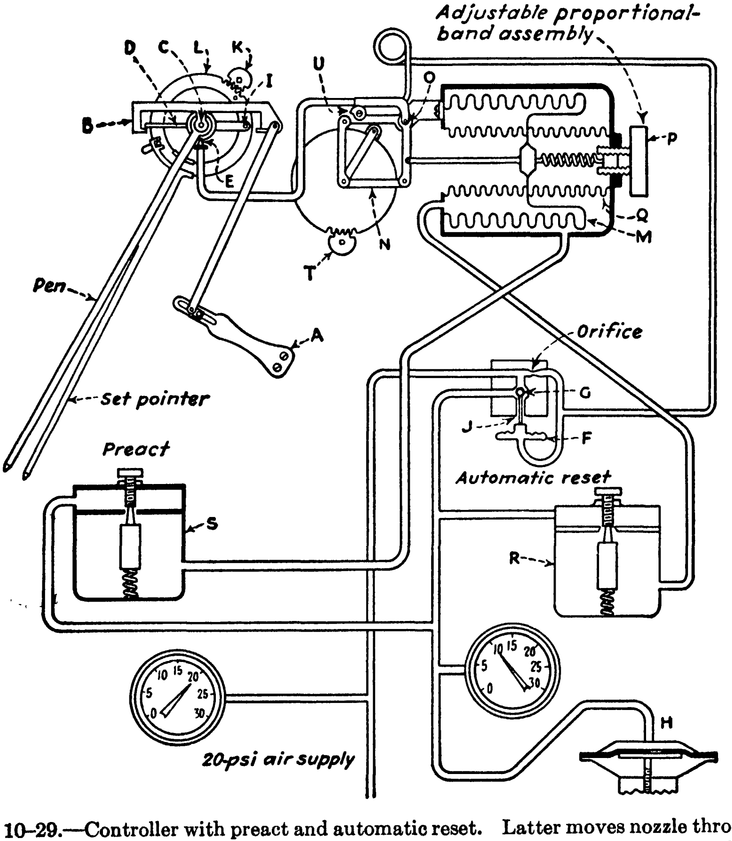


Fig. 10-29.—Controller with preact and automatic reset. Latter moves nozzle through a parallelogram instead of acting on the flapper. (*Courtesy of Taylor Instrument Cos.*)

When temperature reaches the set point, the left end of arm *B* lifts circular baffle *D*, which pivots at *I*, away from nozzle *E*. Air escapes and deflates capsular chamber *F*, which lowers ball valve *G* to some position between its upper and lower seats. (Passage *J* acts as a bleed vent to adjust or relieve pressure on the diaphragm-valve line.) Air flow then partly closes the diaphragm valve to maintain the temperature at the desired point. If the temperature exceeds its set point, arm *B* moves the baffle still farther away from the orifice. The capsular chamber then deflates, lowering the ball to its lower seat and shutting off exhaust passage *J*. Full air pressure then acts on the valve diaphragm.

So far we have discussed only the simple operations of the controller without considering its various corrective elements. Assume that the controller is at equilibrium and we wish to lower the set point. Turning knob *K* clockwise rotates gear wheel *L* to lift baffle pivot point *I* and raise the baffle away from the nozzle. The resultant decreased pressure in the capsular chamber lowers the ball toward its bottom seat, which allows more pressure to act on the valve diaphragm. This pressure also acts on the outside of bellows *M*, pushing it to the left. A pin transmits this movement to parallelogram *N*, which swings the nozzle upward around its pivot *O* to follow the circular baffle until the air pressure is properly regulated. Turning hand knob *P* adjusts for correct output pressure and synchronizes the set point to the set pointer.

This restoration is performed automatically by inner bellows *Q* and needle valve *R*. Any new pressure applied to the diaphragm valve is also applied to the needle valve *R* and the outside of bellows *M* through needle valve *S*, causing nozzle *E* to move upward until the circular baffle again throttles the escaping air. Differential pressure leaks through needle valve *R* to the inside of bellows *M* and moves it to the right. Thus the nozzle is moved away from its baffle, which results in a further increase in output pressure from the capsular valve to maintain the same differential pressure across the needle valve.

The increasing output pressure slowly closes the diaphragm valve, and the decreasing flow lowers the temperature toward the set point. Air flow through the needle valve to the inside of bellows *M* decreases with decreasing pressure differential, which is proportional to the pen distance from the set point. When the pen reaches the set point, the nozzle is at equilibrium position, equal pressure is acting both outside and inside of bellows *M*, and the reset response ceases to act since no differential pressure exists across needle valve *R*.

An adjustable proportional-band controller produces an output-pressure change proportional to the amount of pen deviation. The preact device *S*, interposed between the capsular valve and the follow-up bellows *M*, adds a response proportional to the rate of pen deviation by retarding air flow to and from the proportional-band adjusting assembly.

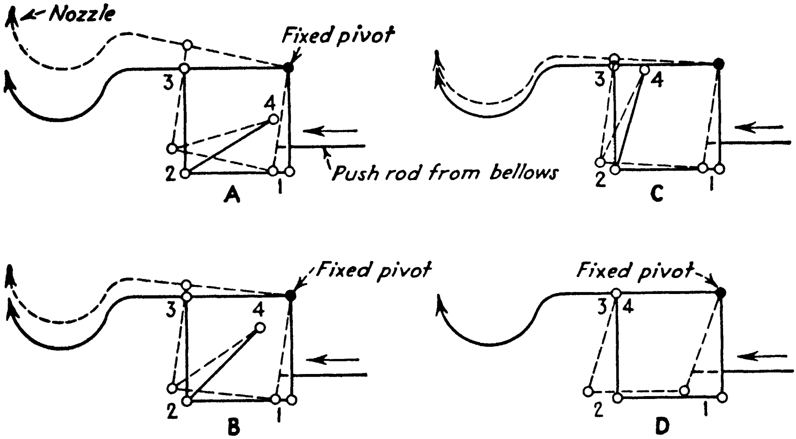


FIG. 10-30.—Parallelogram action on nozzle can be changed by adjusting link pivot 4.

The operation of the parallelogram *N* (Figs. 10-27 and 10-29) can be easily explained by referring to Fig. 10-30. In *A*, a small movement of the push rod to the left moves hinge 1 a small amount around the fixed pivot. This results in greater motion of point 2 around pivot 4 and still greater movement of point 3 around the fixed pivot. The nozzle moves a considerable distance, which greatly reduces the instrument proportional band. Now, if we move pivot point 4 upward by turning gear wheel *T* (Fig. 10-29), the nozzle movement resulting from the same push-rod movement is considerably less, B (Fig. 10–30). Moving point 4 upward still farther, C, results in very little nozzle movement under the same conditions. When pivot 4 coincides with point 3, D, the push-rod movement is not transmitted to the nozzle, and the instrument band is narrow.

With the parallelogram, set as in A, the nozzle will follow the circular baffle, and the pen will have to travel over a considerable portion of the chart to open and close the nozzle completely. Thus the proportional band is wide.

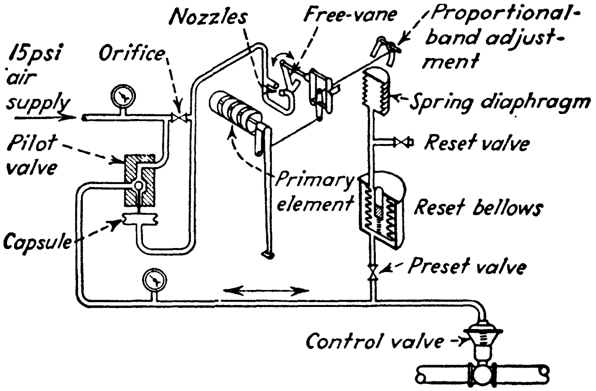


FIG. 10–31.—Free-vane controller equipped with automatic reset and rate-response effect (preset). (*Courtesy of The Bristol Co.*)

The knurled adjusting knob P and its spring (Fig. 10–29) determine the initial position of the push rod and therefore the set point. If the spring tension is great, the output pressure must be high to make the nozzle follow the baffle; therefore, when the pen and set pointer are together, the output air pressure is high. If the spring tension is low, the reverse is true. Thus the adjusting knob can be used to synchronize the pen and set pointer manually. However, if the parallelogram is adjusted as in D (Fig. 10–30), with pivot 4 coinciding with point 3, the push-rod movement is ineffective, and therefore the knob adjustment is ineffective. Under these conditions, the pen and set pointer can be synchronized by adjusting the cam U (Fig. 10–29).

Although the operation of each part of this controller has been described separately, all the parts operate in unison on a change in the controlled variable. Starting at equilibrium, pressure from the capsular valve is equalized on both sides of bellows *M* and the valve diaphragm. If the circular baffle moves in relation to its nozzle, the capsular valve changes the air pressure on the valve diaphragm and at the entrance to needle valves *R* and *S*. The rate at which the air pressure changes on both the inside and the outside of bellows *M* depends on the adjustment of each needle valve. In the free-vane controller in Fig. 10–31, the rate action (preact-preset) is incorporated by installing a needle valve in the air line to the automatic-reset bellows.

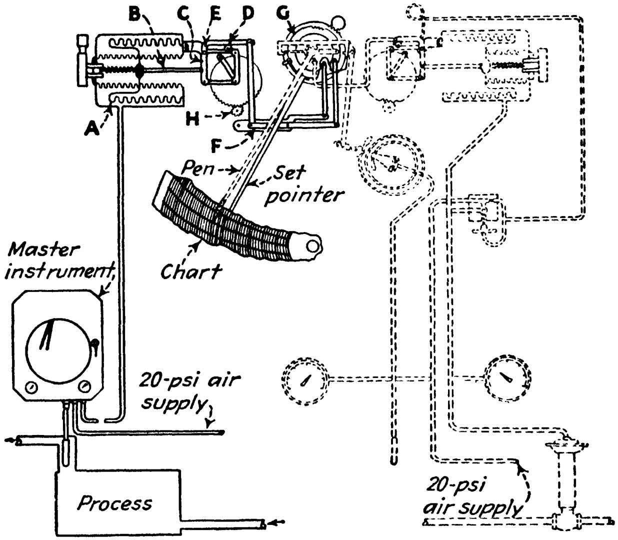


FIG. 10–32.—Typical controller (dotted lines) has its set point changed by a master unit actuated by an uncontrollable variable such as outside temperature. (*Courtesy of Taylor Instrument Cos.*)

In a pneumatic-set control system, the set pointer (set point) is adjusted by another instrument acted upon by an uncontrollable variable. For example, a temperature transmitter affected by outdoor temperature can vary the set point of an inside-temperature controller. The adjusting mechanism is shown by solid lines in Fig. 10–32 to differentiate it from the controller proper. In operation, an increase in air pressure sent out from the master instrument enters the chamber outside bellows *A* and forces pin *B* to the right. Parallelogram *C* transmits this movement to arm *D*, pivoted at *E*, swinging it upward to lift the left end of lever *F*. Downward movement of the right end turns floating gear *G* and moves the set pointer to the left. If the air pressure sent out by the master instrument drops, the reverse action takes place. The extent of set-pointer travel, per pound per square inch change in bellows pressure, is controlled by adjusting knob *H*.

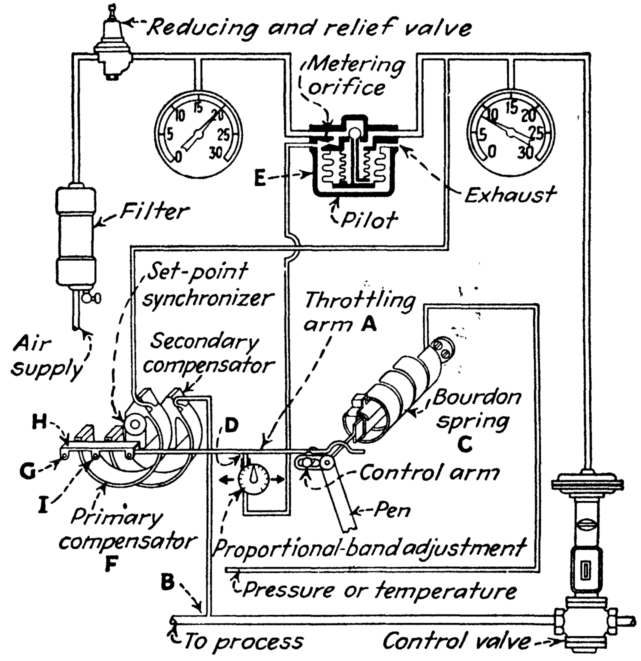


FIG. 10–33.—Unit equipped with a secondary compensator that corrects for fluctuations in pressure beyond the control valve. (*Courtesy of C. J. Tagliabue Mfg. Co.*)

Another supplementary control (secondary compensator in Fig. 10–33) corrects for fluctuations in the processing medium before it

causes a serious upset. Through this device, throttling arm *A* feels **instant changes** in pressure beyond the control valve at *B*. Assume **that the** controlled temperature or pressure increases and uncoils **control** helix *C* slightly. This lowers the right-hand end of the **throttling** arm to reduce the air discharge from nozzle *D* and increase the pressure outside pilot bellows *E*. The bellows stem moves up and lifts the ball valve from its seat to increase the air pressure inside the inner bellows, to the valve diaphragm and primary compensator *F*.

The immediate effect is that the valve starts moving, and compensator *F* tends to straighten slightly and move point *G* downward. As lever *H* pivots at *I*, this movement raises the left-hand end of the throttling arm away from nozzle *D* to proportion the air output according to the change in the variable. Lower pressure beyond the control valve at point *B* permits the secondary compensator to coil slightly, raise point *I* (pivot of lever *H*), and assist in raising the left end of the throttling arm. After the controller is at equilibrium, the secondary compensator detects and corrects for pressure changes at point *B*.

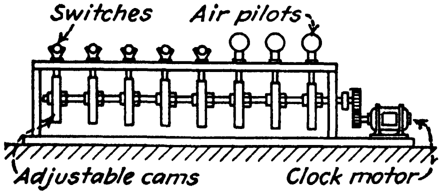


Fig. 10-34.—A time-cycle controller carrying both electric contacts and air pilot valves.

Controllers so far discussed do not take into account the time element, which is important in some processes. This requires an independent instrument (cycle controller or program clock) which, when connected into the control circuit, accurately times the application of heat, cold, or other required services. In this device (Fig. 10-34), a clock mechanism (usually synchronous-motor-driven) rotates a shaft carrying a series of adjustable cams that operate electric switches or air valves (Fig. 10-35) to control the supply valves or dampers.

Clocks are available that run continuously in one direction or that operate in one direction for half the cycle and then reverse and stop at the end of the second half. Some timers repeat either immediately or after a predetermined interval, whereas others must be reset manually after each operation. The simplest use is to turn on the building heat and admit the steam in short cycles to eliminate prolonged boiler overload. For cooking or presswork, these timers shut off the steam, relieve the pressure, and admit and shut off the cooling water.

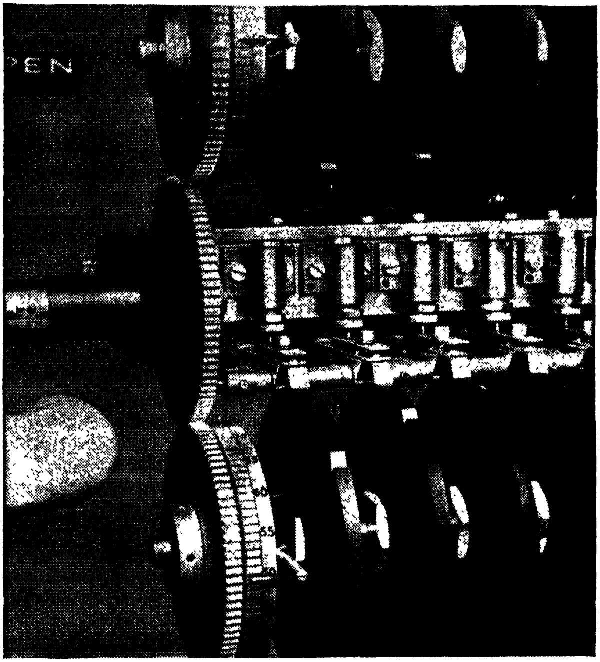


FIG. 10–35.—Driven by an electric-clock motor, a series of cams operates the air valves. (*Courtesy of Automatic Temperature Control Co., Inc.*)

Miscellaneous Controllers.—Figures 10–36 and 10–37 illustrate two conventional proportional controllers of different design. As explained previously, a proportional controller is suited only for processes where load changes are small and of short duration, because the instrument has an inherent offset characteristic—it is unable to hold the variable at a constant value. Where this factor

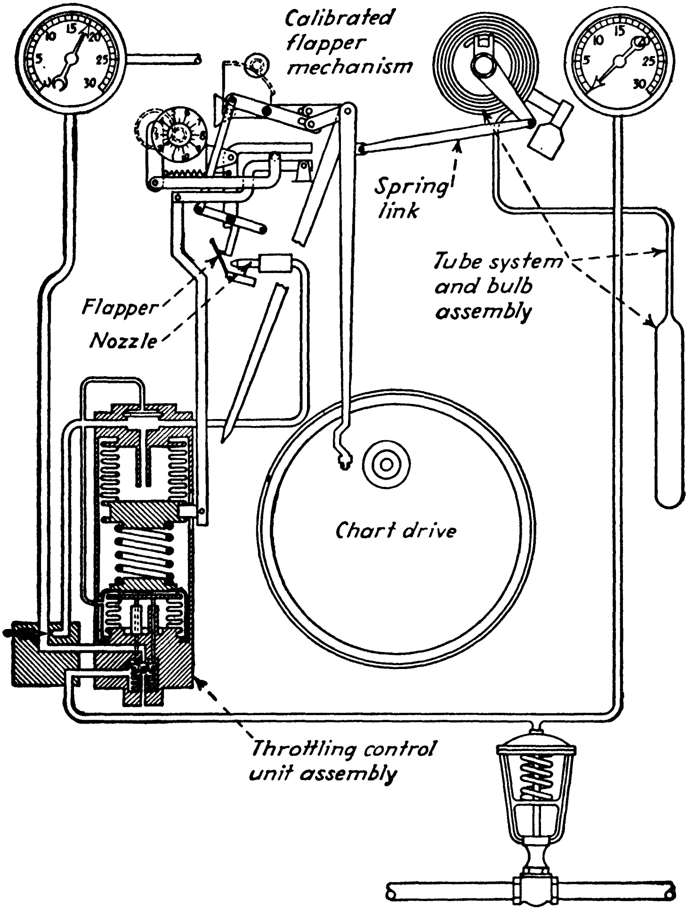


FIG. 10–36.—The C. J. Tagliabue proportioning controller without added features.

is important, a reset mechanism must be added. In Fig. 10–36, the air pressure as adjusted by the flapper, and the nozzle acts inside the upper bellows. On increased pressure (flapper near nozzle), the bellows expands against the opposing spring and pulls down the connecting link, which, through a system of levers, moves the flapper away from the nozzle. This keeps the flapper in a throttling position until the set point is reached. During the action, air pressure from the upper bellows passes into the air-valve bellows (lower one) and causes it to adjust the air flow to the diaphragm valve.

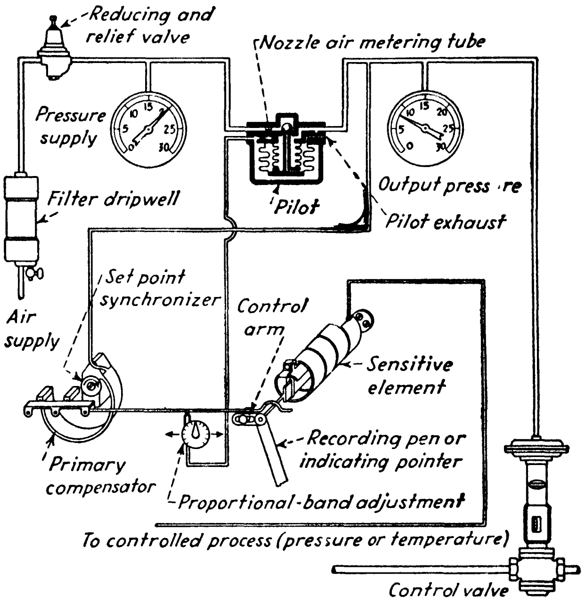


FIG. 10–37.—Mason-Neilan proportioning controller without added features.

In Fig. 10–37, the proportioning unit (primary compensator) is a bourdon tube fixed at one end and fed with air from the output pressure. The free end of the tube supports one end of the throttling arm. The right-hand end of the throttling arm is moved by the primary element. An increase in controlled pressure acting on the primary element lowers the right-hand end of the throttling arm toward the nozzle. The resulting increase in output pressure tends to uncoil the primary compensator and raise the throttling arm to its original position in relation to the nozzle. Thus for every position of the measuring element, there is a corresponding position of the primary compensator and a corresponding output pressure. Proportional-band adjustment is obtained by moving the nozzle to the left or right in relation to the throttling arm.

The device in Fig. 10–38 is a proportioning controller equipped with an automatic reset. Assume that the instrument is in control and conditions are in equilibrium so that 8 psi air pressure exists in the controlled-valve diaphragm motor line. The same pressure will exist in the proportioning and reset bellows.

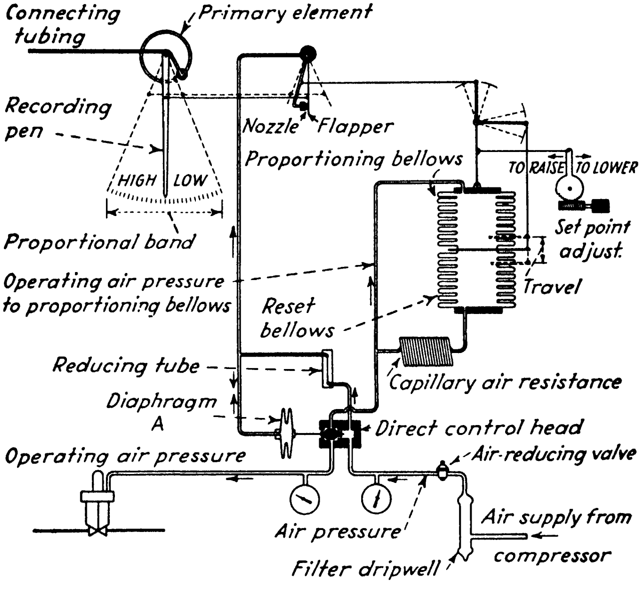


FIG. 10–38.—Foxboro proportioning controller with automatic reset.

If an upset occurs, making the primary element read lower, the flapper uncovers its nozzle and reduces the pressure in diaphragm *A*. The diaphragm contracts and admits pressure through its valve until increased pressure in the proportional bellows pulls the nozzle against its flapper again. The bellows is spring-loaded so that the air-pressure change is strictly proportional to the nozzle deflection, producing a uniform proportional band.

This increased pressure immediately starts to bleed through the capillary-air-resistance restriction into the reset bellows, which is equal in area to, and opposes the motion of, the proportioning bellows. As pressure in the reset bellows increases, the nozzle will be forced away from the flapper. However, when this happens, diaphragm *A* will open its valve farther, raising the pressure in the proportioning bellows as much as necessary to keep the nozzle just tangent to the flapper. The resetting effect continues until the controlled-valve opening is adjusted to meet the new conditions. When this occurs, pressures in both bellows are equal, and equilibrium is restored.

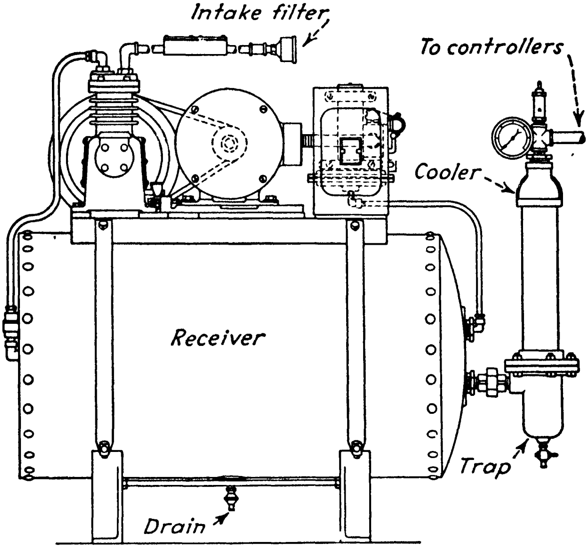


FIG. 10–39.—If no other air supply is available, install a compressor unit capable of producing at least 45 cu ft of air *per hour* for each connected controller. Use varies with design, but this is a safe minimum figure.

Pneumatic controllers need a supply of clean dry air, because moisture, oil, and dirt plug the small orifices and nozzles. If no other source is available, install a compressing unit (Fig 10–39) adequate in size and complete with intake filter, cooling unit, and moisture trap (Fig. 10–40) and receiver. Supply the cooler with chilled water if necessary so that the air will be at a temperature lower than any it will encounter in the distribution lines; this eliminates any possibility of moisture condensing out. To obtain best results, pipe 90 to 125 psi air up to the controller, and insert a filter *A* or *B* or screen *C* (Fig. 10–41) just ahead of the pressure-reducing valve. From here on, use copper or brass pipe to avoid rust particles. Blow out the line thoroughly, attach the filter and reducing valve, and adjust the latter to maintain the correct air pressure at the controller. Blow out the copper jumper before putting it in position between the reducing valve and the instrument. Make all air-line connections to the top of the main header to avoid moisture and dirt carry-over. If thread compound or shellac is used on pipe joints, apply it to the male threads sparingly.

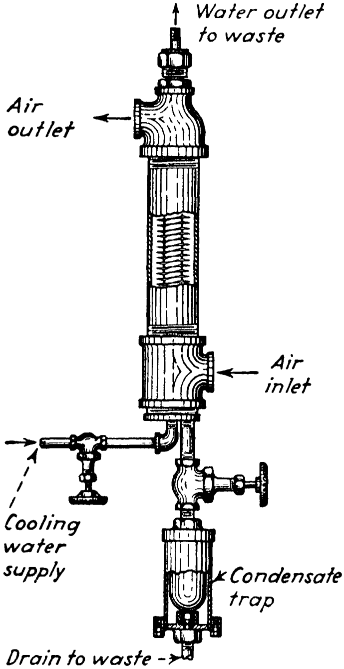


FIG. 10–40.—See that the compressor is equipped with a suitable aftercooler and moisture trap. (*Courtesy of The Powers Regulator Co.*)

Several types of air systems meet the principal requirements of pneumatic controllers and recorders:

1. A central system entirely isolated for instrument use only. It requires a compressor, aftercooler, moisture separator, air receiver, drying apparatus, and distribution system.

2. Use plant air compressors when they are reliable, but give instruments preference over other uses. Take the supply directly from the receiver, and install control equipment to ensure that other services do not drag the pressure below a safe minimum. Use an adequate separator and drying equipment.

3. Use the regular distribution system, but install drying equipment and auxiliary equipment at each take-off area that cuts in automatically if the plant pressure fails.

4. Install an individual air supply at each process unit or area. This is similar to item 1 but on a smaller scale. The only drawback to this method is the difficulty of obtaining compressors that have long life with minimum carry-over of oil.

Most instruments require air at 20 psi gage. Because pressure-reducing valves perform satisfactorily with a 5 psi pressure drop, the valves need at least 25 psig supply pressure. A leeway allowance for flexibility dictates that this pressure be 35 psig to meet all operating conditions. Further allowance of 10 psi for drop through drying apparatus and distribution lines requires that 45 psig be maintained at the central compressor station.

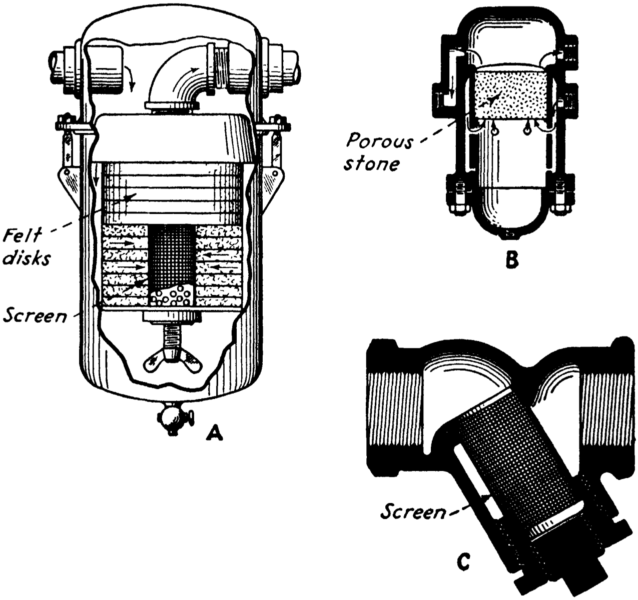


FIG. 10-41.—Types of filters and screens. [*Courtesy of* (*A*) *Staynew Filter Corp.*, (*B*) *Fisher Governor Co.*, *and* (*C*) *Sterling, Inc.*]

When designing an air-supply station for instrument use, allow 0.5 cfm of air for each pilot. For instance, if an instrument has a nozzle and pilot valve, use a factor of 2, etc. Experience shows that valve positioners should have a factor of 2 also. Knowing the number of instruments, apply the necessary factors to get the equivalent number of air users, and multiply by 0.5 to find the required amount of air-compressor capacity. Add 15 to 20 per cent for blowdown and future instruments, and select two heavy-duty continuous-service compressors (one a spare) of this capacity. Avoid machines that use the splash system of cylinder lubrication. Select drying apparatus, either cooling, adsorptive, or absorptive, capable of maintaining a dew-point temperature of 40 F for summer and –10 F for winter. When designed for summer conditions, they are usually amply large for winter.

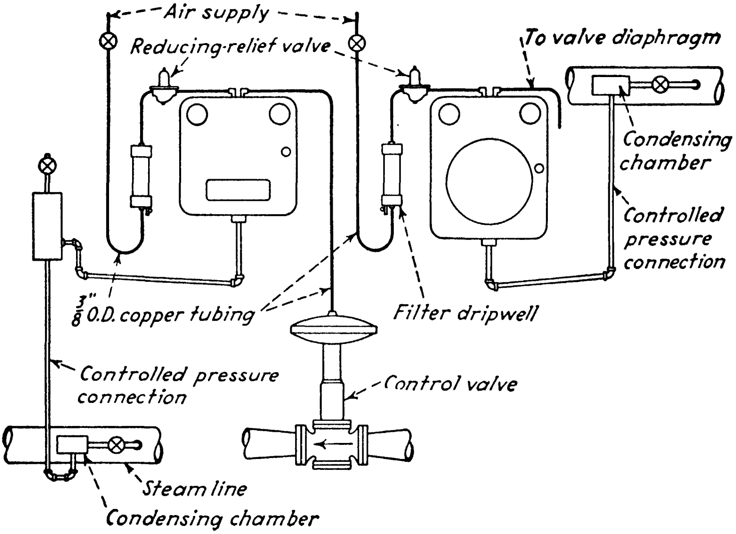


FIG. 10–42.—On fluid-flow lines, make instrument pressure tap on the side of pipe.

When making pressure-sensitive-element connections to fluid-flow lines, make the pressure tap into the side of the main line (Fig. 10–42). Never insert a tee in a large line and bush it down to fit the controller line, because strange things can happen at a pressure tap of this kind. High velocities may create eddies in this cone-shaped area that will produce false pressures at the controller. Always drill and tap large lines to take a fitting the same size as the line to the controller. Screw the fitting into the line, and make its inner end flush with the inside wall of the large pipe.

Another source of trouble is the location of the controller tap in relation to the controlled-valve position or elbows in the line. Put it far enough downstream to avoid eddies and other disturbances (see Chap. XV). On liquid and steam lines, slope the pressure tap to facilitate the venting of air and vapor. When high points are unavoidable, provide vents at such points. On air or gas installations, slope the pressure tap to drain the condensate, and install suitable drip legs.

ACKNOWLEDGMENTS

Figures 10-2 to 10-6, 10-8, 10-17, 10-19, and 10-21 are from:

BRISTOL, E. S., and PETERS, J. C., Some Fundamental Considerations in the Application of Control to Continuous Automatic Processes, *ASME Transactions,* November, 1938, pp. 641-649.

PETERS, J. C., Getting the Most from Automatic Control, *Industrial and Engineering Chemistry,* Vol. 33, September, 1941, pp. 1095-1103.

PETERS, J. C., and OLIVE, THEODORE R., Fundamental Principles of Automatic Control, *Chemical and Metallurgical Engineering,* May, 1943.

CHAPTER XI

CONTROLLER APPLICATIONS

Automatic control devices, selected and installed properly and maintained in first-class condition, will provide the answer to all industrial control problems and open up new methods of mechanization. Every industry, large or small, has some operation involving control of temperature, pressure, flow, or other variable that can be improved by applying one or more mechanical or electrical control instruments. The improved operation invariably results in higher quality of product, increased output, and lower operating costs. The examples of application here illustrated show to a small extent what can be accomplished with control instruments.

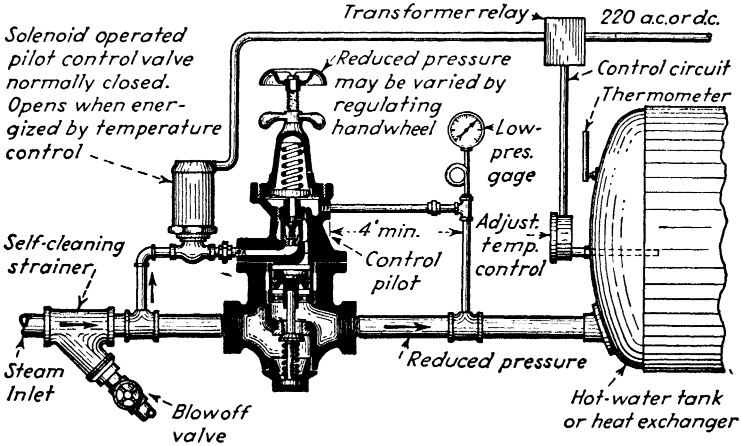


FIG. 11-1.—Temperature element opens and shuts off pilot line to pressure-reducing valve. (*Courtesy O. C. Keckley Co.*)

Figure 11-1 illustrates a combination electric and mechanical control to regulate water temperature in a tank or heat exchanger.

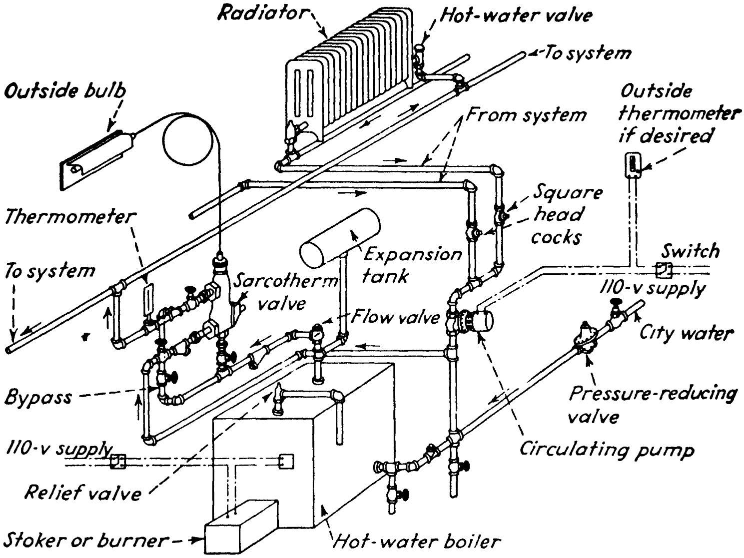


FIG. 11-2.—Outside thermometer bulb controls mixing valve in heating system. (*Courtesy of Sarcotherm Controls, Inc.*)

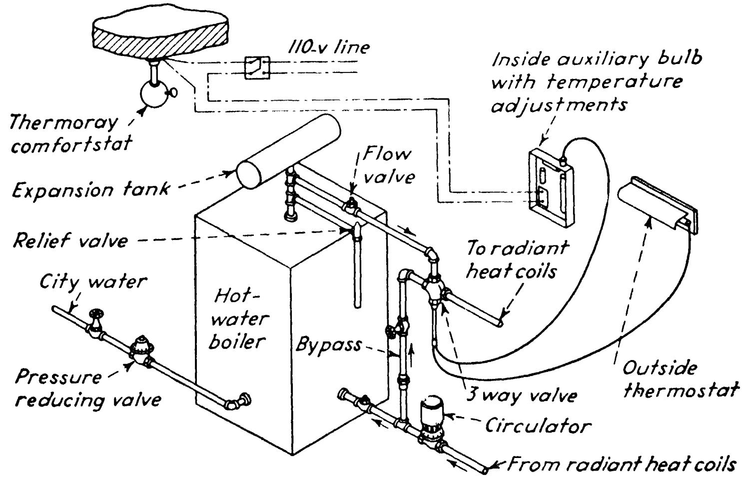


FIG. 11-3.—Outside thermostat and Thermoray controls for radiant heating systems. (*Courtesy of Sarcotherm Controls, Inc.*)

Steam from the main supply is admitted to the control pilot of a pressure-regulating valve by a solenoid-operated valve controlled by a thermostat. When the water temperature falls below the set value, the thermostatic element energizes the solenoid to open its valve. This admits steam to the regulating-valve pilot, and the main valve then operates as a pressure-reducing valve until the water temperature again comes up to normal. The thermostat then closes the solenoid valve, and the pressure-reducing valve is closed by an opposing spring under its disk.

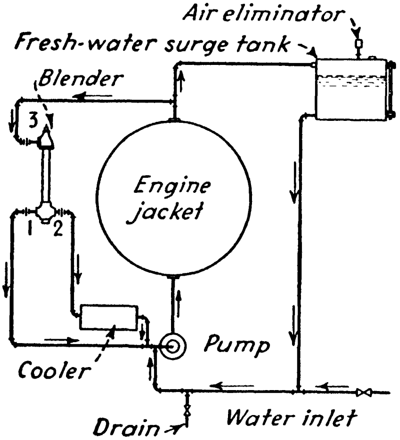


FIG. 11-4.—Mixing valve with internal thermal element controls engine jacket cooling water. (*Courtesy of Sarco Co., Inc.*)

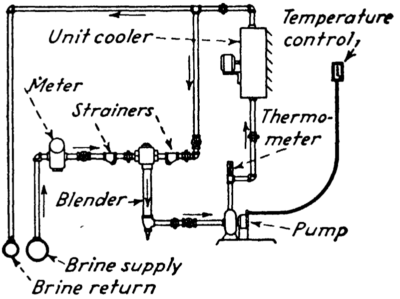


FIG. 11-5.—Thermostatic mixing valve controls brine flow through a unit cooler. (*Courtesy of Sarco Co., Inc.*)

The Sarcotherm control (Fig. 11-2) uses a thermal-sensitive bulb, subjected to outside temperature, to regulate an indoor heating system. The Sarcotherm is a mechanical three-way valve that recirculates a varying portion of the system water, adding only enough boiler water to maintain the temperature desired. Figure 11-3 illustrates its use in conjunction with a Thermoray to control a radiant-heating system. The Thermoray is sensitive to both air temperature and radiant rays and is thus responsive to the heat-loss conditions affecting the human body.

The three-way blending valve also serves to control the engine- or compressor-jacket cooling (Fig. 11–4). Heated water from the jacket flows in at point 3 and down over a thermostatic element. As long as the water remains cool, it flows out branch 1 and back to the system. When the water temperature approaches the point for which the thermostatic element is set, a portion of the water flows out branch 2 and through the cooler before being recirculated.

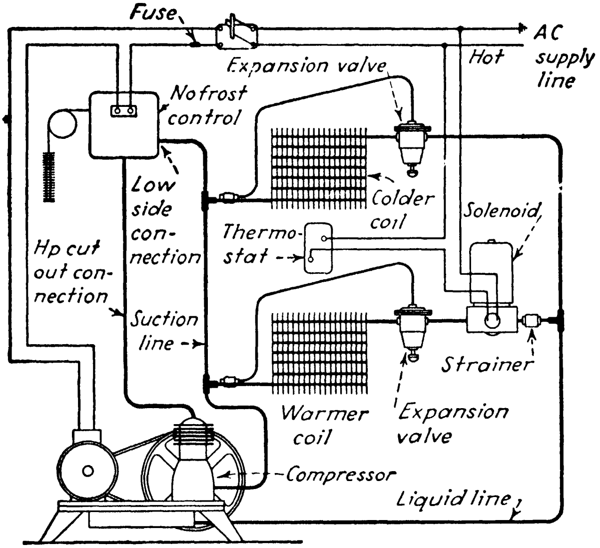


Fig. 11–6.—Thermostat and solenoid valve control a two-unit refrigerating system. (*Courtesy of Penn Electric Switch Co.*)

For cold-storage work (Fig. 11–5) the blender valve recirculates brine through the unit cooler; or, if the room temperature gets too high, it closes the by-pass branch and forces the used brine back to the refrigerating units. Refrigerating systems find numerous uses for control devices. In Fig. 11–6, individual thermostatic valves control the liquid-refrigerant flow to the coolers according to the suction temperature. In this system, one condensing unit takes care of a two-temperature system by using a thermostatic-controlled solenoid valve on the liquid line to one cooler. Figure 11–7 represents a two-unit system in which space thermostats control solenoid valves in the liquid line to each evaporator. When the solenoid valves are open, thermostatic valves, with their sensitive elements attached to the suction lines, regulate the liquid flow to the evaporators. A pressure switch starts and stops the condensing unit as the suction pressure changes.

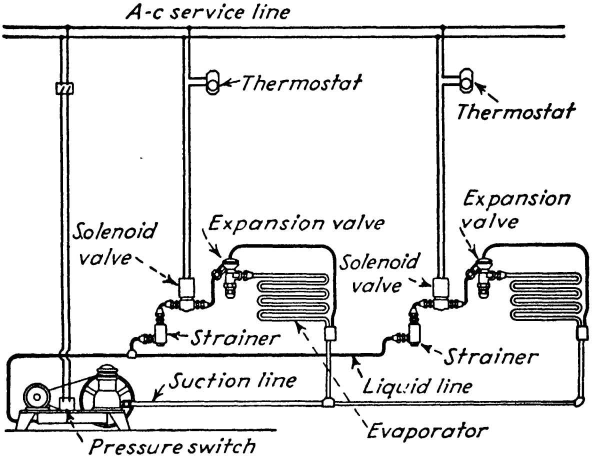


FIG. 11-7.—Superheat of suction gas regulates liquid flow to cooler after thermostat opens solenoid valve. (*Courtesy of Detroit Lubricator Co.*)

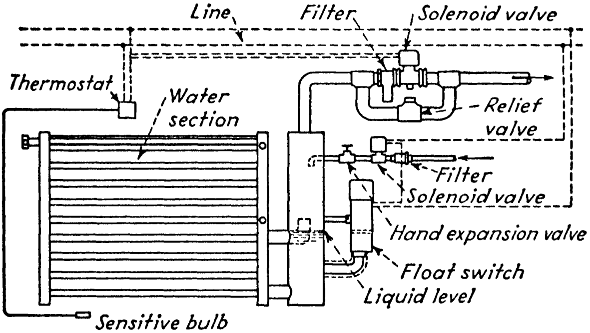


FIG. 11-8.—Float in surface cooler operates solenoid valve to regulate refrigerant flow. (*Courtesy of Alco Valve Co.*)

Another important application is where one or more full-flooded

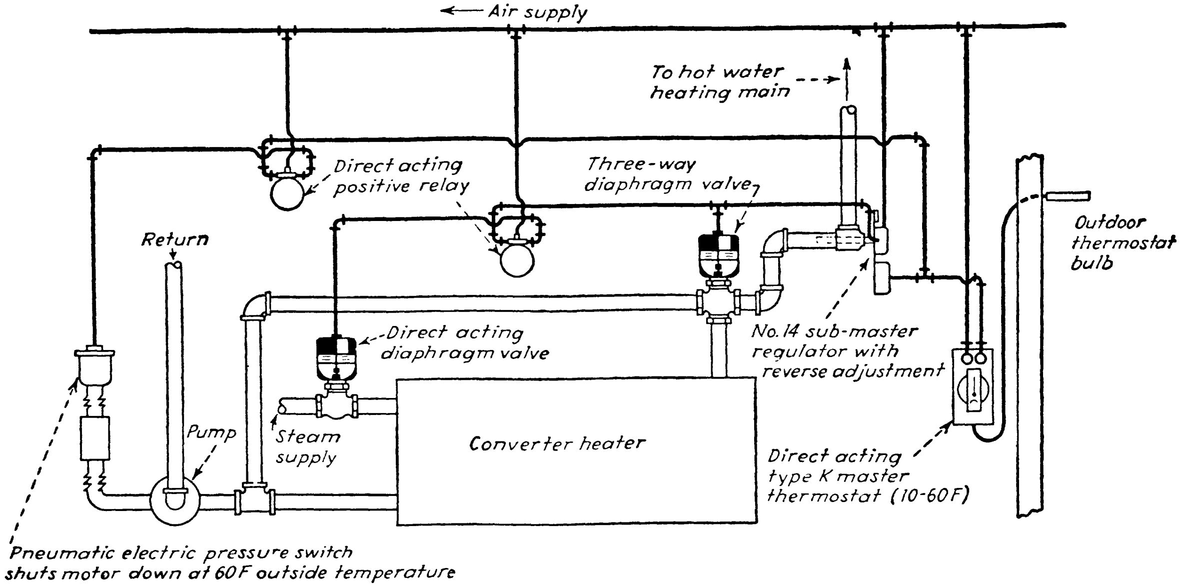


FIG. 11-9.—Master thermostat resets submaster thermostat to regulate heating system in accordance with outside temperature. (*Courtesy of The Powers Regulator Co.*)

evaporators are desired in an otherwise *dry* multiple system. In such cases, a solenoid-operated liquid stop valve, followed by a manually adjusted throttling valve, is placed in the liquid line leading to the shell or drum in which the liquid level is to be maintained by an electric float switch (Fig. 11–8). The solenoid stop valve is operated by the electric float switch, and, when the level falls below a predetermined point, the switch opens the valve. If close temperature control is desired, a solenoid-operated valve, by-passed by a relief valve, can be placed in the suction line.

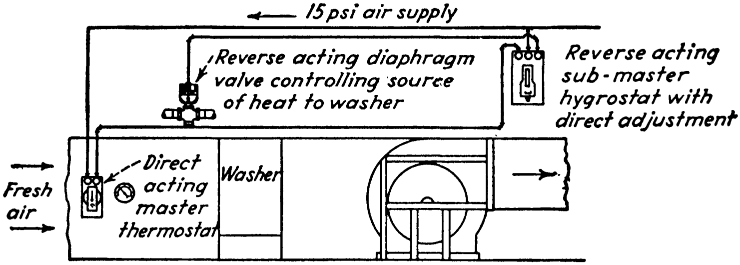


FIG. 11–10.—Master thermostat resets submaster hygrostat, which controls heat supply to air washer. (*Courtesy of The Powers Regulator Co.*)

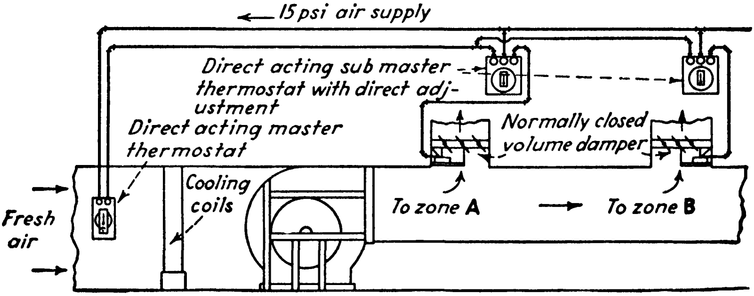


FIG. 11–11.—Master thermostat resets each zone thermostat in an air-conditioning system. (*Courtesy of The Powers Regulator Co.*)

In the forced hot-water-heating system (Fig. 11–9), a master outdoor thermostat raises or lowers the set point of the submaster regulator in accord with variations in outdoor temperature. The direct-acting submaster regulator, installed in line to a hot-water-heating main, operates a three-way valve to mix hot water from the converter with the system return water to maintain the desired delivery temperature.

When the submaster regulator closes the hot port of the three-way valve, the positive relay closes the direct-acting valve in the steam supply to the converter. When the outdoor temperature reaches 60 F, a positive relay operates a pneumatic-electric pressure switch to deenergize the pump motor. As the outdoor temperature rises, the master thermostat lowers the temperature setting of the submaster regulator, which in turn maintains a lower circulating-water temperature.

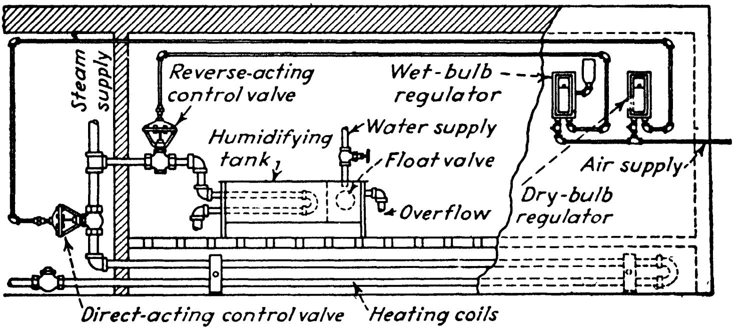


FIG. 11-12.—Wet- and dry-bulb thermometers control a humidifying system. (*Courtesy of Atlas Valve Co.*)

To prevent frosting of windows and the formation of moisture on walls and material, a master thermostat in the fresh-air intake (Fig. 11-10) raises or lowers the setting of a submaster hygrostat according to variations in outdoor temperature.

The air-conditioning system in Fig. 11-11 uses a master thermostat in the fresh-air intake to reset the control point of submaster thermostats in zones *A* and *B* and maintain the desired differential between outdoor and indoor temperatures.

A humidifying system (Fig. 11-12) uses a dry-bulb instrument to control the steam-heating valve. A wet-bulb thermometer controls steam to the humidifying tank, which is kept filled with water by a float-operated valve.

The water heater in Fig. 11–13 uses a thermostatic valve to apply water pressure to the diaphragm of the steam-supply valve. When the regulator valve is closed, pressure bleeds out the discharge vent. A sensitive bulb immersed in the contents of the jacketed kettle (Fig. 11–14) controls the steam-supply valve according to predetermined temperature limits of the cooking process.

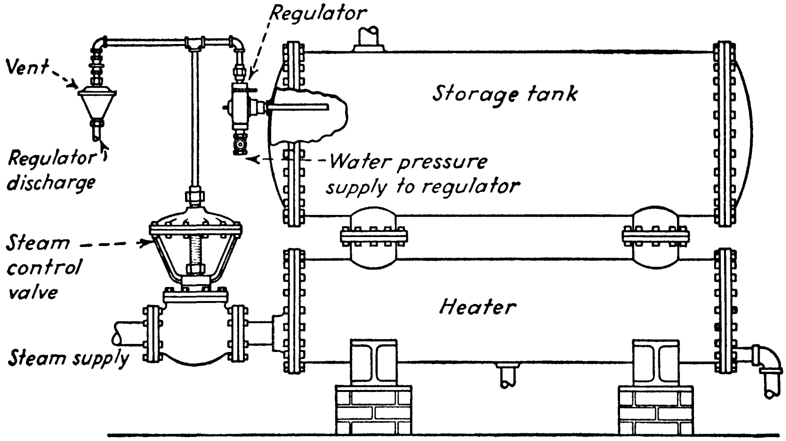


FIG. 11–13.—Thermostatic regulator controls steam-flow valve hydraulically. (*Courtesy of Atlas Valve Co.*)

The pressure-reducing and desuperheating station in Fig. 11–15 is designed to take 410 psi 625 F steam and pass it into a low-pressure header at 120 psi and 380 F. The equipment provides accurate temperature control over the entire range of flow with complete evaporation of desuperheating water.

Each reducing valve is controlled by a regulator responsive to steam pressure at a control point in the low-pressure header some distance downstream from the desuperheater. Each reducing-valve regulator also operates a water shutoff valve in one of two parallel desuperheater water lines; the operation of each shutoff valve is synchronized with that of its respective reducing valve so the water will be shut off when no steam flows through the reducing valve.

The flow of water-atomizing steam is controlled by a restricting orifice that passes the required amount of high-pressure steam at a constant rate. Also, in the atomizing-steam line is an air-pressure-operated diaphragm-motor shutoff valve. It is held closed when both regulators hold their respective reducing valves closed.

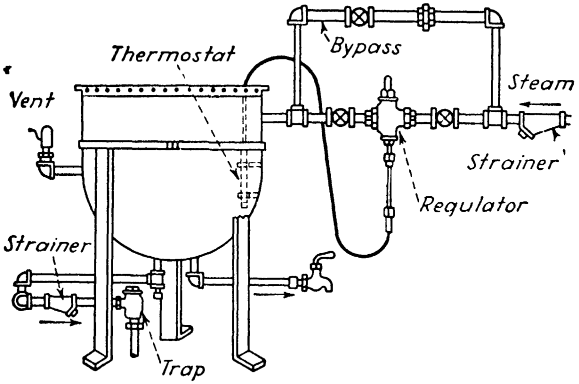


FIG. 11–14.—Regulating valve in steam line is actuated by thermal bulb inside the kettle. (*Courtesy of Sarco Co., Inc.*)

As soon as one of the regulators begins to operate its reducing valve, however, a control valve in the regulator releases the air pressure and allows the atomizing-steam shutoff valve to open.

The amount of moisture remaining in the paper or cloth during the finishing stages of its manufacture is determined by measuring the electrical resistance from a detector roll in direct contact with the sheet of material as it leaves the machine. The detector roll illustrated in Fig. 11–16 is mounted on a spring-loaded bracket and connected to a power amplifier that contains the measuring bridge. The instrument provides a continuous chart record of moisture content and can be equipped with contacts to operate signal lights for high and low moisture content. The bridge consists of resistors R_1, R_2, R_3, R_4, and the paper resistance in series with R_4. R_1 is a slide wire adjusted by the mechanical linkage of the recorder,

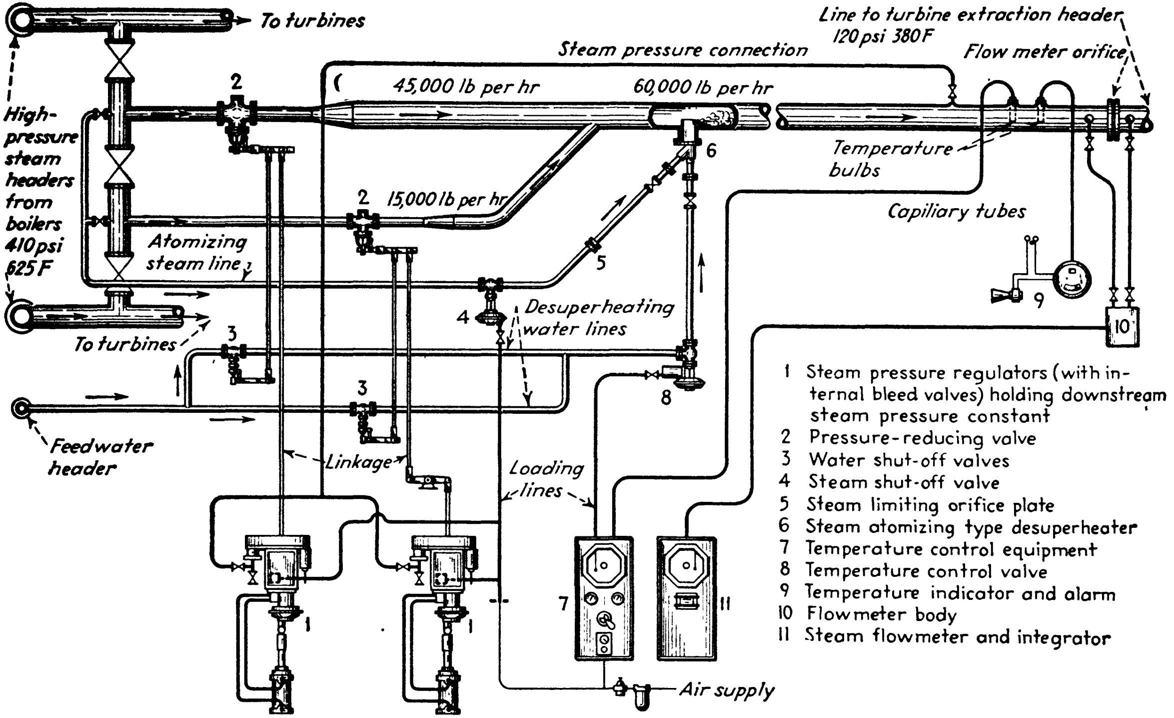


Fig. 11-15.—Steam pressure and temperature are reduced by this automatic desuperheating station. (*Courtesy of Republic Flow Meters Co.*)

which in turn is positioned by the galvanometer, thus keeping the bridge in balance for changing values of paper resistance.

The temperature of the rotating field of a generator can be determined by measuring its resistance with a Kelvin bridge (Fig. 11–17). In the simplified circuit (insert), lead *d* connects the field winding *X* and a low-resistance standard shunt *R* in series with the field supply circuit. Brush contacts on the collector rings, representing potential points at the terminals of the field winding, are connected to potential posts on the shunt, through fixed resistances *B, A* and *b, a* with a slide-wire resistance between *B* and *A* and another between *b* and *a*. The galvanometer terminals connect to terminals *e* and *f* on these slide wires. A condition of electrical balance is established by adjusting these contacts until the galvanometer deflection is reduced to zero. Resistances are then in the relation $X/R = B/A = b/a$.

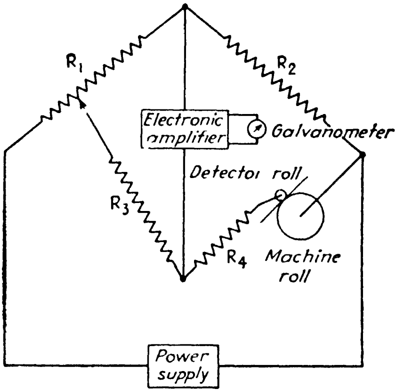


FIG. 11–16.—Electronic moisture detector gives constant record of moisture in paper and fabric. (*Courtesy of Brown Instrument Co.*)

Since the resistance of *R* is fixed, the ratio of *X* to *R* varies as the resistance of *X* changes with temperature, and the ratios of *B* to *A* and *b* to *a* must be adjusted accordingly to maintain electrical balance. A recorder automatically changes the setting of slide-wire contacts in the ratio circuits simultaneously and equally. This changes the ratio of *B* to *A* and of *b* to *a*, but *B* equals *b* and *A* equals *a* at any setting. The contact disk is turned until the ratios of *B* to *A* and of *b* to *a* equal the ratio of *X* to *R*, under which condition the galvanometer deflection is zero.

The control system in Fig. 11–18 is used to regulate the temperature of the plating-solution tanks. The controller maintains a fixed temperature in the circulating water rather than by direct control of the bath proper. The accumulator tank provides extra heat-

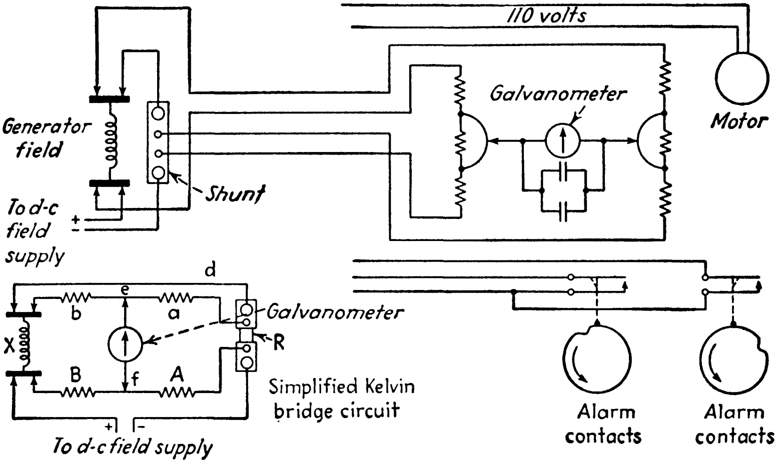


FIG. 11-17.—The temperature of a generator rotating field is determined by its resistance. (*Courtesy of Leeds & Northrup Co.*)

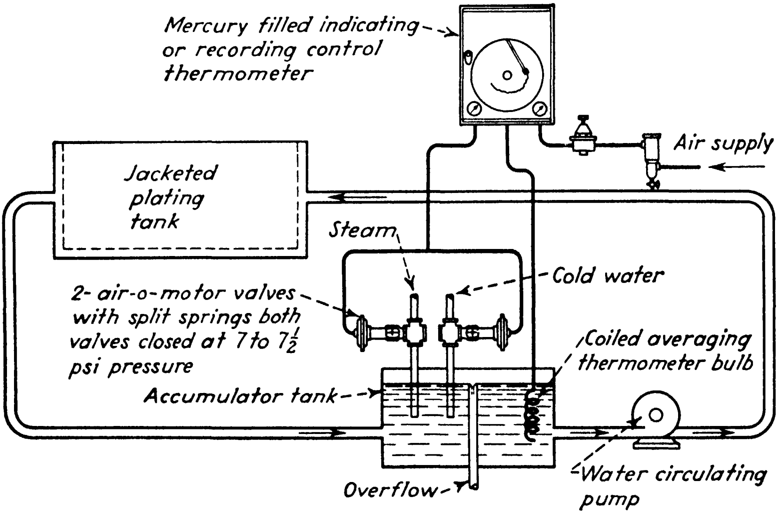


FIG. 11-18.—Mercury-filled thermometer regulates the temperature of a plating bath. (*Courtesy of Brown Instrument Co.*)

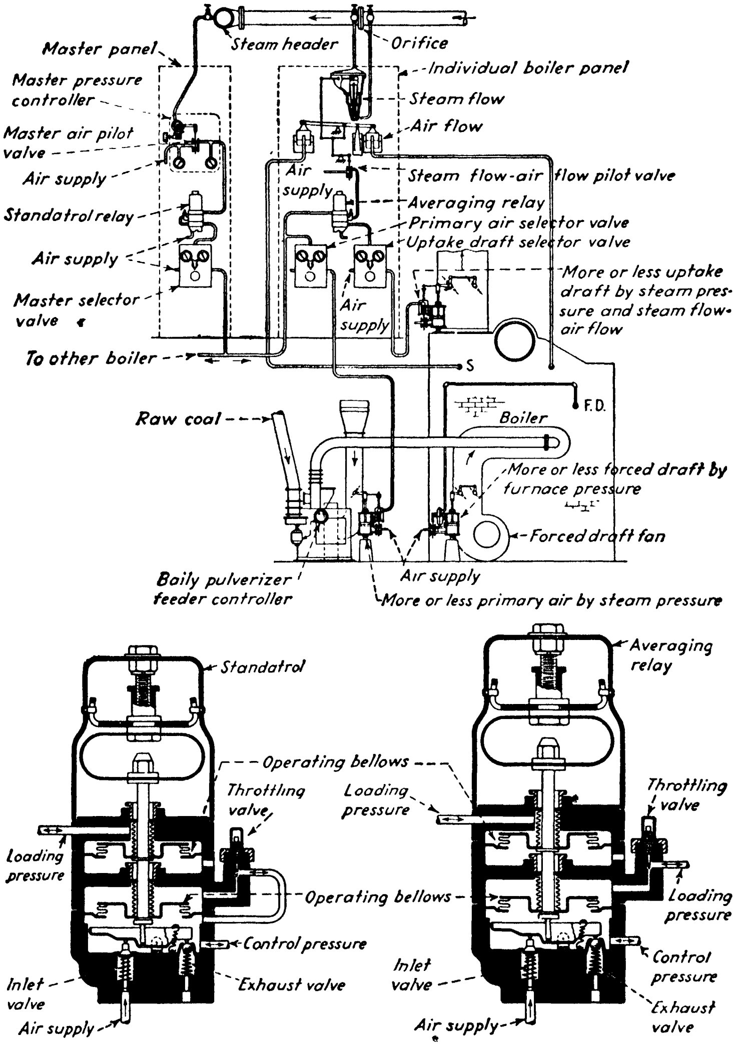


FIG. 11-19.—In a combustion-control system, master controller superimposes a loading pressure on the output from steam-flow-air-flow element to control uptake damper and coal feed. Forced draft is controlled by furnace pressure. (*Courtesy of Bailey Meter Co.*)

storage capacity, and the circulating pump ensures a rapid rate of heat transfer with a minimum temperature differential between the water and the bath by circulating the heating water at high velocity. The thermometer uses a long bulb to read average temperature of the water.

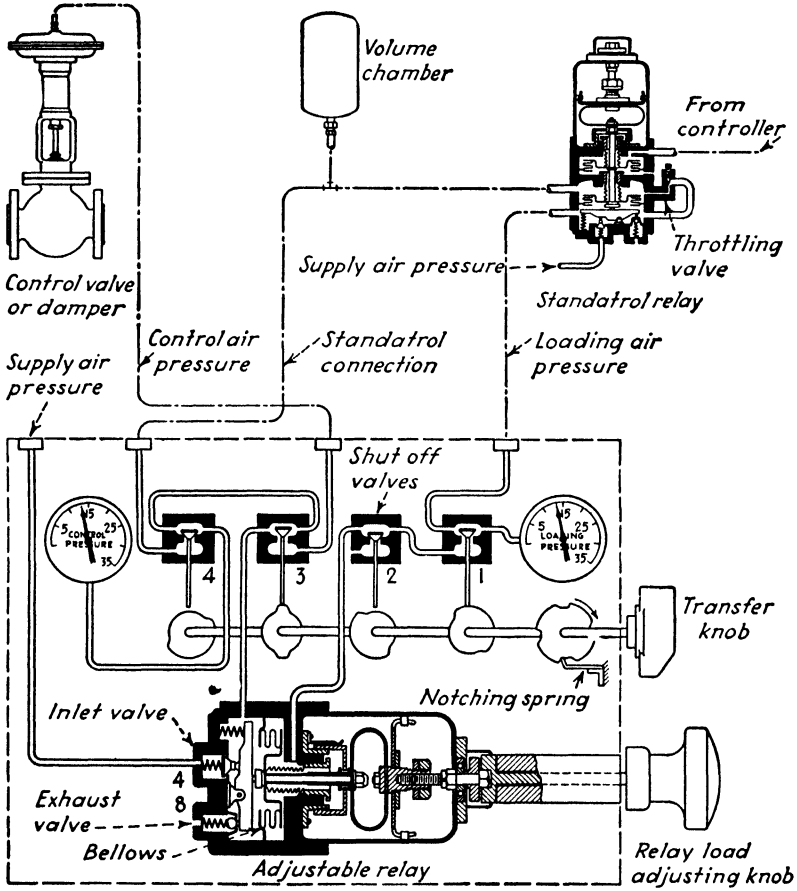


FIG. 11-20.—Schematic layout of the selector valve and Standatrol relay shown in Fig. 11-19.

Combustion control on boilers automatically maintains steam pressure by adjusting the fuel and air feed to the furnace in accordance with changes in steam demand. A book could be devoted to the subject of combustion control and the many systems available, but those illustrated here show the general working principle.

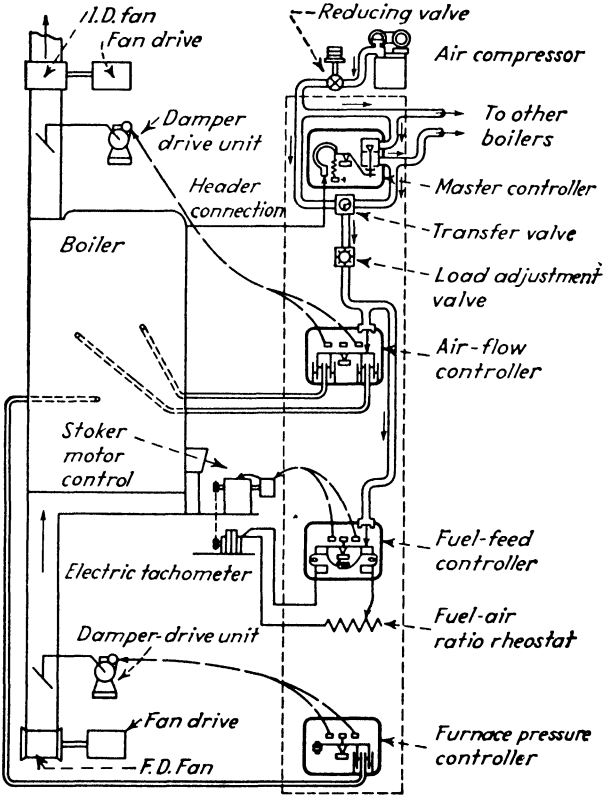


FIG. 11-21.—Master controller imposes pneumatic load on the air-flow and fuel-feed regulators in this combustion-control system. (*Courtesy of Leeds & Northrup Co.*)

Figure 11–19 is a system in which a master pressure controller (top left panel) connects to the main steam header and serves as a master unit over all boilers in the plant. In operation, if the steam pressure changes, the bourdon-tube element in the master controller adjusts its output loading air pressure, which is transmitted through the Standatrol relay and master selector valve to each individual boiler panel.

At the boiler panel, the loading pressure is directed through two different paths. One is to the averaging relay, through the uptake-draft selector valve and then to the uptake-draft control drive. The other path is through the primary air-selector valve to the primary air-control drive.

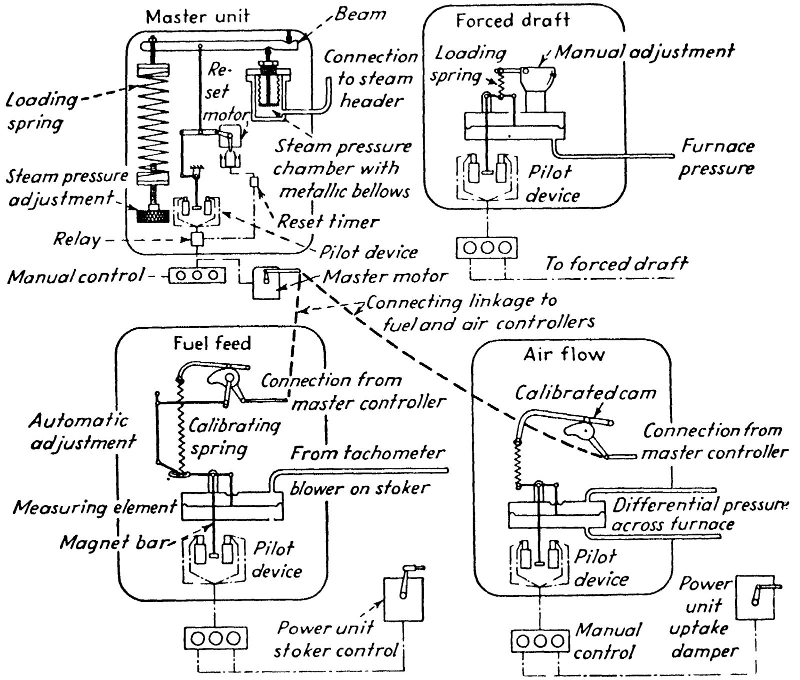


FIG. 11-22.—Master unit connects to the air-flow and fuel-feed regulators in this combustion-control system. (*Courtesy of The Hays Corp.*)

Control pressure is applied to a small metal bellows in each control drive. The bellows moves a relay pilot valve that controls air admission to a power piston that positions the damper. Thus the air pilot valve in the master controller sets up a loading pressure that positions the fuel- and air-control drives on all boilers in accordance with the steam demand as indicated by changes in steam pressure.

In conjunction with this, each boiler panel contains an air-flow–steam-flow instrument that operates an air pilot valve to set up a loading pressure for readjustment of the fuel-air ratio. This loading pressure is superimposed on the master loading pressure at the averaging relay. The resultant pressure from the averaging relay positions the uptake damper. The forced draft is controlled directly by furnace pressure acting on a spring-loaded diaphragm-operated air pilot valve. Selector valves, shown in schematic arrangement in Fig. 11–20, are provided to permit manual control of any part of the system. The Standatrol and averaging relay are shown in cross section in Fig. 11–19.

The combustion-control system illustrated in Fig. 11–21 uses a master controller that superimposes a loading pressure to unbalance the air-flow and fuel-feed controller on each boiler. The air-flow unit measures pressure differential in the furnace and operates the uptake damper. The fuel-feed controller is actuated from an electric tachometer on the stoker shaft and adjusts the stoker speed. The forced-draft damper is positioned by a furnace-pressure controller working independently of the other units.

The master unit for combustion control in Fig. 11–22 masters the operation of both fuel-feed and air-flow controllers. The forced-draft instrument operates on furnace pressure and controls the forced-draft damper.

CHAPTER XII

EMERGENCY INSTRUMENTS

The primary purpose of an emergency instrument[1] is to sound an alarm or shut down equipment upon some abnormal operation. In most instances, the same kind of device built for control service can be used for an emergency trip or alarm, but it is not advisable to use the same unit for both services. Independent connections from the emergency instrument to the system give added protection. Whether an operator or instrument controls the changes that occur in a process, the need for alarms and shutdown devices to protect against the failure of man or machine cannot be denied.

Assume a piston type of gas holder riding on a system between low-pressure blowers and high-pressure compressors. If the electric circuit to the blower motors trips, the high-pressure compressors soon draw the holder pressure dangerously low. Another instance: In a large three-electrode electric furnace, if oil circulation to the transformer stops, it is vital that the operator know this immediately. Since his reactions should not be relied upon, however, the furnace must be shut down automatically.

The same variables that we strive to control with instruments are the ones that may offer a hazard to life or equipment if they exceed a maximum or minimum value. We therefore deal with the following functions when selecting equipment for alarms: (1) high and low pressure, (2) high and low level, (3) high and low temperature, (4) high and low flow rate, (5) electric-circuit characteristics, (6) chemical change, and (7) movement.

The methods by which an operator is notified of an emergency condition are (1) audible and (2) visible. Experience has shown that, in some aggravated cases, it might be an excellent idea to employ a type of alarm acting upon the sense of touch. Distinctive-

---

[1] **Beard, C. S.**, Instrument Engineer, *Power*, May, June, 1946.

sounding horns or bells serve in locations having only a limited number of alarms, but it is difficult to obtain enough tones to differentiate among a large number. Consequently, the best practice is to attract the operator's attention with one horn or bell; he then refers to a panel board to identify the nature of the trouble.

The central control panel lends itself to a well-located and engineered signal system, because it provides for a substantial increase in desirable alarms as new functions are installed. The tendency to add alarms often leads to decentralization and a multiplicity of systems, which can only confuse the operators. On the other hand, the desirability cannot be overemphasized of centralizing identification mediums where the operator can immediately learn what is happening.

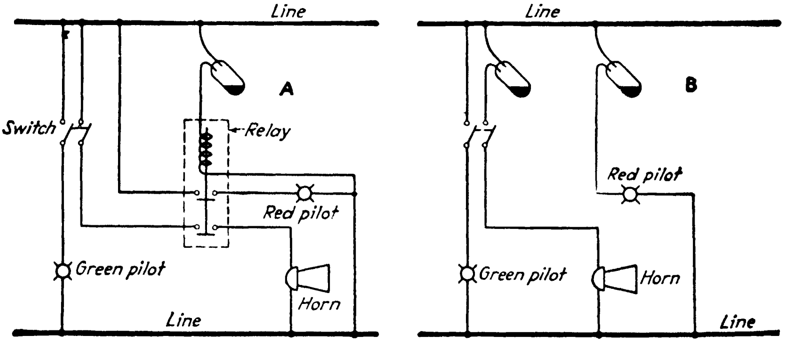


FIG. 12-1.—A switch and relay or two switches provide a visible and audible signal to warn of emergency.

If the signal board carries a single lamp for each indication, it is best to have the lamp lighted when the operating function is normal, and extinguished upon emergency. This method precludes the possibility that an emergency might occur without any evidence if a lamp on the alarm board burned out. Using two lamps for each alarm has the advantage of showing when the circuit is energized so that the tripping-control device lights a second lamp (Fig. 12-1*A* and *B*). In *A*, a mercoid switch controls a relay to operate the light and horn. When the mercury switch is open and the alarm circuit energized by closing the supply switch, a green lamp burns.

Abnormal operating conditions tilt the mercoid to energize an alarm circuit. The horn may be silenced by opening the supply switch, but a red light shows that the relay is still in the alarm position. The system in *B* is the same except that two mercoid switches operate simultaneously instead of the relay. Instead of lamps, an annunciator board can mount a series of tabs, which drop out to an open position when energized. These identify the instrument concerned.

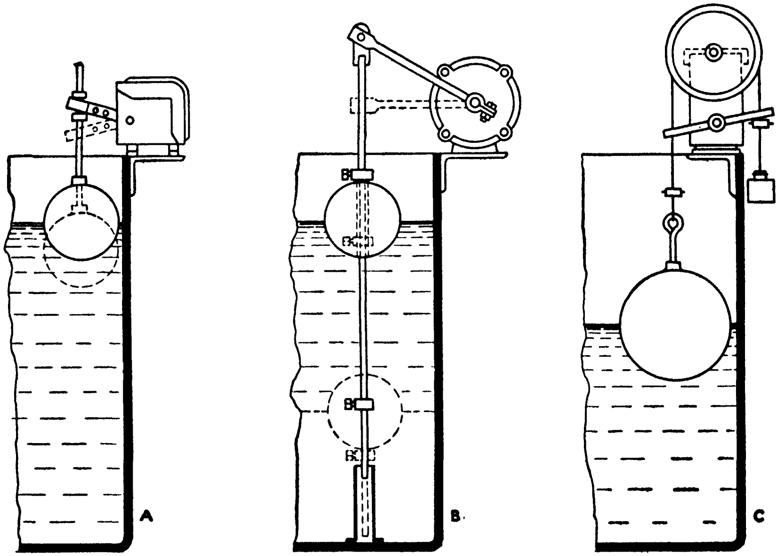


Fig. 12–2.—Floats can be mounted on a vertical rod, *A*, ride on the rod, *B*, or hang on a counterbalanced chain, *C*. (*Courtesy of Power.*)

Many valuable emergency devices, such as relief valves, explosion disks, and vacuum breakers, do not lend themselves to an identifying alarm; but, in practically every application, an instrument that alarms upon approach to the operating point of the relief can be installed. A study will determine whether it is desirable to have advance notice of this abnormal condition.

Relief of steam from a boiler through the safety valve is usually just an inefficiency, which can be corrected by adequate fuel control. On the other hand, one plant came close to disaster when a large gas holder was evacuated. It ruptured, and air was drawn into a system that contained a product readily oxidizable to the point of combustion.

Inasmuch as equipment that can be used for alarm service deals with the maximum and minimum limits of the variable, it is not necessary to incorporate the refinements essential to most controllers for throttling, or resetting, which is so vital to process control. Since the installation of shutdown and alarm equipment offers great possibilities for Rube Goldberg inventions, care must be exercised so that the expectancy of successful operation is not less than that of normal control.

For example, consider the protection of an ordinary tank. Selection must be based upon a determination of the circumstances if the tank runs dry, or overflows. An overflow line protects the surrounding equipment, but economics demands a notification of this so that the condition can be corrected. Low level can be just as critical as high. Practically all methods of determining liquid level may be adapted for alarms. This chapter discusses those most readily adaptable.

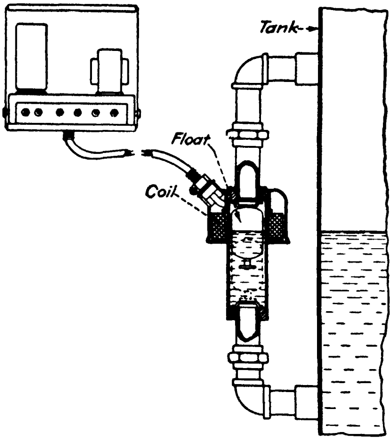


Fig. 12-3.—Movement of a float in and out of an inductive field operates alarms and shutdown devices.

Floats are the simplest devices for open tanks. They can be attached to the end of a rod as in Fig. 12-2, *A;* ride on a vertical rod, *B;* or hang on a counterbalanced chain, *C.* The switch may be a mercoid or knife type. It is neither good practice nor necessary to try to adapt the vertical-rod device to a totally closed tank. Numerous assemblies are available that fasten to the tank side. Here the float moves within the limits of the bracket because the holding tube is flexible. The switch is actuated by a rod inside the compressed tube.

Kidney floats (Fig. 12–3) offer the advantages of easy removal for inspection, less agitation at the liquid surface, being out of the way of mixers or agitating lines, and being installed on full tanks. In the inductive float installed as a kidney on a closed tank, movement of the float into the field of the coil operates the alarm system. Exercise care in selecting a float. Choose one that stands up best in the liquid, is large enough to operate the alarm, and has correctly designed shafting from inside to outside switch linkage.

Any of the accepted methods of level indication utilizing the liquid head pressure, either directly or as a differential, may actuate an alarm. These include the air diaphragm (Fig. 12–4) or the bubbler (Fig. 12–5). Changes of level vary the pressure applied to the actuating device.

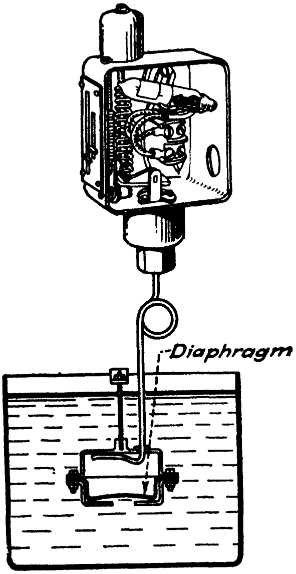


Fig. 12–4.—The air diaphragm can be arranged to operate mercury switches.

A differential-pressure instrument compensates for the vapor pressure in a closed vessel because the connection above the liquid transmits the vapor pressure, while the line at the desired depth in the liquid transmits a total pressure equal to the vapor, plus the liquid head pressure. The difference is the liquid-column weight, which may be corrected to linear units for the kind of liquid in the tank. Every differential gage may be used as an alarm, either by using the indicator shaft to actuate a switch or by completing the circuit through contacts in the mercury of a mercury-filled gage. Where hazardous vapors exist, the pneumatic flow controller may operate a remote switch advantageously.

Many instruments use the electrical conductivity of a liquid to complete a circuit or short-out a coil (Fig. 12–6). When selecting these instruments, take full advantage of the manufacturer's experience relative to a liquid's conductivity. Its chemical characteristics and inflammability determine the circuit and type of electrode.

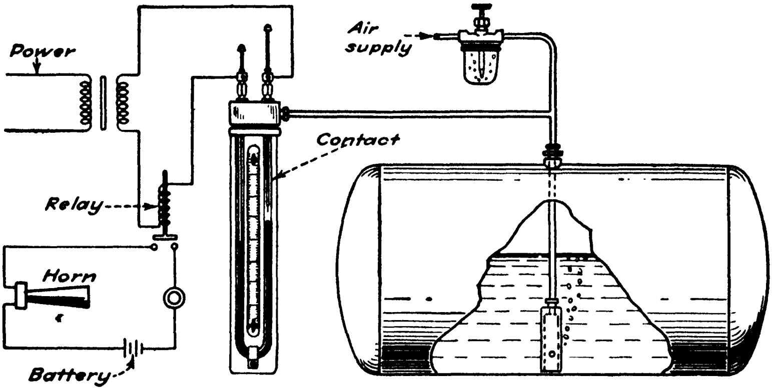


FIG. 12-5.—Contacts in a mercury U tube actuated by a bubbler device ring alarms. (*Courtesy of Power.*)

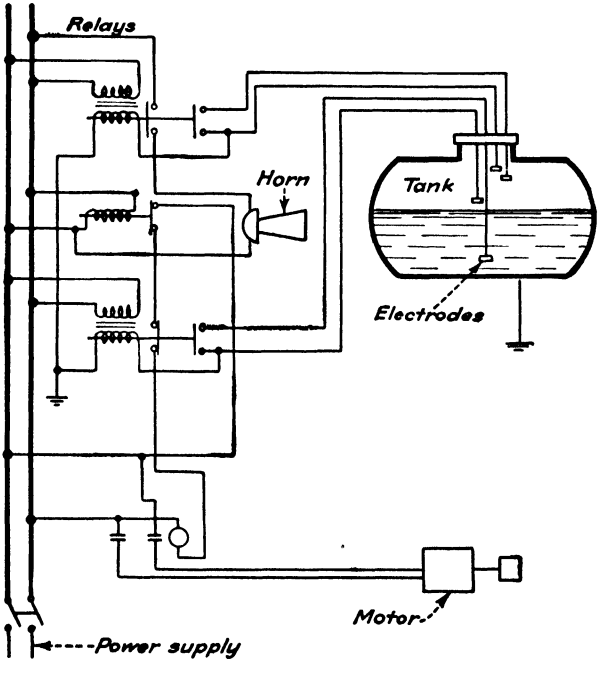


FIG. 12-6.—An electrode device uses the liquid for the electrolyte.

Any accepted method for indicating pressure may be used as an alarm or shutdown. The most widely used device, the bourdon

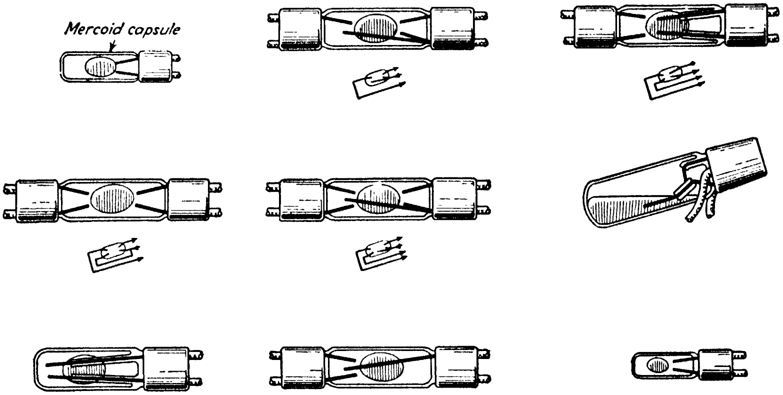


FIG. 12-7.—Mercury switches come in many sizes and forms.

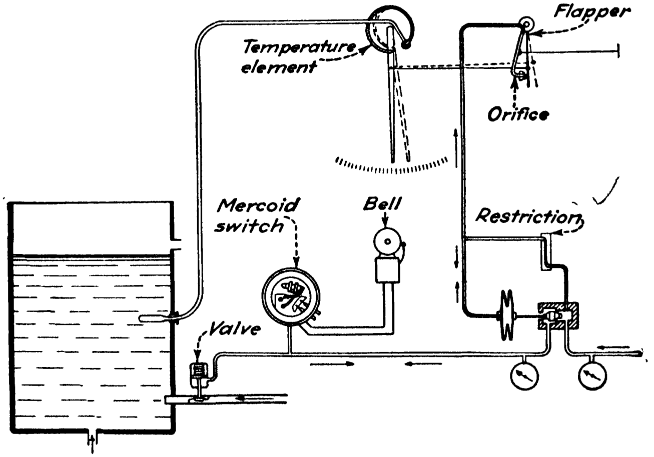


FIG. 12-8.—A pneumatically operated switch sets off an alarm when valve pressure varies beyond limits.

tube, is readily adaptable, because the same movement that rotates the pointer can operate either the flapper of a pneumatic controller or a mercoid switch. The latter consists of a glass or metal capsule enclosing electrodes in such a position that, upon tilting, a small amount of mercury shifts to make or break the circuit (Fig. 12–7). The variety available, ease of mounting, and safe construction make it one of the most widely used devices for controlling low-amperage electric circuits. When circuit amperage exceeds the rated capacity of the mercoid, it energizes a relay to do the heavy job. By proper linkage, rotation of the bourdon-actuated shaft also may be utilized to trip electric contactors.

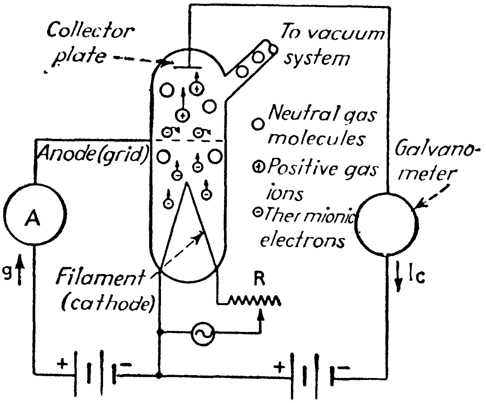


Fig. 12–9.—The ionization gage, used for extremely low pressures, can be equipped with an alarm. (*Courtesy of General Electric Co.*)

The movement of a pressure or temperature element may be connected to any of the various pneumatic controllers (Fig. 12–8). Here a mercoid switch is inserted to operate an alarm upon an increase to maximum, or a decrease to minimum, of the valve-diaphragm pressure, which varies with the controller's deviation from the control point. Pressure variation caused by restricting the nozzle may be used in any way desired.

For high pressures, the spring- or weight-loaded piston or diaphragm may be used for alarm or shutdown. Serving extensively for low pressures are the inverted-bell instrument, manometer with a float riding on the liquid of one leg, and limp-diaphragm gage or bellows.

Many instruments that indicate low absolute pressures are difficult to convert into alarms or shutdown devices. Some of these are the McLeod or Dubrovin gages and micromanometers. When the pressure is low enough to permit using a Pirani (Fig. 2-35) or an ionization gage (Fig. 12-9) or others that deliver even a small voltage, it is not difficult to use a contactor on the millivoltmeter or potentiometer.

Exercise extreme care in locating the sensitive element of a temperature alarm to be sure the response is a true indication of emergency conditions. An excellent example is insertion of a thermal element, to guard against overheating a steam-heated exchanger, so far away in the discharge line that discontinuance of flow allows the line to cool rather than heat up excessively as liquid in the heater does (Fig. 12-10). The element must be in the liquid section to be protected. This condition becomes critical in direct-fired equipment.

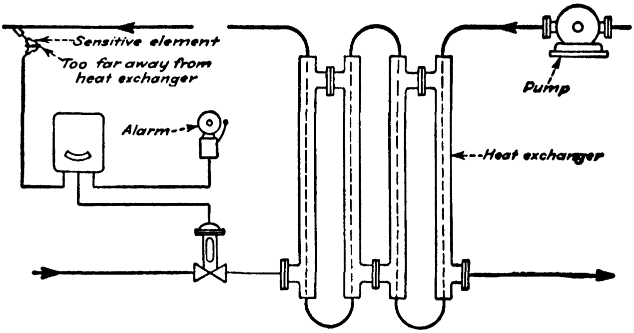


FIG. 12-10.—Place the protective thermal element where overheating may take place. (*Courtesy of Power.*)

The same rules that apply to thermal-element installation for controllers apply to alarms. Make every effort to have the element give rapid reaction, even to the point of eliminating the thermometer well when possible. Means of physically strengthening the element must then be used to guard against erosion.

Distension of a bimetallic strip, or helical spring, from temperature changes may operate an alarm. Although it is not considered a precision method, low-melting metals may be applied in ways similar to their use in automatic-fire systems.

Temperature is measured by a galvanometer-type pyrometer, potentiometer, or resistance thermometer, and it is possible to utilize their movement relative to the temperature that is being indicated for operating the alarm (Fig. 12–11). It may be rotation of the slide-wire contactor or pen-arm shaft or linear movement of the indicator. Pneumatic temperature controllers, like those for pressure control, are readily adaptable as alarms. The radiation pyrometer may operate high-temperature alarms by applying the signaling device to the indicating or recording instrument.

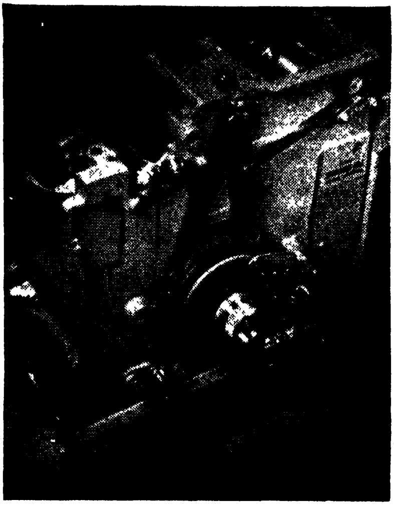


FIG. 12–11.—Potentiometers used for temperature and pH measurement can be fitted with alarm contacts.

In flow control, the most frequent need is for an alarm to show that flow has decreased below a desired minimum. Applications include cooling-jacket water for engines, cooling oil in transformers, air flow to soaking or annealing furnaces, and air flow for ventilating vaults and shafts. A large percentage of measuring devices, especially recorders or those having indicating pointers, may be adapted for an alarm. The integrating unit, such as a displacement meter, does not adapt itself so readily.

For lines small enough to make a pipe-size fitting economical, the area meter is particularly good (Fig. 12–12). The float movement can operate a contact electrically, magnetically, or mechanically. Here a permanent magnet attached to the float moves within close proximity to, and operates, a mercoid switch or contactor. It is possible to set this instrument quite accurately for very low or zero flow of fluid or gas.

The movement of a check valve can be made to energize an alarm, either by using a shaft from the clapper to a contactor outside the fitting, or by transmitting the clapper movement with a magnet mounted on it to a counterpart outside the fitting.

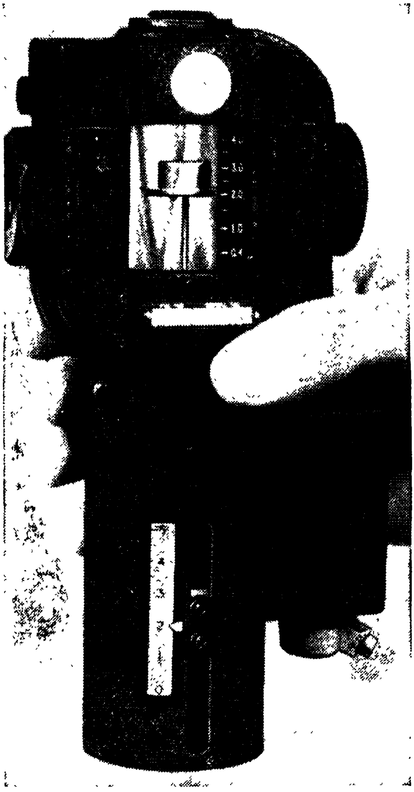


FIG. 12-12.—A flowmeter can operate a contactor either magnetically or mechanically.

Flame-failure devices used extensively on stills and direct-fired heaters employ (1) current flow through the flame or (2) an electric eye.

In Fig. 12-13, the control current flows from an electrode in the main or pilot burner through the ionized flame to the metal parts. Flame extinguishing from any cause opens the circuit immediately. Because ionization, caused by combustion, closes the circuit, alarm action after flame failure does not wait until the electrode cools, a delay common in many less critical bimetallic alarms and shut-offs. The system is not applicable to liquid fuels, which might themselves complete the circuit.

The electric eye (Fig. 12-14) serves for oils and pulverized coal. This instrument focuses a photoelectric tube on the burning fuel or, for gas firing, on an incandescent element in the flame. This control may be set up to alarm simply upon flame failure or to evacuate the firebox, relight the pilot, and relight the main flame. The speed with which an explosive mixture can accumulate in a firebox is often faster than that at which the firebrick loses its incandescence, and the relatively small investment in flame-failure protection can save thousands of dollars of damage and decrease the danger to personnel.

Among the many applications where continued movement of equipment is critical to personnel or process are (1) an endless belt transporting a product from a hopper to a kiln and (2) a fan or blower. It may be simpler to signal rotation of the machinery than to detect and sound an alarm for a change in effect, such as differential-pressure loss across a drier or accumulation of dangerous gas.

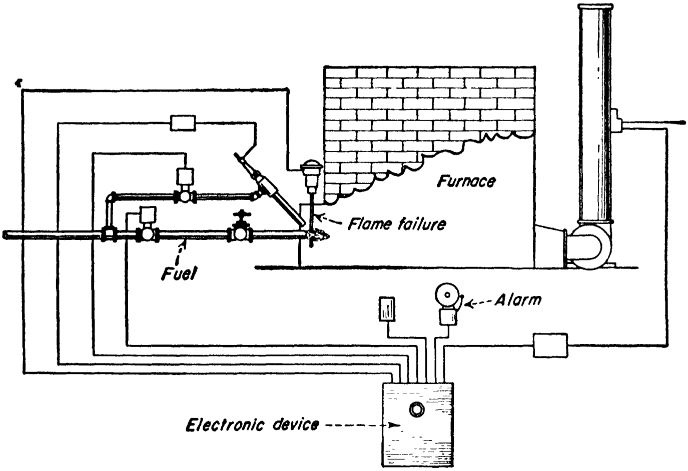


FIG. 12-13.—Ionized gas in the burner flame completes a restraining circuit in the alarm.

One instrument, a plugging switch, acts like a governor to make or break contact at a critical speed. Contact with the moving equipment accomplishes the desired action (Fig. 12-15). This method is sometimes more positive than connecting an alarm on the electric circuit to the prime mover. The key in a blower shaft may shear off, the belt may break, or other accidents may occur that do not affect continuous movement of the prime mover but cause decided changes in the equipment.

Movement of a piston- or telescopic-type gas holder becomes hazardous when it approaches the filled or empty position. Here there is essentially no pressure change, and some other means of alarm must therefore be used. Trigger switches actuated by arms mounted on the moving part of the holder can be located to notify that limits are being approached.

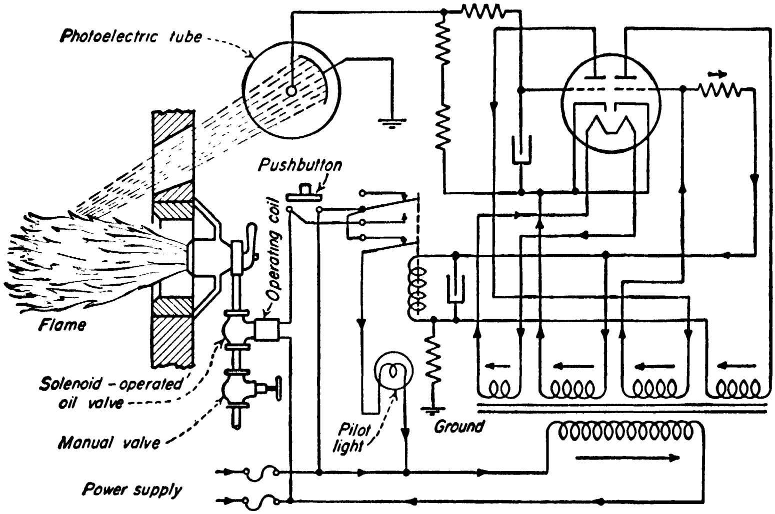


Fig. 12–14.—Flame-failure protection for an oil-fired furnace uses a photoelectric tube to watch the flame. (*Courtesy of Power.*)

A selsyn motor system consists of two motors having their rotors locked in phase with each other electrically. Thus movement of the transmitting unit causes the receiver to follow equally. The transmitter shaft can be rotated by mounting a wheel on it and allowing the wheel to make contact with the moving equipment, as when an electrode is fed to an electric furnace. Here the transmitting-motor speed must be such that some predetermined rotation, *e.g.*, 180 deg, is equivalent to the electrode length that can be fed before a new one must be inserted. Contactors on the receiver energize an alarm.

Probably the commonest instrument in chemical processes is that used for pH control. The complete unit, consisting of potentiometer and cells, is quite expensive and from an economic standpoint might justify deviating from the practice of using a separate instrument for the alarm. Alarms for pH are adaptable to such applications as waste-water and condensate treatment and chemical blending. Standard on-off electric units or pneumatic controls may be used as high- or low-limit switches.

Electrical-conductivity alarms are adaptations of the indicating or recording instruments employing conductivity cells as legs of a Wheatstone-bridge circuit—one leg containing the standard solution and the other the sample. This alarm is valuable in boiler-water control and to indicate possible leakage of an electrolyte into the condenser water.

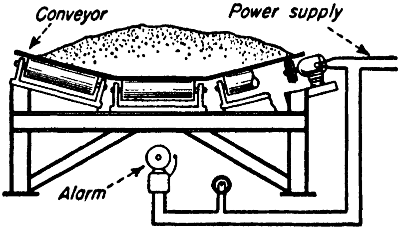


Fig. 12–15.—A flyball switch makes contact if speed rises or falls below normal.

Indication by thermal conductivity finds application when it is necessary to identify the presence of gas or limit its concentration in a gas mixture. In many industries, the atmosphere must be controlled or a warning sounded if it becomes toxic or explosive. A number of dependable combustible-gas alarms are available (Fig. 12–16). Samples may be drawn from various points into one instrument, and an alarm lamp can identify the location. Small amounts of carbon monoxide are readily measured by detecting the heat given off by the oxidation of iodine pentoxide, hopcalite, or other substances. The effect of the heat upon a thermopile may be calibrated to alarm at unsafe concentrations.

There is no one answer to every emergency shutdown- and alarm-instrument problem. The best solution is to select the simplest and most positive device that will do the job, because it is the last barrier between normal operation and possible destruction of equipment and loss of life.

A workable protective installation is not complete unless a procedure for the inspection and actual operation of each emergency instrument is set up in detail and an inspection schedule decided upon. Consider the importance of the operation protected and the possibility that the instrument may be thrown out of adjustment.

It is difficult to determine the necessary frequency of inspections, for the instrument could become inoperative from external causes immediately after testing. Experience shows that monthly coverage is about right for most commercial instruments, however.

When testing, simulate actual operating conditions; bring the emergency instrument to the alarm point by manipulation of the equipment being protected. Thus the jacket-water alarm on an engine may be energized by throttling the water flow so that the temperature rises. A valve on the discharge pump may be throttled to give the necessary abnormally high discharge pressure to operate the instrument. Where this procedure is detrimental to plant operation, set up a comparable set of conditions to energize the instrument artificially. Carefully design the piping for this to minimize the time and labor needed to carry out the tests.

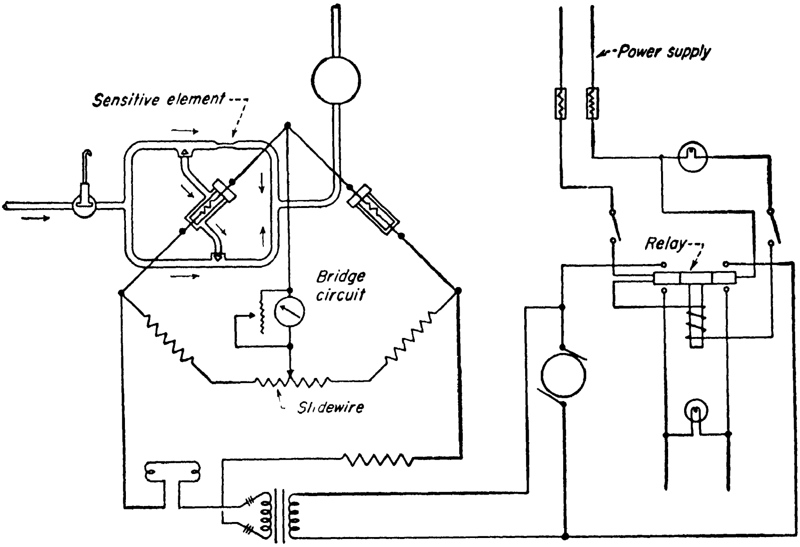


FIG. 12-16.—In a gas alarm, samples may be drawn into the instrument and an individual lamp used to designate the faulty location.

With proper cooperation from the departments concerned and thoughtful detailing of test equipment and procedures, one man can inspect a large number of instruments. He must have or acquire a working knowledge of instruments, safety valves, electrical equipment, and plant processes. The organization and size of the plant determine whether (1) separate departments handle the different classes of instruments, (2) the instrument department handles safety valves, (3) the electrical department handles instruments, or (4) some other possible combination is in effect.

In any event, the inspector should not be a maintenance repairman in one of the departments, because he must have authority to report directly to management, sending copies of his report to the head of each maintenance group. An item on this report showing improper operation of any emergency device must be considered an order to the department concerned to remedy the trouble immediately.

CHAPTER XIII

FEED-WATER CONTROL

Automatic feed-water control, a specific and familiar application of liquid-level control, promotes safety and increases steam-generating efficiency. To understand its value better, let us first see how water acts inside boilers.

A modern steam generator whose heating surface consists largely of water walls, economizers, superheaters and reheaters has less than one-half as much water storage in the drum as the average boiler of 25 years ago. Today's unit can evaporate the drum empty in a few minutes if the feed-water supply stops during heavy loads.

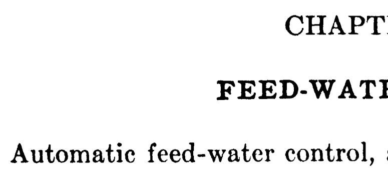


FIG. 13–1.—Swell effects in a cross-drum water-wall boiler containing about 86,000 lb of water. (*Courtesy of Bailey Meter Co.*)

When a boiler produces steam, the density of the circulating water is reduced by both its heat and the vapor bubbles entrained in it. Therefore, as the steaming rate rises, a fixed weight of water has increasing volume, or *swell*. During high steaming rates, a large percentage of boiler volume below the water line is occupied by steam bubbles; a 26,000 sq ft cross-drum boiler equipped with water walls showed 24 per cent swell when delivering 500,000 lb per hr (Fig. 13–1). Because the drum normally holds only 10 to 20 per cent of the total boiler water, any change in steaming rate seriously affects the level. For instance, a sudden increased load causes a slight drop in boiler pressure, which lowers the boiling point of the water and results in increased steam generation from heat already stored in the water. The water level then rises momentarily because of the increased volume of steam bubbles below the surface. Similarly, a sudden decrease in load momentarily raises the pressure and boiling point, reduces steam-bubble volume, and lowers the water level.

Swell is more important at relatively low pressures than at higher pressures, because 1 lb of low-pressure steam occupies a much greater volume than 1 lb of water at the same temperature (Fig. 13-2*A*). It is more difficult to stabilize the level in a boiler operating at 400 psi than in one of similar design operating at 1,500 psi, because, as the critical pressure of steam (3,226 psi abs) is neared, the specific volume of the steam approaches that of water, and swell is considerably less.

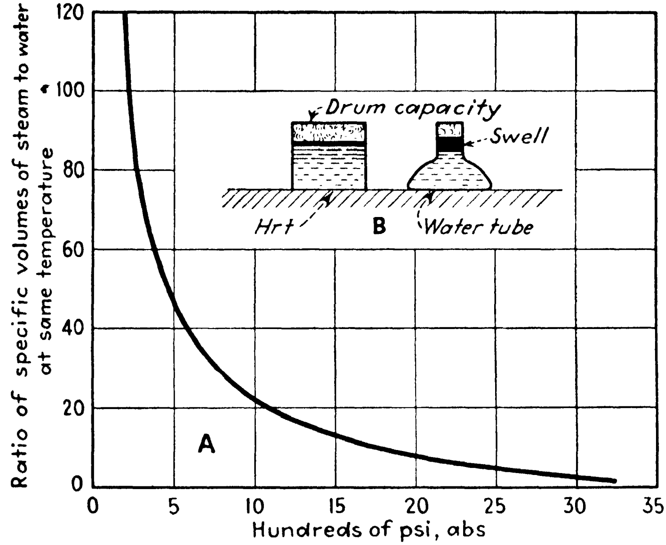


FIG. 13-2.—Swell becomes less as pressure increases *A*, because steam and water approach equality in their specific volumes. (*Courtesy of Northern Equipment Co.*)

A boiler that has large heat-absorption area in tubes and water walls, with small drum storage, is more affected by swell than is a fire-tube unit (Fig. 13-2*B*). Heavy concentrations of solids and impurities cause a boiler to foam, which may also raise the water level high enough to prime or carry over into the steam header.

This inherent characteristic (swell and shrink) materially assists in maintaining constant boiler pressure. On a sudden load increase, the water level rises, and the feed-water regulator temporarily reduces the lower temperature feed supply; this helps to increase evaporation. On sudden load decrease, the water level falls, and the regulator increases the feed; this helps to lower the boiler pressure. The temperature of the feed water in comparison with that of the water in the boiler and the rate at which water enters the boiler reduces swell to a greater or less degree.

Although the effects of swell cannot be eliminated entirely, best results sometimes call for regulation with a *rising characteristic* (rising level with increasing load). Carrying low water at light loads avoids the danger of carry-over from swell during sudden load increases. Conversely, it may be desirable to carry high water on heavy loads to avoid exposing the tube area from water shrink during a sharp load drop.

The boiler-water level is always higher than is shown by the gage glass. The glass is merely one leg of a U tube, and the boiler itself constitutes the other leg. The U-tube levels balance as far as the unit weight of the two legs is concerned, but their relative height depends on their relative density. For instance, suppose you half fill a U tube with water and then pour a lighter liquid into one side. The difference in the height of the two columns is considerable, and it increases with an increase in tube length.

Since the gage glass contains water free of steam bubbles at a temperature considerably lower than that in the boiler, this water is heavier and can balance a higher column of water in the drum. Should the glass connect directly to the drum, there would be no appreciable density difference (from temperature) because the gage water would be practically as hot as that in the boiler.

Consider a boiler with a water connection to its gage glass 30 in. below normal water level. The water column is piped out several inches away from the boiler, where its temperature is about 70 F lower than that of the boiler water. The coefficient of expansion of the water within this range is approximately 0.000643. Multiplying this by 70 (number of degrees difference) and then by 30 (water-connection length) equals 1.35 in. Allowing 10 per cent for steam bubbles gives 3 in.; 3 + 1.35 equals 4.35 in.—the difference between the boiler and the gage-glass levels. Another discrepancy in gage-glass indication comes from uneven circulation in water-tube-boiler downcomers that leave opposite ends of the drum. A glass installed on one end of the drum does not give a true picture of level conditions across it.

Any action inside the boiler that causes untrue gage-glass indications also affects an automatic feed-water regulator the same way. For instance, if the gage glass shows half full and the boiler-water level is actually several inches higher, causing moisture carry-over, an automatic regulator adjusted to operate at this point permits the boiler to discharge wet steam continuously. The regulator can feed enough water to maintain the level for which it is set, but it has no way of determining whether this setting is too high or too low.

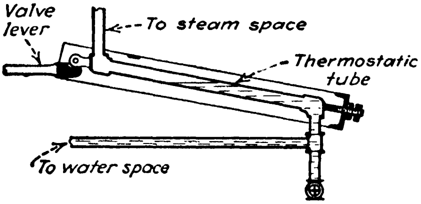


FIG. 13-3.—Thermostatic-tube regulator mounts alongside the boiler so that water level cuts across tube near its mid-point. (*Courtesy of Northern Equipment Co.*)

Simple, rugged, reliable, feed-water regulators are capable of adjusting the feed with minimum time lag. Many of the previously described primary elements (Chap. I, II, III) serve admirably for this purpose. Additional ones include (1) thermostatic tube, (2) steam generator, and (3) varying-pressure chamber. The control can be one-element—in which the feed valve is controlled by water level only; two-element, with both water level and steam flow acting to control the feed valve; or three-element, in which steam flow, feed-water flow, and water level act to control the feed valve.

One-element Regulators.—Thermostatic-inclined tubes (Figs. 13–3, 13–4, and 13–5), usually of brass or other alloy with a high coefficient of expansion, are mounted alongside the boiler so that the water level cuts across the tube near the center of its working length. Force made available by expansion of the metal is limited only by its elastic limit. The lower end of the tube is anchored and connects by piping to the boiler below its water level; the upper end is free to move and connects through a long-radius pipe bend to the boiler-steam space. An insulated steam lead properly sloped toward the boiler minimizes condensate flow into the tube. A shutoff valve in the water line prevents emptying the entire line when blowing down the tube.

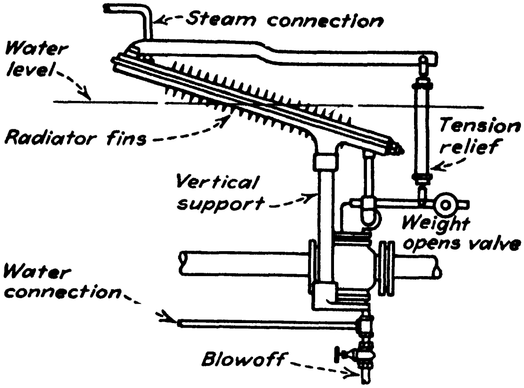


FIG. 13-4.—Radiating fins attached to the water end of thermostatic tube accelerate cooling. (*Courtesy of Henszey Co.*)

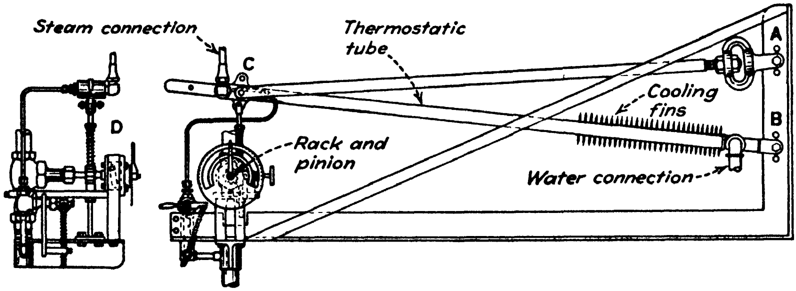


FIG. 13-5.—Pivoting the thermostatic tube's free end to produce a rigid triangle multiplies tube expansion. (*Courtesy of J. A. Campbell Co.*)

When the boiler-water level changes, the tube-water level moves the same vertical distance; but lengthwise change in the tube level depends on tube slope. This water does not circulate; therefore, it cools off by radiating heat to the atmosphere. Some tubes (Figs. 13-4 and 13-5) carry radiating fins on their water end to accelerate cooling. Since the steam end of the tube is always hot, any change in water level either reduces or increases the surface with which hot steam is in contact, thereby contracting or expanding the tube. To complete the regulator, the free end of the tube connects through suitable linkage to a relay, or regulating valve, in the feed-water line.

The design in Fig. 13–5 multiplies expansion by pivoting the lower end of the tube to a vertical frame member that carries a pivoted rigid bar that connects to the free end of the tube. Pivoting points A and B are 7 in. apart and produce a rigid triangle. Since the ratio of the length of the long sides to the short is 6:1, a change in tube length raises or lowers tube end C approximately six times the amount of that change. The free end of the tube connects to the rotating stem of the control valve through rack and pinion D. A small pipe between the feed-water line and the steam end of the tube permits filling it with relatively cool feed water after blowing it down.

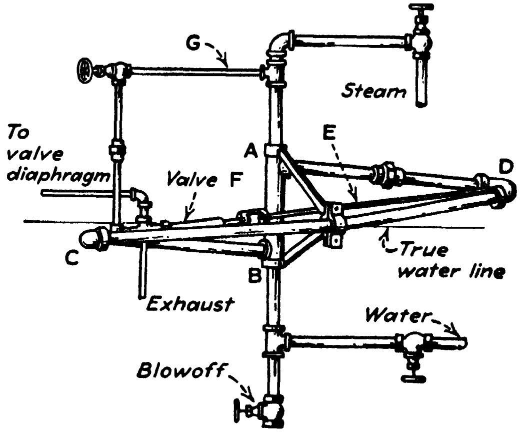


FIG. 13–6.—Harp-shaped thermostat tube uses boiler pressure to operate diaphragm-motor feed valve. (*Courtesy of the C. E. Squires Co.*)

The harp design in Fig. 13–6 utilizes boiler pressure to operate a diaphragm-motor feed valve. The top and bottom of column AB are connected by harp-shaped copper tubes with their extreme ends C and D in a straight line. Expansion and contraction concentrate at the tube ends and act on rod *E* and valve stem *F*. Valve *F* regulates steam flow from pipe *G* to the regulating-valve diaphragm. (Painting reduces the ability of a thermostatic tube to radiate heat and makes it sluggish. Better operating results come from cleaning the surface occasionally with fine emery cloth.)

The most important tube function is to prevent low water; therefore, reserve movement for opening the valve should be about twice that needed for closing. If the regulator sets too high, the tube is filled mostly with steam and is expanded almost to its maximum so that it has no reserve movement to open the valve should the level drop. If the regulator sets too low, the reverse is true, and the tube is so nearly contracted that it has no reserve for closing the valve should the level rise.

Inconsistencies in thermostatic-tube operation result from two causes: (1) Hot condensate from the steam connection runs down and warms the lower end of the tube. (2) The cool-water connection, after being blown down, is refilled with hot boiler water. With all the cool-water reserve blown out, hot water equalizing back from the boiler expands the tube and momentarily opens the feed-water valve.

The steam-generator regulator (Fig. 13–7) consists essentially of two tubes, an outer one, *A* (insert), fitted with cooling fins, and an inner one, *B*, which serves as a heating element, connecting to the boiler drum above and below the water level. The inner tube then contains steam and water, the water level being the same as that in the boiler drum. The annular space between the inner and outer tubes forms a closed system with the connecting copper tubing and metal bellows of the regulating valve, which is filled with water.

Volumes in the closed system are so proportioned that, when steam completely fills the annular space, the regulating valve is wide open, but, when water fills this space, the valve is closed. Normally these extremes are not reached, and a continuous feed is delivered with the water level near the center of the tube.

The generator shell has two openings, one at the bottom, *C*, that connects to a copper tube leading to the regulating valve, and the other at the top, *D*, for filling the system with water. This opening is placed on the side of the expansion head to prevent overfilling the generator.

The water leg from the boiler carries cooling fins to ensure a supply of relatively cool water at the bottom of the inner generator tube. When the boiler-water level rises, this cool water passes up into the tube and helps to condense the steam in the annular space of the outer tube.

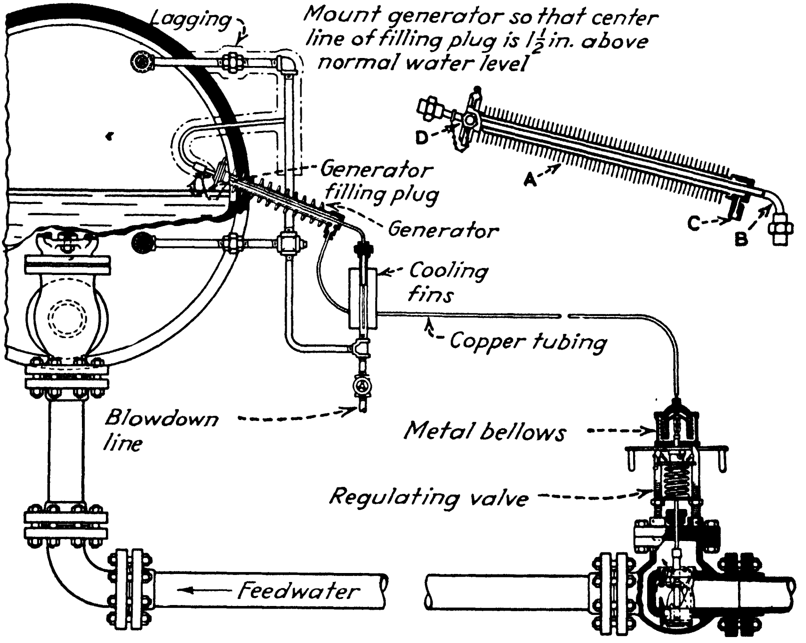


FIG. 13–7.—Steam-generator regulator consists of two tubes. Inner tube acts as heating element; outer one generates pressure to operate feed valve. (*Courtesy of Bailey Meter Co.*)

The varying-pressure-chamber unit in Fig. 13–8 operates on the theory that, when dry steam flows through chamber *F*, having an inlet orifice smaller than the outlet, the pressure in *F* is in proportion to that in *J* as the areas of the two orifices are in proportion to each other. If, however, the steam contains moisture that immediately flashes to steam, the pressure in *F* increases in proportion to the moisture content until hot water enters.

When a properly designed open-end pipe extends from chamber *F* down into the water column, dry steam enters the pipe when the water level is low, whereas water enters when the level is high. Pressure inside the pipe is lower than that in the water column because of constant discharge through *D*; consequently, the water level is higher inside the pipe than it is outside. Holes drilled

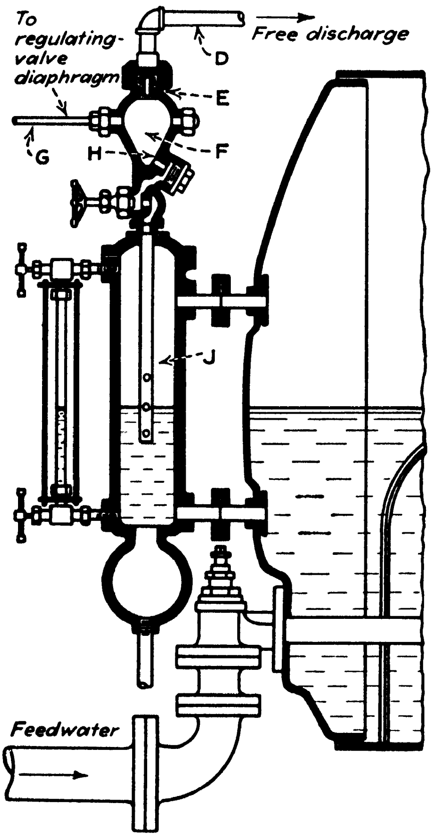


Fig. 13–8.—Quality of the mixture entering chamber *F* determines pressure at *G*. (*Courtesy of Atlas Valve Co.*)

through the pipe permit steam to mix with the water inside. This gives a gradual increase in the moisture content of the steam until the rising-water level covers all the holes. When the boiler level is low, all the holes in pipe *J* are exposed. Then only dry steam passes to chamber *F*; and, since pressure is not sufficient to close the diaphragm-operated valve connected to line *G*, feed water flows to the boiler.

As the water level increases, it rises in sampling pipe *J* and mingles with the steam entering through the upper holes, and a mixture of steam and water passes into chamber *F*. Part of the water flashes to steam, and the volume of the mixture increases. The frictional resistance of orifice *E* to its outflow also increases, building up pressure in chamber *F* until it positions the control valve properly.

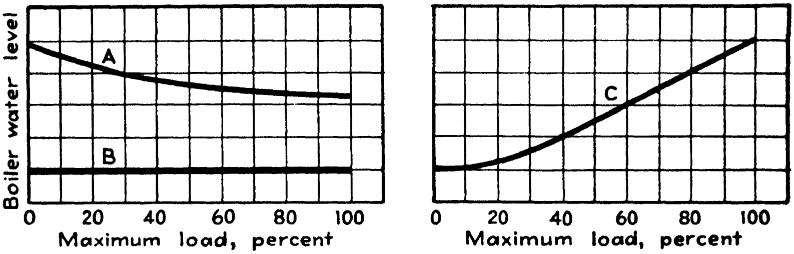


FIG. 13-9.—Liquid-level regulators are available that allow the level to fall as the load increases, maintain a constant level, or increase it with load.

One-element regulators (those operated only by the boiler-water level) provide continuous feed in proportion to steam output and hold the level steady so long as one constant load prevails. But, because they change the feed-valve position only with a change in drum level, the valve opening increases only when the water level falls. This produces a falling characteristic *A* (Fig. 13-9), carrying a lower level with each load increase. The original and simplest one-element regulator is a float operating in a chamber connected with the boiler drum. Any complete one-element system (Fig. 13-7) is entirely satisfactory for many applications, because it offers low first cost and greatest simplicity. For applications requiring

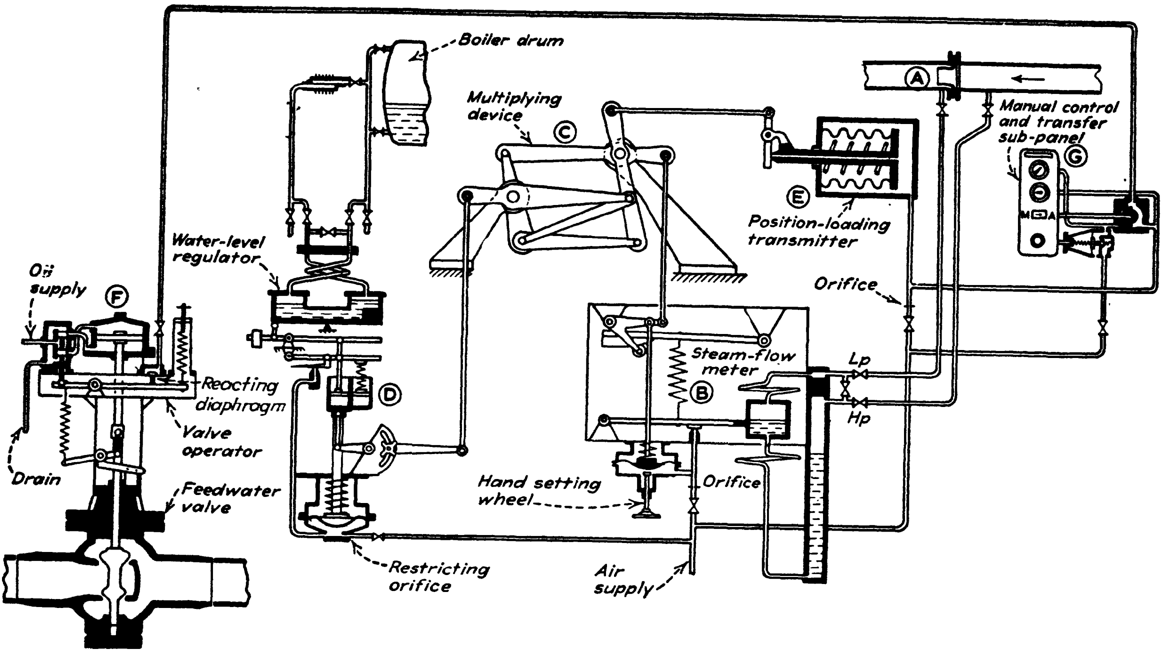


FIG. 13-10.—With the two-element regulator, primary control by steam flow anticipates the rate of change and stabilizes the level. Water-level element determines the stabilization point. (*Courtesy of Republic Flow Meters Co.*)

other than a falling characteristic, however, regulation must be under the control of more than one element.

Two-element Regulators.—With the steam-flow, or two-element, regulator (Fig. 13–10), primary control by steam-flow rate from the boiler anticipates the rate of change and stabilizes the level instantly. The water-level element determines at what drum level stabilization occurs. The feed-water valve is controlled jointly by both the water-level and the steam-flow elements. The steam-flow unit may be actuated by a pressure drop through the superheater (Fig. 13–11) or by an orifice placed in the main steam line (Fig. 13–10). Increased steam flow creates a greater pressure drop through the orifice or superheater, which causes the feed valve to open a definite distance. Because the steam-flow element can be adjusted for any predominance, it is capable of maintaining a constant level, regardless of load, or of giving a rising characteristic (*B* and *C*, Fig. 13–9).

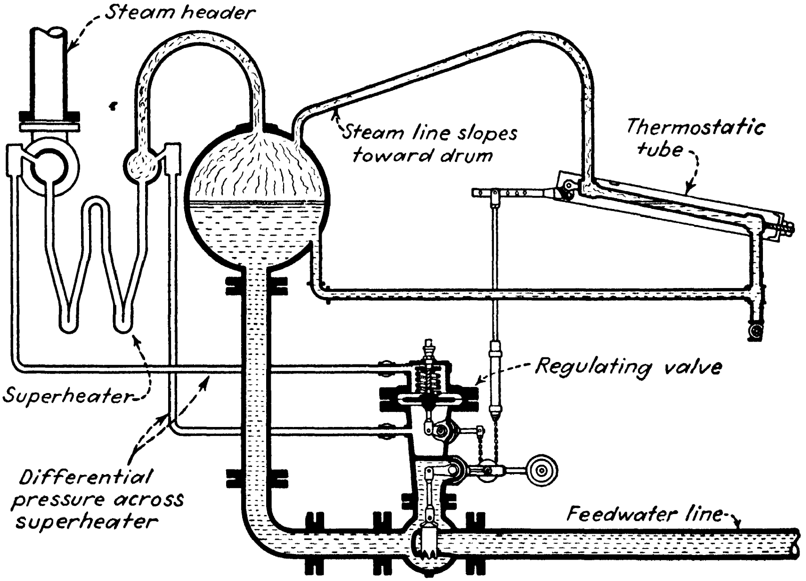


Fig. 13–11.—This two-element regulator measures steam flow by pressure drop across the superheater. (*Courtesy of Northern Equipment Co.*)

In Fig. 13–10, changes in flow through the steam nozzle *A* change the differential pressure on the mercury manometer at *B* to move the air-orifice throttling arm and position the diaphragm rod connected to multiplying device *C*. Level controller *D* positions its connecting rod to multiplying device *C*, and the joint control acts on nozzle-throttling device *E* to regulate the air pressure on the reaction diaphragm of regulating-valve positioner *F*. Change from automatic to manual control or vice versa is made by first adjusting *MA* regulating valve on control panel *G*. Multiplying device *C* can be adjusted to limit maximum water feed to the boiler when the nonreturn valve closes. Thus it controls the amount of water diverted from other boilers if a tube ruptures.

Three-element Regulators.—The balanced-flow, or three-element, regulator (Fig. 13–12) is a two-element unit to which has been added a third element—one responsive to the rate of feed-water flow. The water-flow element is similar to the steam-flow element but is actuated by a pressure drop through the feed-water heater or an orifice in the feed line. The predominant control is from steam flow.

Manometers *A* and *B* accurately measure steam output and water flow, respectively, and balance these measurements through the ratio beam that operates an air pilot valve. Control functions primarily from steam flow, modified slightly by water level when necessary to hold the level within desired limits. When the water input falls below or rises above the steam output, pilot valve *C* varies the air loading pressure beneath the relay upper diaphragm. As a final check, pilot valve *D* actuated by water level varies the air pressure above the relay upper diaphragm. The resultant of these two pressures moves the relay stem to operate the valve in the relay lower chamber that modifies the loading pressure to the feed-water regulating valve. Setting the link *E* at the midpoint of the ratio beam maintains a constant water level. Moving it to the left gives a rising characteristic with increasing load.

Feed-water-control systems that incorporate a relay between the regulator and the control valve relieve the regulator of moving the heavy valve and permit it to operate more sensitively and accurately. In general, relay operation is recommended when (1) boiler pressure exceeds 600 psi, (2) a 5-in. or larger feed valve operates with a normal pressure drop across it exceeding 75 psi, or (3) normal flow through the control valve exceeds 400,000 lb per hr. It is also recommended for any two-element regulator when pressure drop through the superheater at maximum load exceeds 50 psi, and for any three-element regulator.

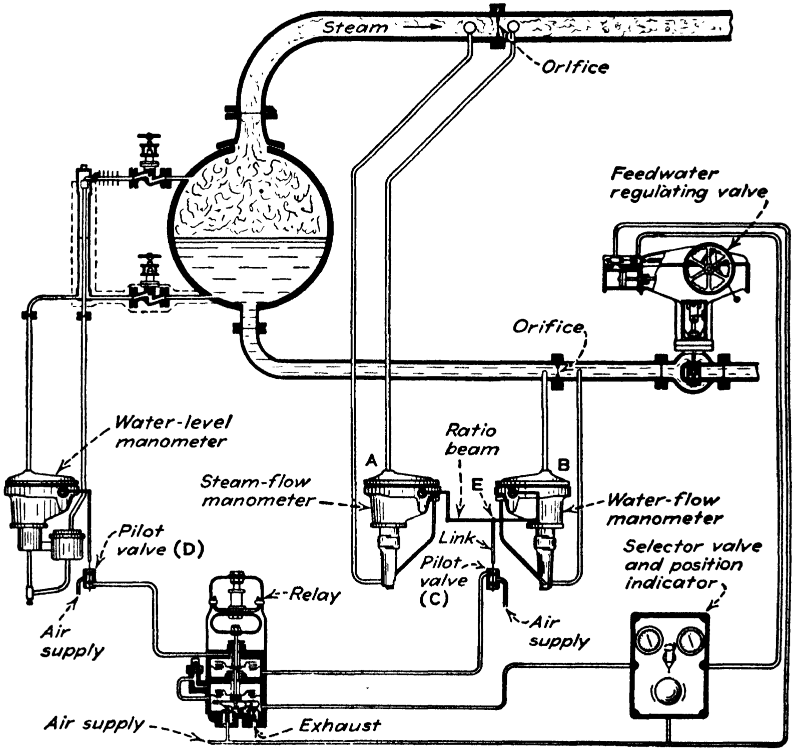


FIG. 13-12.—This three-element regulator measures pressure drop across orifices in steam and water lines. Elements operate common pilot valve in control line. (*Courtesy of Bailey Meter Co.*)

The rate of flow through a feed-water regulating valve depends primarily on (1) the area of the valve opening and (2) the pres-

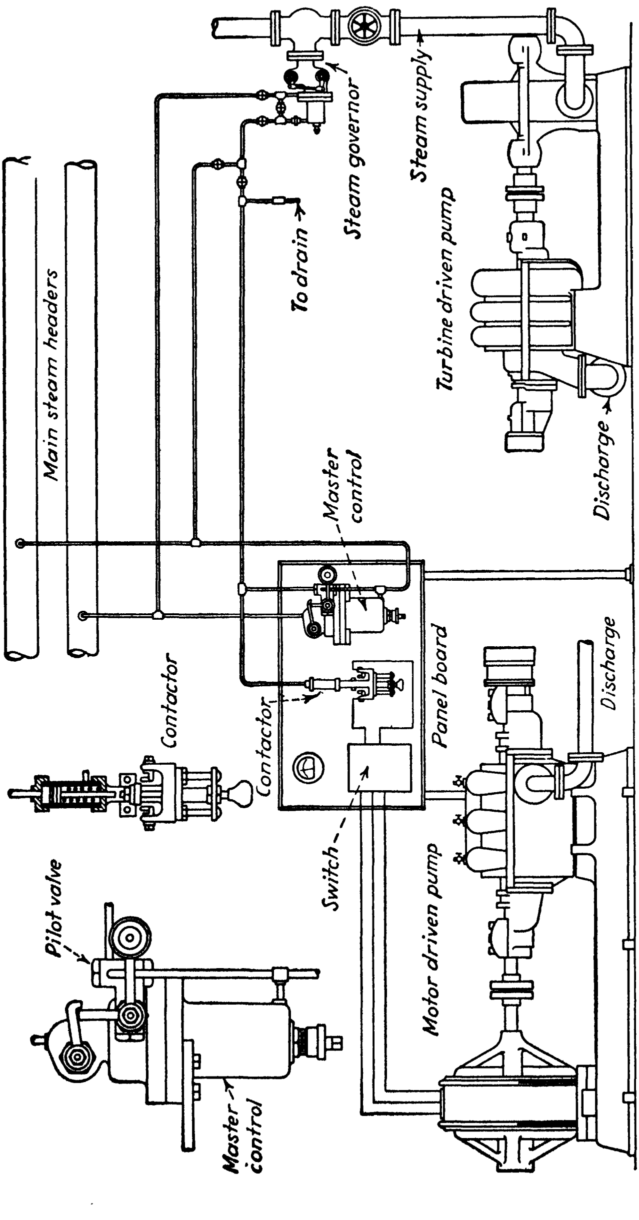


FIG. 13-13.—Control arrangement provides a predetermined excess pressure in feed line over that in the steam header and cuts in stand-by pumps. (*Courtesy of Northern Equipment Co.*)

sure drop across the valve. In a one- or two-element control (not involving a water-flow element), a certain feed demands a certain valve position. If the pressure drop across the valve is subject to wide variations, the feeding rate varies widely for a given valve opening.

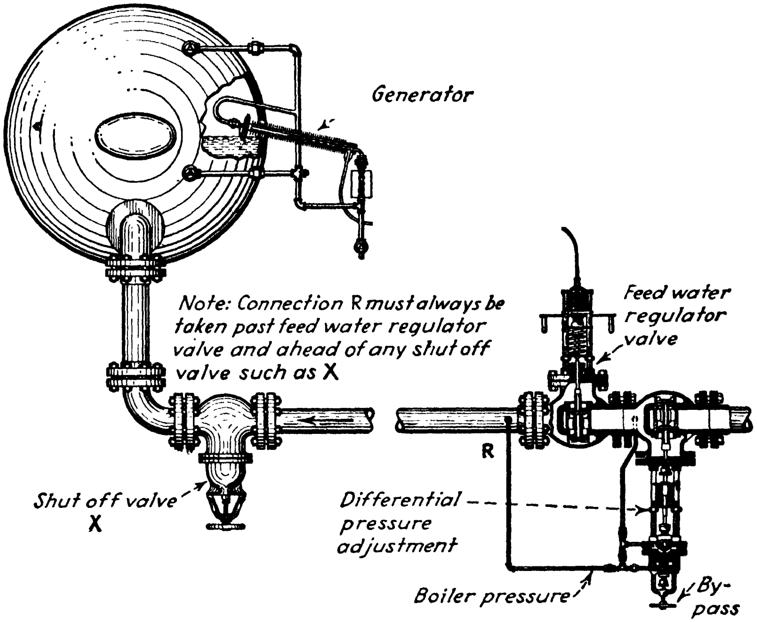


FIG. 13–14.—A sensitive differential-pressure valve placed ahead of the control valve maintains a constant pressure drop across the latter. (*Courtesy of Bailey Meter Co.*)

Pressure variations may be caused by pump characteristics, by faulty pump governors, or by friction loss in the feed line between the pumps and the boiler. If the pump pressure is maintained at a constant excess over the drum pressure (Fig. 13–13), the available pressure drop across the feed valve falls whenever the feeding rate increases. Constant pressure drop can be maintained by a differential-pressure control valve (Fig. 13–14) installed ahead of each feed-water regulating valve. This is simply a diaphragm- or bellows-operated valve actuated by the differential pressure across the main feed valve. With the pressure drop held constant, the flow is proportional to the valve opening, regardless of load changes.

One major advantage of automatic feed-water control is that it eliminates any tendency to over- and underfeed. A boiler can be made to reduce its steam output considerably if the feed valve is suddenly opened too wide. The effect of overfeeding on efficiency when several boilers operate in parallel is well known. The boiler produces less steam, and other units on the line must take up the load. With correct feed-water regulation, the load equalizes, and the tendency toward load swapping is largely eliminated. The steam pressure is more uniform, and the efficiency is higher.

CHAPTER XIV

MECHANICAL GOVERNORS[1]

For some reason, a steam-turbine governor is almost universally assumed to be a mysterious device, yet the fundamentals of turbine control are simple. It is only when the governing system is considered as a complete unit that its action becomes befuddling. So let us start at the beginning and look at only one element at a time.

Normally a governor has two main functions: The first is to respond to minute changes in speed. The speedometer is a governor that performs only this office. The second function is to transform changes in speed into proportionate steam flow, thus restraining the machine against overspeeding and preventing its stalling under load.

Fortunately, these two jobs have a common denominator—both depend upon speed or changes in speed. Here, then, is the governor's most fundamental characteristic. It must be sensitive to speed changes and must have some means of indicating them. Thus far the governor has turned out to be nothing more than a speedometer, and consequently it is incapable of doing anything to offset a speed change.

This brings up its second duty. The speed-change indication must be utilized to operate a valve or other control mechanism so that energy input to the turbine is readjusted to compensate exactly for a change in speed.

Remember the flyball governor on grandpa's threshing machine? That is probably the first and simplest practical steam governor, and for many modern turbine applications it is still adequate. It consists basically (Fig. 14–1) of two metal spheres connected by

---

[1] SCHWENDNER, A. F., What's Mysterious about a Governor's Action, and Choosing the Right Governor Isn't an Arbitrary Affair, *Power,* January and March, 1944.

straps to two collars on a shaft that is driven by the turbine. One collar, *A*, is fixed solidly to the shaft; the other, *B*, is free to slide up and down. Between them is a spring, tending to keep them apart. If the turbine gains speed, the flyballs move outward under centrifugal force and pull the movable collar, *B*, against the spring. Hence, a change in speed has been translated into a straight-line movement of the collar along the governor shaft. Connecting the collar through a mechanical linkage transmits this movement to a governing valve that changes steam input in proportion to speed change. This basic flyball governor has the one unparalleled virtue of simplicity. But it is also a weakling, a laggard, and, on occasion, a liar.

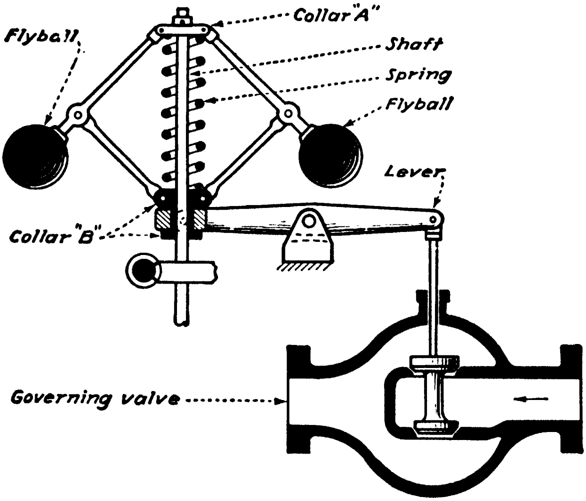


FIG. 14–1.—Flyballs linked solidly to steam valve must take a position corresponding to correct valve opening for every load from zero to full load. (*Courtesy of Power.*)

It is a weakling because the only energy it can contribute to move a valve is produced by the difference between the centrifugal force of the weights and the opposing spring force. Disregarding friction, the governor collar has a definite position for each speed; and, in each of these positions, the centrifugal force of the weights and the spring force balance each other.

It is a laggard because it must lag behind its true position enough to build up sufficient force to overcome spring opposition and friction resistance. As a result of lag, it becomes a liar because it is seldom in the true travel position as determined by speed, and, besides, its travel in line with a set speed change may vary at different points of the governor position.

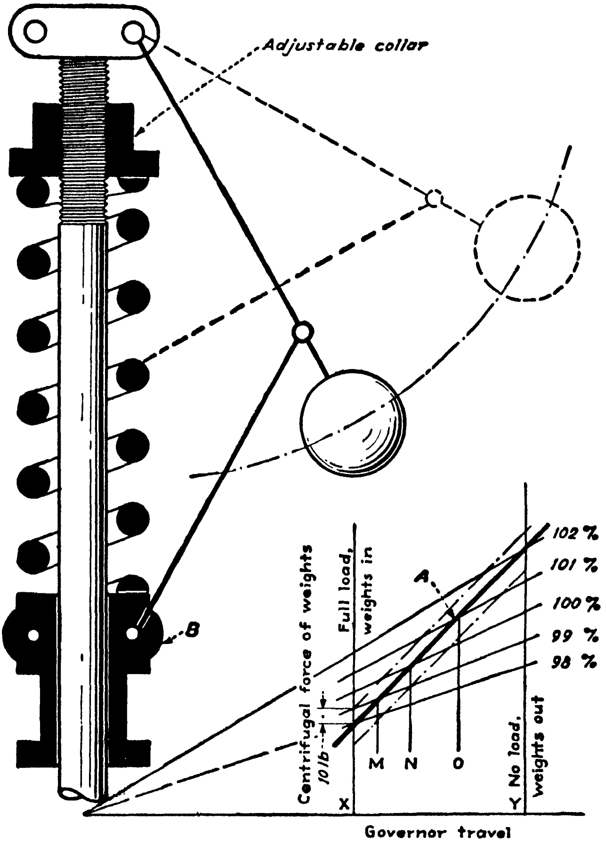


FIG. 14-2.—Rotate governor at constant speed, and change compression of a calibrated spring to obtain weight movement for each point along speed curve. (*Courtesy of Power.*)

The spring in Fig. 14-2 is calibrated, and we know its force for every length of compression. Let us rotate the governor at constant speed, 2 per cent below normal, and change the spring compression to obtain a weight movement for each compression point. We can then lay out a curve showing the corresponding spring forces for each governor position at 98 per cent speed. In like manner, we obtain the 99, 100, 101, and 102 per cent speed–centrifugal-force curves (Fig. 14–2). The curve shape depends on the governor design and may deviate considerably from a straight line because of link and radius distortion. For our purpose, however, we assume that they are straight lines converging toward the center of rotation, or zero centrifugal force.

The two vertical lines X and Y enclose that part of the governor travel needed to give full travel to the governor valve. The right-hand line Y, or the weights-out position of the governor, indicates the governor valve in no-load position. The left-hand line X, or the weights-in position of the governor, indicates the governor valve at its rated-load position. We now need a spring with a compression that is equal to the force shown at the intersection of line X with the 98 per cent speed line and that also matches the force shown at the intersection of line Y with the 102 per cent speed line at the corresponding travel or spring compression.

With this spring in place, the governor will open the governor valves far enough to carry rated load at 98 per cent speed and will close them enough to hold 102 per cent speed without load. The unit speed change from no load to rated load is 4 per cent and is called the *governor speed regulation.* It is obvious from the curve that, if a higher or lower governor speed regulation is required, the governor spring must be changed to match the different centrifugal forces of the weights at the different speed lines. If, for instance, a 3 per cent speed regulation is required, we shall have the same centrifugal force at 98 per cent speed with valves open, but the centrifugal force at 101 per cent speed and the valves closed will be less than we had at 102 per cent speed. The governor travel will be the same, because steam flow and valve travel required do not change. The governor spring force must then be made less for the same amount of compression. On the other hand, if a higher regulation is required, the governor spring force must be increased for the same compression. The amount of change depends on the centrifugal-force change from the original to the new value.

The curves in Fig. 14–2 show the governor travel for each 1 per cent speed change, at the point where the spring-force line *A* crosses the speed lines, provided that the governor sleeve is absolutely frictionless. Suppose the governor weights and linkage are so proportioned that the centrifugal-force change, for 98 to 99 per cent speed change, is 10 lb at governor collar *B* when it is in rated-load, weights-in position, or line *X*.

Suppose we restrain the collar *B* by adding friction so that a 10-lb force, over and above spring opposition, is required to move it from any position. The governor is now running at 98 per cent speed with full load; and, if we remove load from the unit, its speed starts to increase. However, collar *B* will not move until the speed increases to 99 per cent and the governor force increases above that of the opposing spring by 10 lb. The change in governor force for 1 per cent speed change without motion is the governor's power at that particular position. This particular governor has a power of 10 lb in rated-load position.

Suppose we further unload the unit and check the governor position in relation to its speed. We find it gives us a line parallel with *A*, but the 10-lb power of the governor will be obtained with less and less speed change as the weights travel outward. It is obvious that, when we talk about the power of a governor, we have to mention weight position. In the no-load position, line *Y*, about ⅔ of 1 per cent change produces the 10 lb necessary to overcome governor-collar friction, or we now have about 15 lb governor power for 1 per cent speed change.

The governor is now in the no-load position and at the 102⅔ per cent speed point because of the 10-lb friction on collar *B*. Suppose we slowly load the unit again. First the speed drops to 102 per cent, but this change will not start the governor collar moving because the forces in spring and weights now balance, and the speed must therefore drop another ⅔ of 1 per cent.

From here on, the collar will move as the speed decreases, drawing another line parallel to, but now 10 lb below, line *A*. This continues until the governor reaches rated-load position at a speed 1 per cent lower than the original 98 per cent. The vertical distance between parallel lines above and below line *A* represents the governor's dead band—the speed change in which no governor motion occurs. Since the same amount of friction gives one value of speed at one governor position and a different one at another, the dead band must also be considered in relation to the corresponding governor position.

Suppose we measure the governor travel for each 1 per cent speed change or the distances between points where the speed lines cross line *A*. We find that the governor travel between 101 and 102 per cent speed is almost three times that between the 98 and 99 per cent points, vertical lines *M*, *N*, and *O*. By changing the design of a governor, this characteristic may be reduced but seldom eliminated.

So far we have considered only characteristics of governors that have no means of changing the speed while in operation. When the governor must regulate a turbine that drives an a-c generator, the normal speed of the unit has to be maintained regardless of the load carried. We therefore need a speed changer that can alter the turbine speed while the unit operates. This can be accomplished in various ways, each one affecting the governor performance in a different manner. One of the methods used changes the governor-spring compression. Suppose we check how this affects the governor.

Let us look at Fig. 14–3*A*. Since the speed changer is not an automatic device, a load change will cause a speed change first, which is corrected later. Suppose we start a unit fitted with a governor set to have 102 per cent speed at no load. We bring its speed down to normal by moving the speed changer to decrease the governor-spring compression.

Line *a* then moves down, parallel to its original position, until the point, which originally crossed the no-load 102 per cent speed point, now crosses the 100 per cent point, dotted line a_2. If we now add load, the governor valve position and corresponding decreasing speed follows line a_2. To hold normal speed, we reset the speed changer for every load change up to rated load. At full load, the speed changer has now moved the governor spring to a position indicated by line a_1.

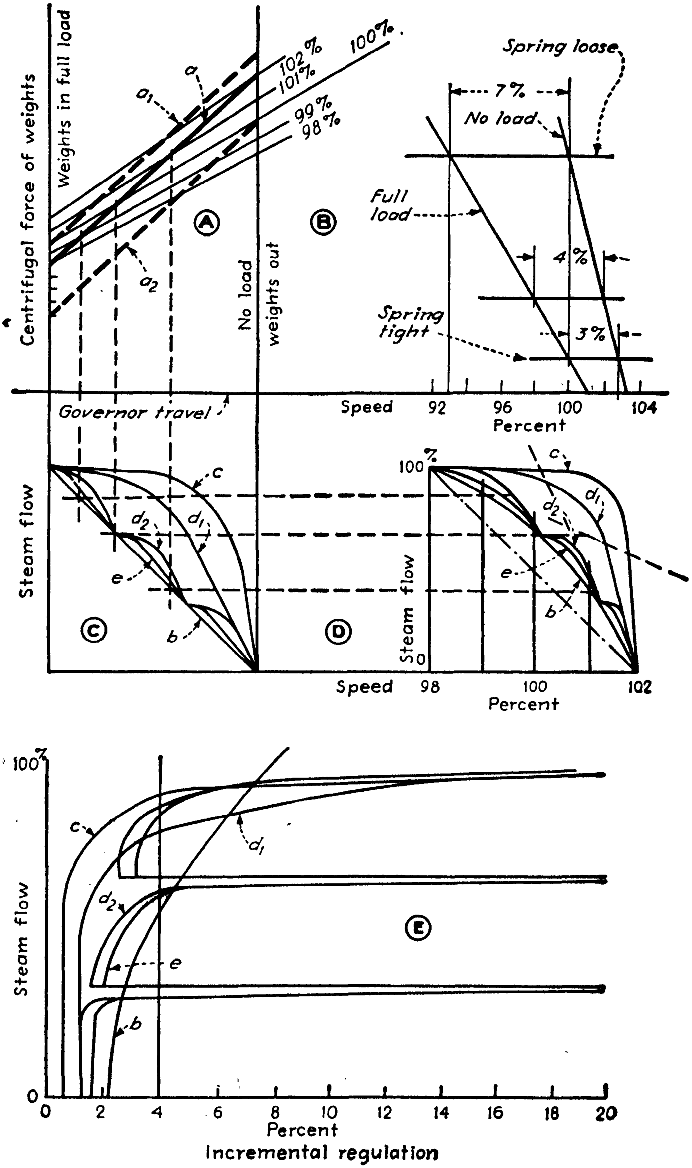


FIG. 14-3.—A speed changer is used to adjust turbine speed while unit is running. Changing the spring compression moves spring-force line *a* to new position. (*Courtesy of Power.*)

Well, there is nothing wrong here; we changed the governor-spring compression only to maintain normal speed through a complete load change. Let us look at that curve a little more closely. We are now in spring-compression position a_1. Suppose we unload the unit without adjusting the speed changer. The governor position and speed follow line a_1; but, when a_1 crosses the no-load line, the turbine speed has increased only 3 per cent. What happened to our 4 per cent governor? Now go back to our first position when we started to add load. With the governor-spring compression in position a_2, suppose we load the unit without adjusting the speed changer. The governor position and speed then follow line a_2, but it crosses the rated-load line 7 per cent below normal speed. We now have a governor that has 4 per cent, then 3 per cent, then again 7 per cent regulation.

We cannot move a line a crossing converging lines without showing differences between the wide and narrow parts between the speed lines. Figure 14–3*B* is a governor speed-regulation curve that shows the change in governor regulation with different speed-changer positions. It is easily obtained from graph *A*. Plot the percentage of governor regulation on the horizontal line and the change in speed-changer position on the vertical line. Then erect a vertical line at the 100 per cent speed point, and draw horizontal lines at the speed-changer position that gives 100 per cent speed with no load, then with rated load. Call the upper line *spring-loose* and the lower one *spring-tight*. Set off the speed change, where a_2 crosses the rated-load line (7 per cent), on the spring-loose line. Connect this point to the intersection of the 100 per cent speed line and the spring-tight line. Set off the speed change where a_1 crosses the no-load line (3 per cent) on the spring-tight line. Connect this point on the spring-tight line to the intersection of the 100 per cent speed line and the spring-loose line. The governor speed regulation for every speed-changer position can then be determined.

Figure 14–3*C* shows the relation between governor travel and steam flow, straight line b showing the ideal condition when steam flow changes proportionately for every governor movement. We can obtain this by connecting the governor to its valve (Fig. 14–1) by links so proportioned that, when the valve is near its seat, the governor weights are in their no-load weights-out position; and, when the governor reaches its rated-load weights-in position, it lifts the governing valve just the right amount to carry the rated load.

Assuming for a minute that the governor valve changes the steam flow in equal increments with travel, line *b* shows the amount of steam-flow change with the corresponding governor travel. This line loses its straightness if we plot steam flow against speed (Fig. 14–3*D*). The dot-and-dash line connecting the 100 per cent steam-flow point with the 102 per cent speed point shows the correct speed-flow characteristics we are striving to obtain. But, before we consider anything else, let us change the governor-travel–steam-flow line to speed and steam flow. Drop a line (Fig. 14–3*A*) from where spring-force line *a* crosses the 101 per cent speed line, until it intersects line *b* of *C*, and bring that intersection over horizontally until it crosses the 101 per cent speed line of *D* to get the correct steam-flow speed point for 101 per cent speed. Repeating the same process with the intersection of *a* with the 100 per cent and 99 per cent speed lines, we obtain the additional points on Fig. 14–3*D*, enabling us to lay out curve *b* on the speed–steam flow chart.

The curvature of this line comes from unequal governor travel for an equal speed change. Even with a perfect governor valve, the speed–steam-flow line is far from ideal. The real effect of this curve is not fully appreciated until we replot it in Fig. 14–3*E*, which is an incremental regulation steam-flow curve.

The incremental regulation of a governor is what the regulation happens to be at any one load point. Suppose we try to convert the governor's speed–steam-flow curve (*b* in Fig. 14–3*D*) into an incremental regulation curve. Start at zero steam-flow point, and draw a tangent to the curve until it crosses the 100 per cent steam-flow point. We find it crosses at a little over the 100 per cent speed point; or slightly more than 2 per cent change in speed is required from zero to 100 per cent steam flow if the initial rate of travel is maintained over the full range of steam flow.

Repeating the same performance at the 100 per cent flow point and extending the tangent to the zero flow line, it crosses somewhat beyond the 106 per cent speed point for a total speed change of 8 per cent. Obtaining a few more points in between allows us to plot curve *b* in Fig. 14–3*E*, which does not look like the dot-and-dash line we have been working so hard to obtain. We start with a little more than 2 per cent regulation and end up with 8 per cent; and only at one point, around the one-half steam flow, do we actually have 4 per cent regulation. To be exact, we should correct this curve in line with the speed-changer curve of Fig. 14–3*B*.

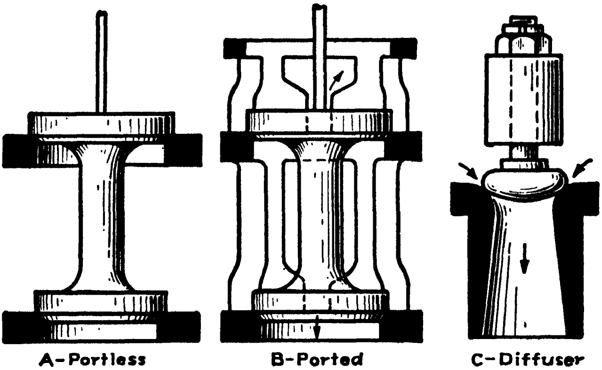


FIG. 14–4.—Valve shape can help or hinder governor action. Restricted ports at small openings help even out velocity flow over full valve travel.

We now have to add the peculiarities of governing valves to obtain a final complete picture. Figure 14–4*A* shows a governing valve having full admission without restriction, and the admission area changes in direct proportion to lift. Ported valve *B* shows the steam-admission area restricted by ports located in the valve cage. The third valve *C*, with a diffuser seat, has characteristics comparable to a cam-operated plug valve.

Returning to the portless valve *A*, let us check its travel-flow characteristics. At the valve-open point, with minimum pressure difference, steam flows at only one-tenth the velocity it would have when the valve just begins to open. The resultant steam-flow curve *c* (Fig. 14–3*C*) is further distorted by the uneven governor travel in Fig. 14–3*D*. More than 90 per cent of the steam flow is admitted with only 1 per cent speed change, and the first part of curve is so steep that the incremental regulation for more than one-half steam flow is less than 1 per cent (curve *c*, Fig. 14–3E).

A much better steam-flow travel line is obtained if the valve admission area is restricted (Fig. 14–4*B*) so that high-velocity steam has considerably less area per valve travel than lower velocity steam at the valve-open point. Resulting curve d_1 (Fig. 14–3*C*, *D*, and *E*) shows considerable improvement over curve *c*. A still further improvement can be obtained by using several steam-admission points on a unit, each controlled by a separate governing valve (curve d_2 on Fig. 14–3*C*, *D*, and *E*). The incremental regulation line shows that, except for the high-regulation points at wide-open valve points, the rest of curve d_2 follows the incremental regulation curve *b* of the governor closer than curves *c* and d_1. The maximum regulation obtained at wide-open valve points could be considerably reduced if the final pressure drop across a valve were increased before the next valve started to open. Of course, an increased pressure drop also affects the efficiency of the unit at valve-open position. So far the governing valves have not helped the governor out any; as a matter of fact, the incremental regulation line of the valves, so far considered, shows considerable variation.

Plug valves with diffuser seats, in sufficient numbers, show the smallest deviation from a straight flow-travel line (curve *e*, Fig. 14–3*C*, *D*, and *E*). The effect of uneven governor travel cannot be eliminated; and, as far as the valves are concerned, minimum deviation from the governor line should give acceptable valve performance.

Another governing-valve characteristic that must be considered, especially when it is operated by the governor directly, is the force required to move the valve through its operating range. Examination of Fig. 14–4*C* shows that this valve is not suitable for direct connection to the governor because full steam pressure tends to hold the valve on its seat. Considerable speed change must occur before the governor can build up power to lift the valve against this pressure; then, as soon as the valve lifts, steam pressure builds up underneath, adding its power to that of the governor. The valve opens wide and then shuts, causing considerable change in turbine speed. So-called "balanced valves" (Fig. 14–4*A* and *B*) are better suited for direct connection to the governor, but even here the forces are not negligible (Fig. 14–5).

The zero line through the middle of the curve represents balanced forces acting on the valve, and the governor has no difficulty in operating a valve with this characteristic. Actually, few valves even approach this line, because the lower-disk area of the valve is slightly smaller than the upper one so that it can pass through the upper seat when the valve is assembled. With steam between the upper and lower seats, unbalanced forces try to lift the valve from its seat (Fig. 14–5), starting above the zero line at *X*.

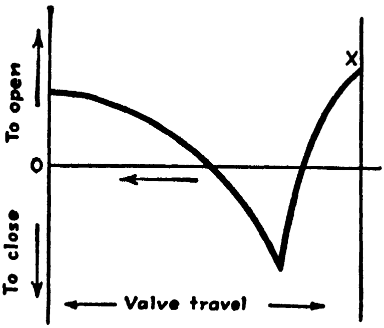


FIG. 14–5.—Double-port valve is balanced at only two points of its travel.

After the valve lifts, the pressure drop and dynamic forces change the lifting to a closing force, with its highest value at approximately one-quarter total lift. From here on, increasing pressure on the discharge side gradually changes the force curve to the opening direction, reaching full value at wide-open valve when the discharge pressure nearly equals the inlet pressure. Changing the valve shape affects the force curve. It is advisable to use this control combination on drives where the no-load steam flow requires a valve opening greater than one-quarter travel; otherwise, the governor-force–travel curve becomes so distorted that good operation is seldom obtained. Valve size and steam pressure generally determine whether the control scheme (Fig. 14–1) can be used successfully.

Knowing a governor's characteristics allows us to eliminate some, but not all, of its objectionable features. Most mechanisms employ a speed changer, whose design determines some of these characteristics. Figure 14–6*A* illustrates a speed changer that acts through a secondary spring *X*. The main feature is that the weights have to be moved to change the governing valves to a different flow position while maintaining the same turbine speed. That is why it is called a *travel governor.*

Thus, the speed–governor-travel curve is distorted, which affects the incremental regulation curve regardless of the kind of governor valve used. Also, the position of the speed changer affects the speed regulation. Since most distortion comes from moving the governor weights, why move them? If we do not, the governor is a nontravel unit (Fig. 14–6*B*). Its speed changer does not change the governor-weight position as long as the unit runs at normal speed. Using the governor collar as a fulcrum, the speed changer moves the valve-position-lever center to obtain different valve positions with one identical governor-weight position.

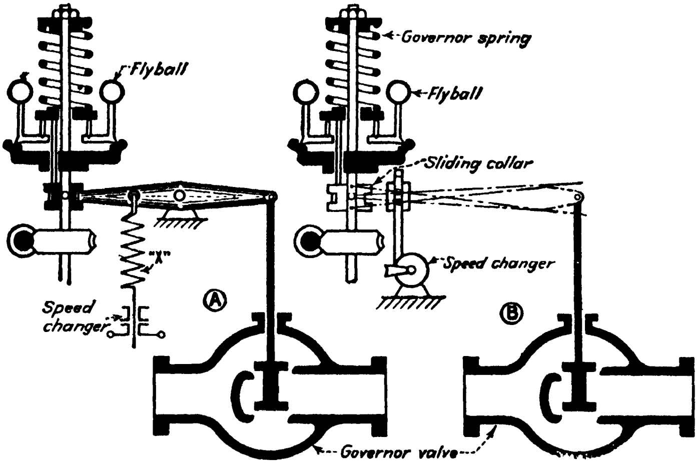


FIG. 14–6.—(*A*) Speed changer acting through secondary spring moves governor weights to change valve position. (*B*) Collar serves as fulcrum; weights do not move. (*Courtesy of Power.*)

How this scheme affects the governor's characteristics is shown by Fig. 14–7. This system has the advantage of the same speed-travel relationship at all loads, since the position of the governor weights does not change. A considerable amount of governor travel is needed to fulfill the requirements that it open the governing valves fully from no-load position, because of speed change, without changing the speed-changer setting, and also drop full load and close the valves from the maximum-load rated-speed position. The speed change above and below rated speed also requires travel, and this governor must therefore have an accurate travel setting to obtain the specified speed range. Thus, with a little change on the speed changer, we can improve the governor performance considerably.

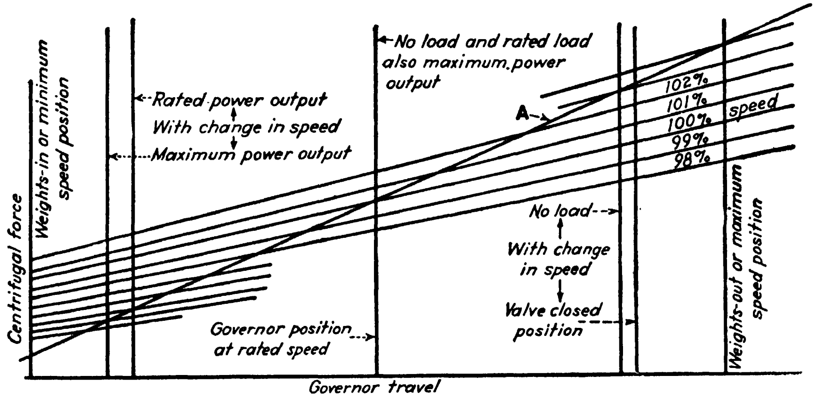


FIG. 14-7.—Characteristics when employing a speed changer (Fig. 14-6*B*): Since governor flyball position does not change, device has same speed-travel relation at all loads. (*Courtesy of Power.*)

Since the governor can rarely be mounted conveniently near the valve, designers began to use more mechanical linkages to transmit motion over greater distances; and every addition increased the over-all friction. To banish this, they used ball bearings at all joints and devised mechanical oscillators to keep the governor system moving ever so slightly, thus substituting kinetic for static friction. But, whether static or kinetic, friction still remains a bugaboo in these flyball governors.

While these changes were going on, turbines were growing up and using bigger valves that required larger governor forces. Yet there was no way to increase the power output of the flyball governor directly, except to increase its size. Considering that a modern turbine of medium size requires a valve-operating force of perhaps

5,000 lb, it is not difficult to see that increases in the size of the governor soon begin to reach a practical limit.

Turbine engineers got around this difficulty in the flyball governor by using its relatively small mechanical output to drive an amplifier, which in turn controls an external energy source for force magnification (Fig. 14–8*C*). The power-magnifier system consists of a relay controlling the admission of high-pressure oil to move the operating piston of a servomotor in line with the governor motion. An extended mechanical linkage between the governor and the servomotor relay required more lag in the governor position than good operation could permit. One stage of amplification was not sufficient, but it is relatively simple to add stages as needed.

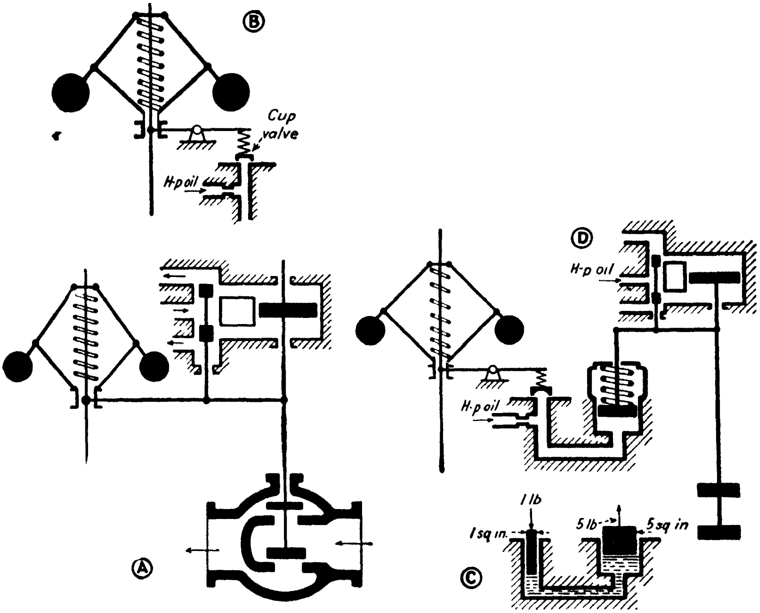


FIG. 14–8.—(*A*) Flyball's small power output moves amplifier piston controlling an output power supply. (*B*) Introduction of additional amplifier. (*D*) Servomotor furnishes power to move heavy valves. (*Courtesy of Power.*)

Since mechanical linkage between the governor and the admission valve requires supports that always increase the friction that the governor has to overcome, the amplifying system adopted not only increases the available force but also transmits the force from the governor to the servomotor without additional mechanical connections. The system then consists of a governor and a transformer, or device, that translates movement into a change of hydraulic pressure (Fig. 14–8*B*).

Pressure change then acts on the amplifier, which, using one of the oldest hydraulic principles, magnifies the pressure change by a given ratio. Consider two pistons in a closed hydraulic system (Fig. 14–8*C*). One piston has a cross-sectional area of 1 sq in.; the other has 5 sq in. If a push of 1 lb is exerted on the 1 sq in. piston, the liquid pushes against the larger piston with a force of 5 lb, or the force is amplified five times. The actual amplifier does not work quite this way, but the principle is the same.

The servomotor is nothing more than a piston connected to the turbine governing valve, free to move in either direction (Fig. 14–8*D*). A decrease in pressure output of the governor-operated amplifier moves the servomotor one way, and an increase moves it the other.

Now what has been accomplished? A little power has been transformed into a lot, and mechanical-linkage oscillators are no longer necessary. Also, since only a little power is needed to actuate the hydraulic system, the governor itself can be made small, light, and relatively free from friction.

Some friction still remains in the system, however. Engineers went back to the speed-sensitive element itself and substituted a metal strap carrying centrifugal weights for the flyballs and sliding collar (Fig. 14–9*A*). In principle, the two devices are identical except that the strap governor produces forces entirely by an elastic change and hence is frictionless. The strap governor also produces forces without motion (maximum motion 0.005) and so eliminates errors from different valve travels at different governor positions for the same speed change. The transformer (Fig. 14–8*B* and *D*), in combination with its governor, is a travel unit with all the travel-flow distortion and changes in governor-speed regulation inherent in this type. On the other hand, Fig. 14–9*A* is a nontravel mechanism giving improved governor performance.

First the governor was entirely mechanical, then hybrid mechanical-hydraulic. Why not completely hydraulic? A pump driven by the turbine shaft becomes the sensitive element, and its output pressure is proportional to the square of the shaft speed. The change in pressure then operates a relay and the rest of the amplification system as before (Fig 14–9*B*).

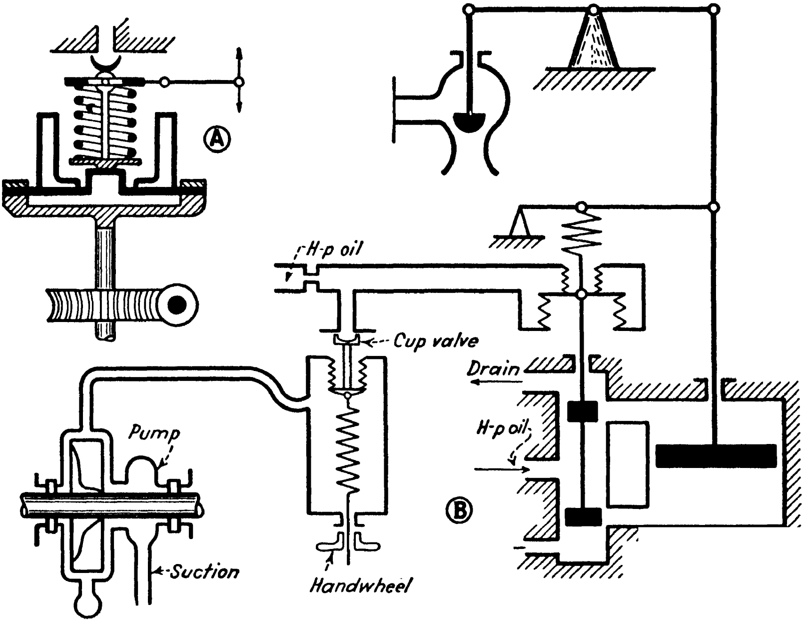


Fig. 14–9.—(*A*) Metal strap and weights provide a frictionless governor head. (*B*) Pump governor operates cup pilot valve to control flow of high-pressure governor oil. (*Courtesy of Power.*)

Except for the fluid viscosity, the entire governing system has now become almost frictionless. Complex mechanical linkages and gear drives have been banished, and there is almost nothing to wear out or require periodic inspection. Since the speed response depends not upon hydraulic flow but upon a change of hydraulic pressure, the governor can be made to respond to extremely small speed changes.

Any pump driven by the turbine shaft may be employed as a sensitive speed-responsive element, provided that its discharge is not affected by any impulse except speed. All pump discharges are influenced by suction-pressure changes and other possible errors, however. A governor pump otherwise known as an *impeller* was developed (Fig. 14–10). Suction effect is overcome by supplying the pump with high-pressure oil, and the effect of other disturbing factors is reduced so that their influence is negligible.

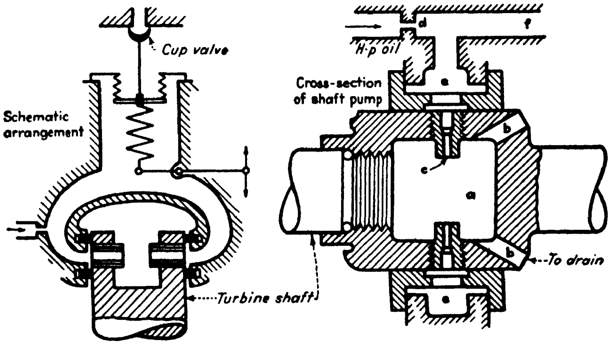


FIG. 14–10.—Pump governor constructed as an integral part of turbine shaft eliminates friction usually present in moving gears and levers of mechanical drive.

The impeller consists of a hollow chamber in the shaft, *a*, connected to a drain chamber by properly dimensioned discharge openings *b*. A series of pumping nozzles *c*, placed radially in the shaft, receive oil through an orifice *d*, from a higher pressure source than their discharge pressure at maximum speed. An annular seal *e* surrounds the shaft chamber so that each nozzle connects constantly to the high-pressure source.

With the turbine at rest, high-pressure oil from the orifice flows through the nozzles into the hollow-shaft chamber and out the discharge openings, and no pressure is applied to the power-amplifying system *f*.

As the turbine shaft rotates, centrifugal action begins to retard oil flow through the nozzles, and pressure builds up in passage *f*, leading to the power amplifier. Pressure generated in the nozzles by centrifugal force and that applied to the amplifier system depend on turbine speed. Since this pressure is always less at normal speed than the orifice pressure *d*, a small amount of oil always flows from the orifice through the nozzles to chamber *a* and out the discharge openings *b*.

Strap and fluid governors are essentially nontravel governors. The hydraulic pressure and power amplifiers used beyond the cup valve are identical for both. Quite often the pressure and power amplifiers are built in separate mechanisms and can be used with either governor. It may seem that these two types of governors should be universally applied. Both have everything—low maintenance, high sensitivity, and versatility. But, just as no one would use a micrometer to measure a garden plot, so it would be equally poor engineering judgment to use a fluid or strap governor on a machine that does not require accurate speed regulation.

Consequently, current design embraces four kinds of governors, varying in cost and performance: (1) a flyball governor with or without hydraulic-power amplifications; (2) a gear-orifice governor with or without power amplification (in sensitivity and power, this unit matches the performance of the first, but in addition it can be used for a wide speed range of 3:1); (3) an elastic-strap governor with hydraulic pressure and power amplification; and (4) a fluid governor with pressure and power amplification.

Applications overlap somewhat, because, with hydraulic power and pressure amplification, an elastic-strap governor can do any speed-regulating job as well as the fluid governor. For precise regulation, either of these systems can be used, because their sensitivity, rate of response, and maintenance characteristics are almost identical.

The fluid governor with a large-capacity oil pump on the turbine shaft furnishes the large instantaneous oil demand required for fast servomotor operation. It also permits locating the oil reservoir quite a distance from the unit in a fireproof room, and it does not need a vertical shaft and gear drive. With its more flexible arrangements, this governor is exceptionally well suited to large-turbine application. On the other hand, the strap governor serves ideally for small units of low oil requirements with attached oil reservoirs.

The two latter governors have built-in performance characteristics that meet the most exacting demands for sensitivity and rate of response, since they respond to minute speed variations almost instantaneously. Other unit governors in the same station must be brought up to reasonably close performance. The more sensitive governor picks up load or drops it quickly, whereas a parallel turbine controlled by an insensitive governor may not be influenced at all by the same frequency swing.

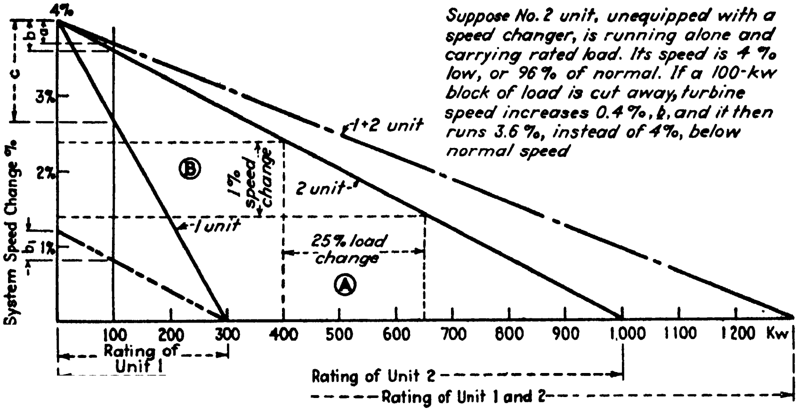

Suppose No. 2 unit, unequipped with a speed changer, is running alone and carrying rated load. Its speed is 4 % low, or 96 % of normal. If a 100-kw block of load is cut away, turbine speed increases 0.4 %, b, and it then runs 3.6 %, instead of 4 %, below normal speed

FIG. 14–11.—A governor with 4 per cent regulation allows turbine speed to fall 4 per cent from no load to rated load. For example, if load on one machine changes 25 per cent of rated load in either direction, *A*, turbine speed will change 25 per cent of 4 per cent or 1 per cent, *B*. (*Courtesy of Power.*)

Another factor to consider when analyzing governor performance is the relative generating capacities of units operating in parallel. Figure 14–11 shows what happens to the frequency when a 1,000-kw and a 300-kw unit operate in parallel or alone. Assuming that both governors are identical and set for 4 per cent speed regulation, the dot-and-dash line shows that, when operating in parallel, a 100-kw load variation changes the system speed 0.3 per cent, *a*. If each unit runs alone, a 100-kw load variation on the large machine changes the system speed 0.4 per cent, *b*, while the smaller unit allows the speed to vary 1.34 per cent, *c*, for the same load change.

If the speed change must be identical on both machines for the same load change, the regulation of the smaller unit must be reduced to about 1.2 per cent, dot-dot-dash line, where a 100-kw load variation changes the speed 0.4 per cent, b_1. This low regulation may not be stable; and, if both units operate in parallel, the smaller unit will completely load or unload with a 600-kw load change on the system. With the governors set at 4 per cent speed regulation, both units pick up load in proportion to their capacity.

Choosing the right governor cannot be an arbitrary decision; the turbine size, speed regulation, and type of load must all be weighed and judged in their relation to each other, to obtain the best governor performance characteristics. Each governor must be selected to fit the job.

CHAPTER XV

HAND VALVES AND PIPING

At the core of every power engineer's job lie piping systems—vast arterial networks that convey steam, water, air, refrigerant, brine, and other power fluids. The first step in laying out a piping system is to specify the proper type of hand-operated valves. If a valve is properly chosen, correctly installed, and conscientiously inspected, the need of spare parts and the cost of repairs will be negligible. Incorrect choice, damage during installation, or lack of frequent inspection, however, will result in excessive repair costs or actual need of replacement. For example, high-pressure valves, having stronger bodies and being more able to withstand stresses, may be the best choice for lines subjected to excessive vibration, even on low-pressure duty. Or, under some conditions, the more easily reseated globe valve may be better suited for cutoff purposes than a gate valve. Every condition that can affect valve life must be studied before a choice is made.

When selecting new valves, or rearranging those in service, choice is determined by the quantity of fluid to be passed, the pressure, the temperature—which is sometimes more important than pressure—the nature of the service, the kind of fluid to be handled, and the physical construction at the point of installation. When valves already installed give an unusual amount of trouble, investigate the cause before placing all the blame on them. As a start in this investigation, consider what effect the size of a valve has on its life.

Common practice is to chose a valve of the same size as the pipe to which it will be connected, disregarding any comparison between valve size and the quantity it must pass under actual working conditions. It is not unusual to find a large valve operating with the disk just barely raised off the seat and yet passing the full required flow. In such a case, service life is rapidly shortened. Better results are obtained by selecting a size that passes maximum flow demand when wide open and using pipe reducers to connect the valve into the line. Then the disk will be out of flow as completely as possible, reducing seat and disk erosion to a minimum. This is also true of automatic regulating valves. Select them of such size that they will be loaded near capacity; because, when they are too large for the service, the constant wire-drawing action of the restricted opening scores the seats seriously. Damage can also occur when a safety pop valve, of the correct size, connects to the pressure vessel through a valve-size pipe extension. The pressure drop within the pipe causes the valve to chatter instead of operating positively. Connect a safety valve directly to the vessel, or use oversize pipe for the extension.

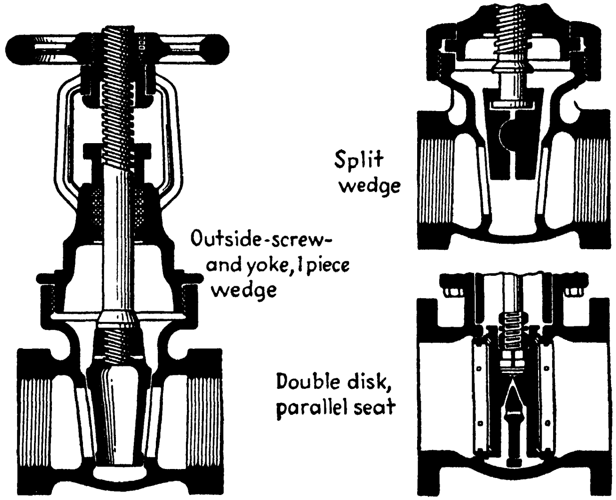


FIG. 15–1.—Gate valves with screwed, union, and flange bonnets and rising stems.

When handling extreme temperatures and pressures or corrosive fluids, select metal that will withstand these conditions. Many normally inoffensive materials prove highly corrosive at elevated temperatures. The body and seat metal of an internal-spring, pop, or safety valve may be capable of withstanding fairly high temperatures; but the spring, if not made from suitable metal, may be damaged at temperatures above 300 F.

Bronze valve bodies in sizes up to 3 in. serve for steam pressures as high as 350 psi at 550 F. Cast iron is suitable for steam of 250 psi at 450 F. Higher pressures and temperatures require steel, monel, or other alloy bodies. Stellite trim is good for 1000 F with steam and 1100 F for oil.

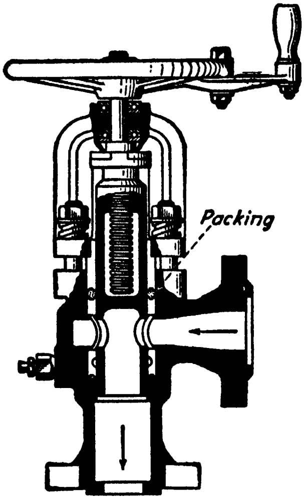


Fig. 15–2.—Seatless valve with nonrising stem and inside thread. (*Courtesy of Yarnall-Waring Co.*)

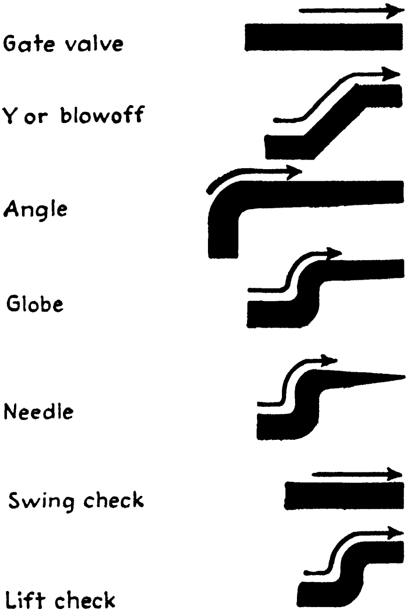


Fig. 15–3.—The closer a valve regulates flow, the more resistance it introduces. (*Courtesy of Jenkins Bros.*)

The fluid to be handled affects valve selection in two ways: corrosive or erosive properties determine the materials, and physical properties influence the choice of valve construction. No arbitrary rule can be applied to all fluids. For steam and condensate, cost and strength considerations are usually more important than corrosion resistance. Raw water, salt water, and most common industrial solutions are electrolytes and may cause corrosion where dissimilar metals are present. The cure for persistent corrosion troubles often lies in switching to an all-iron, all-bronze, or all-steel system. Highly corrosive fluids require special materials, and, for safety, the manufacturer should be consulted.

In general, use one-piece-wedge gate valves (Fig. 15–1) for gummy liquids or when the valve must be installed with the stem pointing downward. Double-disk or split-wedge gates are good for liquids containing suspended matter, because their flexibility permits one disk to seat even though grit prevents full contact on the other side. This flexibility can cause disk binding if the valve is installed in an inverted position. The sleeve, or seatless, angle valve (Fig. 15–2) has no exposed seating surfaces, making it suitable for abrasive fluids.

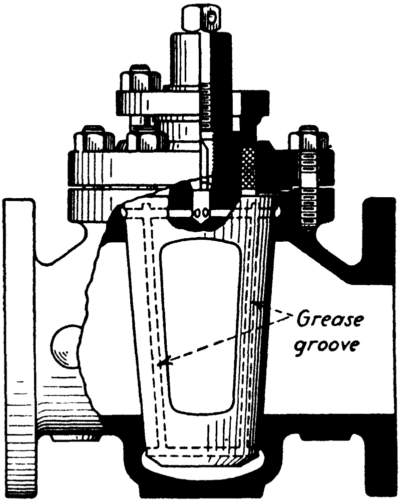


FIG. 15–4.—Lubricated plug valve. Flow is straight through.

Another point to consider is whether the service requires free flow or flow control. A flow-control valve offers some restrictions to flow even when wide open, and the one giving greatest control introduces the most resistance (Fig. 15–3). The gate valve and plug cock in Fig. 15–4, having the least resistance to flow, fit cutoff and free-flow applications, because when open they offer no flow restriction, and, since partial opening is not required, no seat damage results. Throughflow characteristics of gate valves make them ideal for boiler-water columns and blowdown service, drain valves for header flushing, and cutoff valves on pumps and compressors. The ordinary globe (Fig. 15–5), angle, or *Y* valves (Fig. 15–6), offering only slight resistance to flow, are suitable for cutoff and free flow if the service permits the slight pressure drop. On the other hand, plug-globe and needle valves (Fig. 15–7), being designed primarily for flow control, introduce some restriction even when fully open.

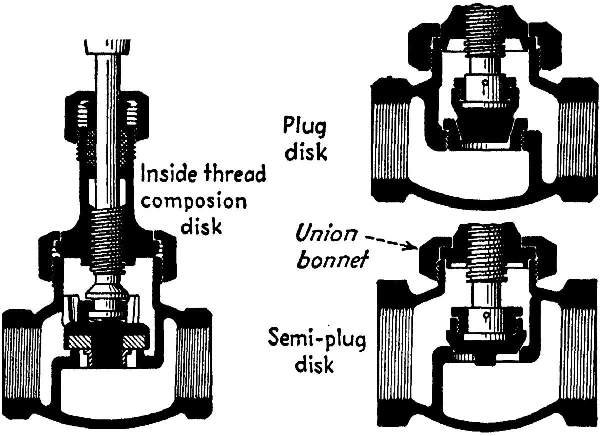


FIG. 15-5.—Union-bonnet globe valves with rising stems and different types of disks.

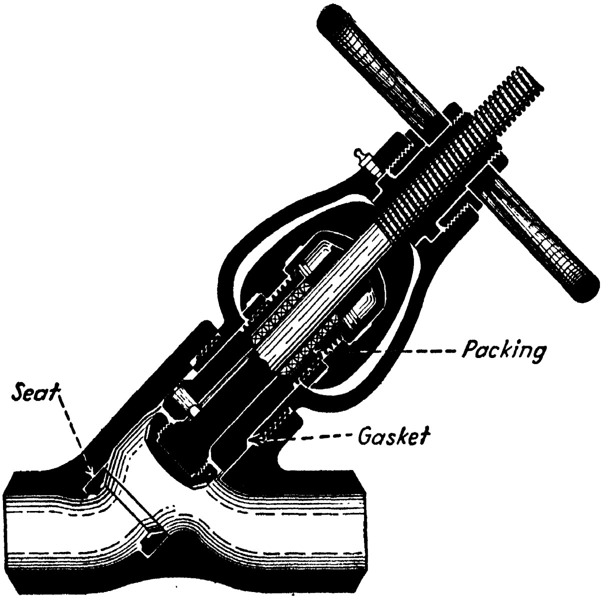


FIG. 15-6.—Pressure-seal bonnet-globe valve with rising stem. (*Courtesy of Crane Co.*)

Consider also the physical condition at the point of installation to ensure convenient operation and inspection. Headroom must be available for the rising stem. A rising-stem valve (Fig. 15–8) gives visual indication as to its open or closed position, making it more desirable from an operating standpoint. Also, if a bend occurs near a globe valve, select an angle valve in preference to a straight valve and elbow. Other factors of importance, and how they are affected by the conditions described previously, are bonnet construction, valve trim, and stem operation.

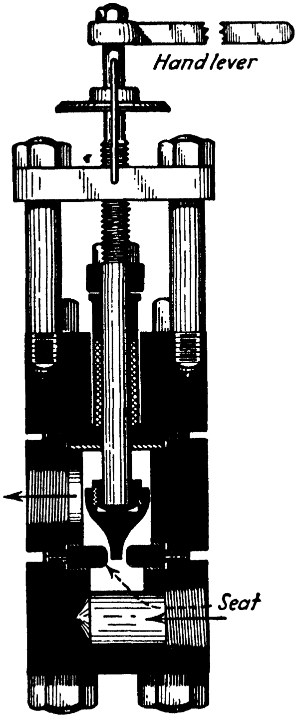


FIG. 15–7.—Needle valve used for continuous boiler blowdown service. Seat is reversible. (*Courtesy of Strong, Carlisle & Hammond.*)

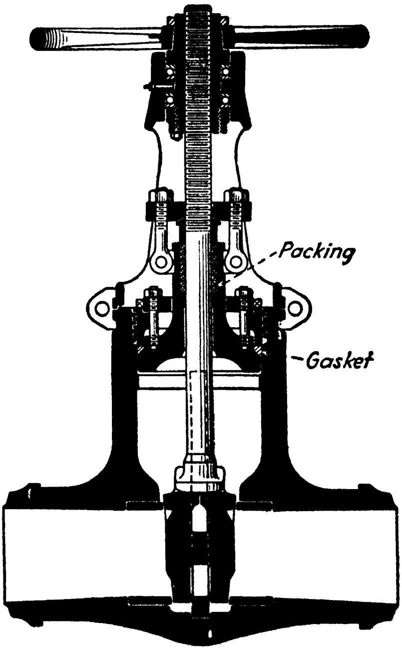


Fig. 15–8.—Pressure-seal bonnet gate valve with rising stem. (*Courtesy of Crane Co.*)

Valve bonnets are fastened to the bodies by threaded, union, or flanged joints (Fig. 15–1). Some operators object to the threaded joint, because, if for any reason the stem thread becomes jammed in the bonnet, as sometimes occurs, effort applied to the handwheel tends to turn the stem and bonnet as a unit. This may happen on a jammed-open valve, and the operator unknowingly, thinking it closed, may apply considerable force to open the valve. This tends to loosen the bonnet and remove it from the valve body. Another objection to the threaded bonnet is the impossibility of using the bonnet as a guide while grinding the seat.

The union-ring construction eliminates the danger of removing the bonnet when turning the handwheel. Furthermore, this construction stiffens the body against internal pressure, makes dismantling a simple operation, eliminates to a large extent any danger of distortion, and provides a part not directly subject to internal pressure to take the wrench action when the valve is being inspected. With this joint, the valve bonnet serves as a guide for the valve stem during seat grinding.

Disks and seat rings, called the *valve trim,* are constructed from metal similar to the body or from an entirely different material. They may be constructed either as an integral part of the body, or separately and screwed or rolled in place to a tight fit. The principal factors influencing the performance of seat metal are its tensile properties at the operating temperatures, hardness and toughness, a coefficient of expansion corresponding closely to that of the valve body (to prevent loosening the seat rings and leakage past their threads), and enough difference in the properties of seat and disk facings to prevent seizing of their surfaces when in sliding contact.

The globe valve disk made from harder metal than the seat resists the forming of a shoulder on the disk. It is also essential that seat and disk be immune from growth or other chemical change from temperature variations. To retain the smooth surface necessary for tightness, they must resist oxidation, erosion, and corrosion under service conditions. Brass or bronze, though entirely satisfactory for water or saturated steam, will not withstand superheated steam or oil temperatures above 550 F. Higher temperatures call for alloy parts.

Severe conditions of temperature and pressure require monel, chromium nickel, stainless iron, nitrided steel, or cobalt-chromium-tungsten alloys (stellite). The latter two are hard-facing operations, nitriding being a heat-treatment of the steel in the presence of ammonia gas, and the other being a welding process in which a layer of the alloy is applied to the parent metal, usually stainless iron. The cobalt-chromium-tungsten, an extremely hard alloy, has the ability to retain its hardness under red-heat conditions such as those found in high-temperature work. Its red hardness and low coefficient of friction eliminate galling when it is used on surfaces in sliding contact.

Galling and seizing occur when certain similar metals are in sliding contact. It cannot be prevented simply by providing a smooth finish of the surfaces. Dissimilar seat and disk metal, such as seats of the ferrous alloys and disks of a nickel-copper alloy, chromium stainless iron heat-treated to give different hardness values, or nitrided-steel seats and disks, prevents seizing. Conditions conducive to galling are encountered when the disk must slide in contact with the seat, a characteristic found more often in gate than in globe valves.

Globe valves have a metallic seat, sometimes integral with the body and in other cases a separate piece screwed into the body, the disk itself being either a composition material or metal. Choose the composition disk best suited for the service. Use rubber in various grades of hardness for cold water, hot water, and air; molded asbestos and rubber for steam service; or oil-resistant artificial rubber for petroleum and its fractions. The safe upper temperature limit on which composition disks should be used is 400 F. Valves containing these disks designed for 150 psi steam pressure can be used for other fluids at 300 psi if the temperature does not exceed 150 F.

The composition disk, with its small bearing surface, although likely to be cut by foreign matter, allows considerable embedding of dirt before leaking. This disk has a shorter life than a metallic plug; the disk can usually be reversed and both sides used; or, if it is not too badly damaged, it can be machined to a new face. This material gives good service under open and shut conditions when operation is not too frequent.

On installations that involve frequent operation, give consideration to semiplug or even to plug-disk globe valves (Fig. 15–5) designed for throttling service, because each closing and opening subjects the seat and disks, for a short period of time, to an erosive effect comparable with that which occurs in a throttling valve. Steam pressures above 150 psi and temperatures over 350 F and particularly severe services require valve trim made from the hardest materials obtainable, such as the hardened alloys. Use only plug-disk valves for throttling work, because the wide seating surface offers considerable resistance to wear and injury from foreign matter. Disks and seats made from the harder alloys increase valve cost, but their long service life makes them an economical investment.

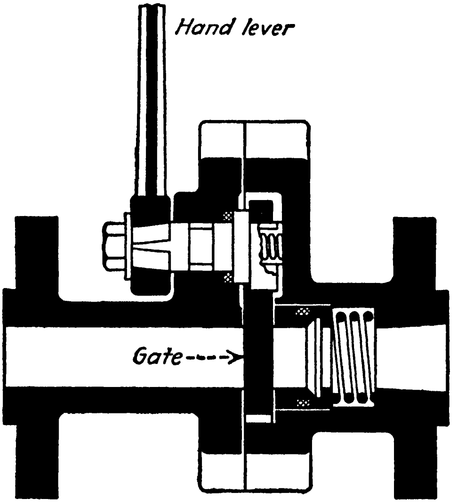


Fig. 15–9.—Swinging gate valve is suitable for liquids carrying suspended particles.

Disks for gate valves are shaped as solid wedges, split wedges, or double disks with parallel seats. Figure 15–9 shows a swing gate valve. The term *solid wedge* is slightly misleading; the wedge, although in one piece, is quite often not solid, being in some designs a hollow casting. The disk seating area can be formed from the wedge metal itself, or, as in some iron valves, a ring of different metal may be pressed into a groove machined in the wedge. Although classed as solid, the wedge in Fig. 15–8 is made flexible by its *H* construction.

The disk in the parallel-seat gate valve slides against the seat during the final closing and initial opening. Consequently, these parts must be made of metal that has no tendency to seize or drag. The solid wedge, being held away by guides, does not come into contact with its seat until the extreme closing point and therefore has no opportunity to seize. As a final point in choosing the valve, determine what kind of stem will serve best.

Stem choice depends on the physical surroundings at the point of location as well as on the fluid being handled. Internal threads (Fig. 15–10) will be affected by the fluid and are unsuited for abrasive or encrusting materials, high temperatures above 450 F, or hot coking oils; external threads (Fig. 15–6) are exposed to atmospheric conditions but are easy to lubricate, and such valves are therefore suitable for handling fluid damaging to internal-threaded stems. The rising stem with its additional height is exposed to the hazard of being bent by foreign objects. The nonrising stem requires no additional headroom, but the absence of any visual indication of position may result in the valve's being subjected to excessive operating strains in the wrong direction, a condition more likely to occur on valves with excessive friction caused by lack of lubrication or dry packing.

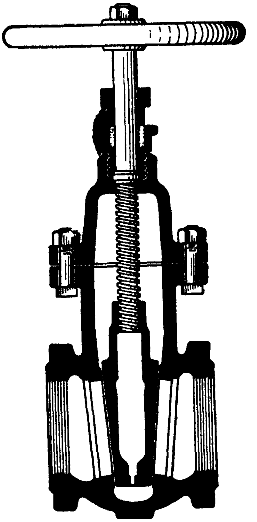
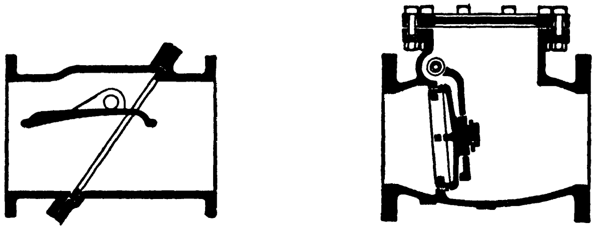


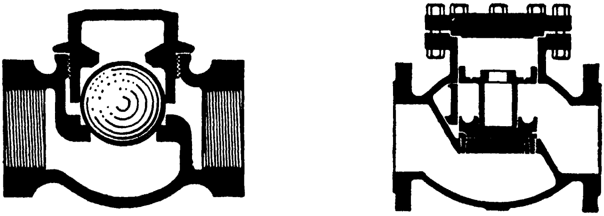
FIG. 15–10.—Gate valve using an inside-thread, nonrising stem and flange bonnet. (*Courtesy of The Wm. Powell Co.*)

Check valves (Fig. 15–11) work automatically and have no means of opening or closing other than the action of the fluid on the disk as it flows through the valve. The disks, either composition material or metal, operate by lifting or swinging away from the seats. Some lift check-valve disks act as a piston in the cylinder formed by the body neck above the seat. They are adapted for pulsating-flow service, because the pistonlike action of the disk, as it lifts, cushions the disk and prevents its slamming against the cover with each stroke of the compressor or pump. In others, a spring above the disk performs the same function and likewise helps return the disk to its seat before flow reverses to minimize water hammer.



Tilt check

Swing check



Ball check

Cushioned lift check

FIG. 15–11.—Different types of check valves.

The basic rule covering the installation of check valves is that they must be positioned so that gravity tends to close them. Swing checks can be used in horizontal lines and for upward flow in any line inclined at any angle up to 90 deg from horizontal. Regardless of the angle of the pipe line, always install swing check valves with the hinge pin horizontal. This ensures that the disk will always swing in a vertical plane.

It is best practice to use swing checks with gate valves and on liquids easy to hold and not highly volatile. Resistance to flow is less in these check valves; and, since liquid velocities are usually low, the destructive effect of slamming, which ruins swing check valves, is greatly reduced. On the other hand, they are not serviceable on rapidly pulsating flows and are difficult to keep leaktight.

Swing checks with integral disk-hinge construction can be used on hot or cold water and on low-pressure steam. The regrinding type can be used on hot or cold water and on other fluids not corrosive to the metal. Leather disk valves are recommended for cold-water service especially if foreign matter is present.

These valves are well adapted for use on the discharge side of centrifugal pumps where water must be delivered to overhead tanks or elevated reservoirs. When high heads are involved, however, there is great danger of tremendous shock pressures building up in the line when the pump shuts down. If this occurs, it indicates that reverse flow is gaining considerable momentum before the check valve closes. A cushioned check valve will not cure the trouble. The remedy here is to speed up the check valve and make it close slightly before the water column stops moving in the forward direction. This can be accomplished by equipping the check valve with an outside lever and weight acting in the closing direction. To eliminate shock pressure, the check valve must close so quickly that the flow has no time to reverse.

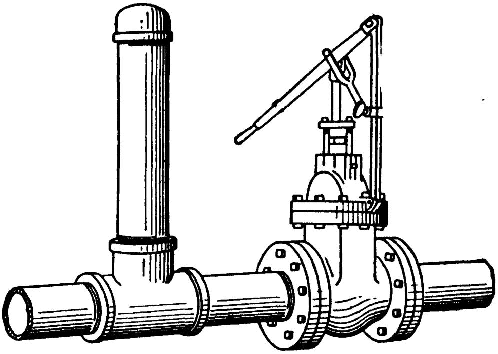


FIG. 15–12.—An air chamber installed ahead of quick-opening valves reduces water hammer. (*Courtesy of The Wm. Powell Co.*)

Lift check valves of the globe type should be used only in horizontal lines, although theoretically they can be placed in lines inclined up to 45 deg from horizontal. For best results, the disk should move in a vertical plane.

It is best practice to use lift checks with globe valves and on steam, air, gas, or general vapor service, provided that there are no severe pulsations in flow. They may also be used on hot- or cold-liquid service if their resistance to flow is not objectionable. For low-pressure air service, soft disks are better than metal ones. On extremely low pressure service, the disk should be fitted with a spring to aid in closing and maintaining a tight seat. With hard-to-hold liquids, use a composition instead of a metal disk.

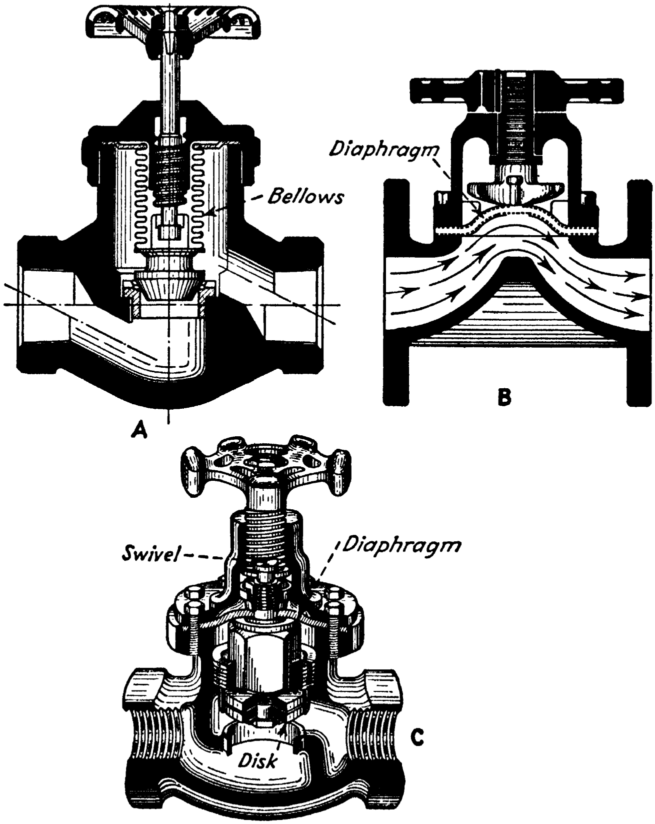


FIG. 15–13.—Different types of packless valves. [*Courtesy of* (*A*) *The Fulton Sylphon Co.*, (*B*) *Hills-McCanna Co., and* (*C*) *The Bastian-Blessing Co.*]

Ball-type lift checks serve on gummy liquids where the ordinary valve fails to perform satisfactorily. Where severe pulsations in flow are present in steam, air, gas, and all liquid service, use cushioned-lift check valves. When quick-acting hand valves are used on liquid lines, install an air chamber or surge absorber to reduce pounding and vibration (Fig. 15–12).

When the valve is to be used for some special purpose where stem leakage cannot be tolerated, the designs illustrated in Fig. 15–13 can be used to advantage. Another way to minimize stem leakage is to use a nonrising stem (Fig. 15–10) or a rising stem that does not turn (Fig. 15–14). In each case, the stem has only one movement instead of turning and sliding as do many valves.

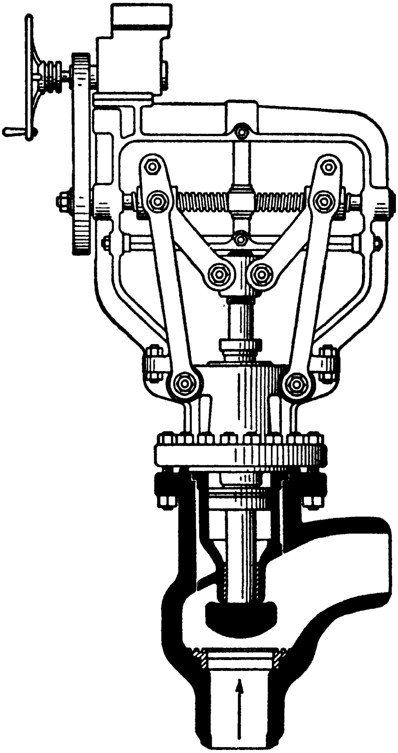


Fig. 15–14.—A rising stem that does not turn. (*Courtesy of Schutte-Koerting Co.*)

The valves in Figs. 15–6 and 15–8 are of the pressure-seal bonnet design. The trend toward higher pressures and temperatures brought about a gradual increase in the weight of the valves. Thus the diameter and thickness of flanges and the number and size of body studs increased accordingly. Higher operating temperatures, particularly above 750 F, introduced problems in maintaining tight flanged joints. Studs not only have to be carefully stressed initially but must be restressed after a relatively short period of operation at maximum temperature to compensate for creep.

Also, leakage occurs at flanged joints in the event of a sudden quench or reduction in temperature caused by a slug of water or saturated steam passing the joint after superheat has been lost. If this periodic quenching is not corrected for by restressing the bolts, leakage rapidly cuts the gasket and flange faces. The new bonnet construction reduces valve weight, prevents leaks, and speeds up the work of dismantling the valve for inspection.

Drawing Up Flange Bolts.—Tightening a flanged connection[1] looks like a simple operation. Moreover, the proper procedure is easy to describe: Pull each bolt up to the same load, and make the total load on all bolts great enough to resist the operating forces and maintain the gasket under compression.

In practice, however, several factors complicate the problem. Getting enough total load on the bolts is not difficult, since general standardization of flanges, number and size of bolts, and wrench length makes it possible for the most inexperienced mechanic to pull the nuts up to a point where the total load on all bolts is enough to resist the design load. Obtaining practically equal loading on each bolt is much more difficult, but the benefits derived certainly justify the effort.

This is particularly true of high-temperature applications where the flanges may bend slowly, gaskets may deform plastically, and bolts may relax. For example, if a flange connection, having bolts originally tightened to loads ranging from 70,000 to 20,000 psi, reaches a temperature of 850 F, the bolts may relax to an operating load of 15,000 psi after several weeks of service. Here the bolts that were originally tightened to 70,000 psi may have accounted for the major part of the total load. However, under high-temperature operating conditions, they can support no more than their relaxation strength of 15,000 psi, and a large portion of the original load is therefore lost. On the other hand, if the tightening had been uniform, the loss in load would have been lower (Fig. 15–15).

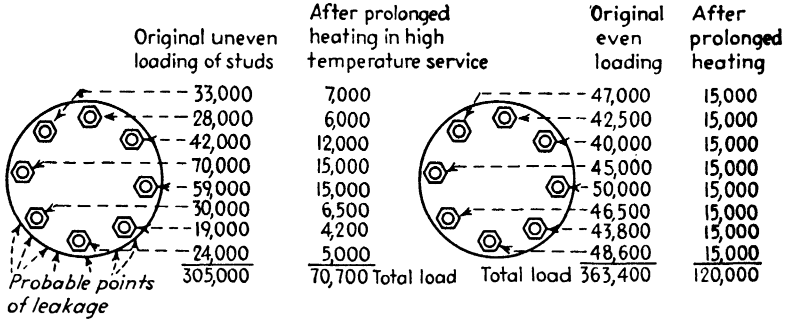
Basically, bolt load, or stress, can be measured by only one method, checking bolt elongation. There are several ways to do this: (1) Measure the bolt length before tightening, and then afterward, thereby determining its elongation. (2) Heat the bolt to use its thermal expansion as a means of creating elongation. (3)

---

[1] CARR, LAURENCE H., Draw Up Flange Bolts Uniformly, *Power*, February, 1946.

Find, by test, the torque required on a nut to give the desired elongation or stress, and then check this torque on each succeeding bolt.

The relationship between the load on a piece of steel and the amount it elongates under that load is a fundamental physical quality. Actual tests show that any steel sample, if loaded well below its yield point, will stretch elastically 0.001 in. per in. of length under a load of 30,000 psi. When this load is removed, the steel returns to its original length. This measurement is like a spring calibration—a measure of elastic stretch under a given load. Fortunately, steel with any degree of hardness, whether carbon, alloy, or stainless, behaves the same way so long as the load is kept below the yield point.



A- Uneven loading, figures in psi

B- Uniform loading, figures in psi

FIG. 15–15.—Evenly stressed bolts maintain a high total load on the flange joint. (*Courtesy of Power.*)

This ratio of load in pounds per square inch, divided by elastic elongation in inches per inch, is called the *modulus of elasticity*. It is surprising, but fortunate for convenience in calculation, that it is so little affected by varying alloying elements and heat-treatments. Strangely enough, a single crystal of iron shows considerable variation in modulus of elasticity along different directions. It goes from 18,000,000 psi in one direction to 42,000,000 in another. The average of these is 30,000,000 psi, and apparently any bar of iron or steel acts as a composite of many tiny crystals whose behavior is the average of the group.

From a practical standpoint, the load up to which a steel sample behaves in this elastic manner is slightly less than the yield point. In other words, if a sample has a yield point of 50,000 psi, it can generally be counted on to give an elastic stretch of 0.0015 in. per in. of length and to return to its original length when this load is removed. If the yield point is 105,000 psi, it can probably be loaded to 90,000 to 100,000 psi. In a fully threaded stud or bolt, there is, of course, a certain amount of stress concentration at the root of each thread; and therefore a little yielding may sometimes take place at loads below the yield point of a smoothly polished bar. As a matter of record, the modulus of elasticity of materials other than steel is subject to considerable variation. Cast iron is about 15,000,000 psi, aluminum 10,000,000 psi, and brass and bronze 10,000,000 to 15,000,000 psi.

The fact that the modulus of elasticity happens to be so constant for steels at room temperature makes it possible to load any steel stud conveniently to a desired load without knowing the analysis, hardness, prior heat-treatment, or even the cross-sectional area at the root of the threads. The desired load need only be divided by 30,000,000 to determine the inches of elongation per inch of length necessary to give that load.

Bolt material commonly used for high-pressure-steel valve flanges is known as SAE 4140 and is heat-treated to conform to ASTM Specification A96–39 Class C and A193–44T Grade B7. This heat-treatment gives a minimum yield point of 105,000 psi; and, even in a fully threaded condition, the material behaves elastically to a stress of about 90,000 psi. This means that any inch of a Grade B7 bolt can be stretched 0.003 in. and returns to its original 1-in. length with the load released. In actual practice, the bolts are given something like 0.0015 in. per in. elongation for a stress of 45,000 psi. This gives a high initial compressive loading to set the gasket firmly in place and stores up considerable springiness in the bolt to take care of subsequent relaxation at high temperatures. However, the loading is not too high to prevent allowances for (1) torsion stresses set up in tightening the nuts; (2) bending that may take place from misalignment between spot-faced surfaces, nut-bearing surfaces, and nut threads; (3) stress concentration at the root of the threads; and (4) possible increase in load in initial service if the flange heats more rapidly and hence expands more quickly than the studs.

Elongation is usually measured across the ends of the bolt. That means the ends must be (1) ground flat and parallel, (2) machined with a small raised section that is ground flat and parallel, or (3) countersunk so that small steel balls or other measuring points can be inserted in the ends. Measuring is done with micrometers, vernier calipers, or specially equipped dial indicators. The measuring device must be readable in thousandths of an inch to be of any practical use.

The fact that the bolts must be measured lengthwise raises the immediate problem that the bolt ends, particularly those portions projecting beyond the nuts, are not stressed. Tests show that the stress in a bolt is reasonably uniform throughout the portion between the two nut faces. At the face of each nut, it begins to fall off fairly regularly until it reaches zero at the top of the nut. In other words, if the main body of the bolt has a stress of 50,0000 psi, the portion at the center of the nut has a stress of about 25,000 psi, and that just projecting from the top of the nut has zero stress.

The total elongation of the bolt within the nut is therefore about half what it would be if that section were uniformly loaded to 50,000 psi. Rather than compute the elongation of the main section of the bolt and the length inside the nut separately, it is more convenient and mathematically just as correct to say that the stressed bolt length is from the center of one nut to the center of the one on the other end and that the stress is uniform throughout this length.

For example, assume we have a 1½-in.-diameter bolt 10 in. long. If the distance between the two opposite spot-faced surfaces is 6½ in., the distance between nut centers on the two ends is 8 in., because the height of a nut is always the same as the nominal diameter of the bolt. If we desire a stress of 45,000 psi, each inch of stressed length should be elongated 0.0015 in., as outlined previously. We consequently multiply this figure by 8 in., the distance between nut centers, to obtain an elongation of 0.012 in. In other words, the original bolt length, as measured by micrometer, should be increased 0.012 in. by tightening the nut.

The constant-torque method of tightening bolts really just supplements the elongation method just described. For reasonable accuracy, it is necessary to tighten sample bolts to the desired elongation while measuring the torque required. Once established, it can be repeated on other bolts, and presumably they will be given the same elongation and stress. Torque can be measured either by one of the many torque wrenches manufactured or with a spring scale attached to a wrench of known length. In foot-pounds, torque is equal to the length of the wrench in feet times the pounds pull on the end of it.

A simple table correlating stress and torque for a variety of common bolt sizes would be ideal, but unfortunately there are too many variables to make such a table practicable. The extent to which the torque on a nut is converted to stress in a bolt depends on (1) the finish of the nut face, (2) the nut thread, (3) the flanged face, (4) the bolt thread, (5) the thread alignment, (6) the thread fit and form, and (7) the degree of lubrication on the sliding surfaces.

For example, a torque figure set up for an unlubricated nut might be sufficient to fracture a bolt in tension if the nut were well lubricated. However, under fairly constant conditions, the torque method of stressing is quite valuable. In any one given plant, where (1) bolts are made of one material on one machine and inspected to a fairly constant degree of finish, (2) the spot-faced surfaces of the flange are given a constant finish, and (3) lubrication is controlled, it is possible to give nuts a certain torque value in tightening and obtain bolt stresses that are constant within plus or minus 5 or 10 per cent.

An initial stress of 45,000 psi in a bolt may seem high, particularly if one were to consult the table of allowable stress values in the ASME Boiler Code and see a figure of 16,000 psi for the same material. This is offset, however, by the fact that the bolts themselves relax, and the flanges may bend toward each other slightly because the steel creeps in elevated-temperature service. Also, the gasket may flow out into voids in the flanges to become actually slightly thinner. These considerations lead directly to the fact that this quality of springiness in the bolts is necessary.

The 0.012-in. elongation imparted to the previously mentioned example is more or less a measure of the ability of the bolt to follow any bending of the flange before completely losing its load. As an example, the flange could give, say, 0.006 in., and the bolt load would be down to about 22,000 psi. Mathematically, the problem is slightly more complicated, but this rough calculation is approximately correct. Hence long bolts are advantageous; because, in reaching the same initial stress, they store up more latent elongation, or spring.

Comparing the $1\frac{1}{2}$-in. bolt, 10 in. long, with a studded rather than a through-bolted flange joint, we might find the stressed length of the stud reduced to 5 in. Here, a stress of 45,000 psi would be obtained with an elongation of 0.0075 in. elastic stretch (45,000 psi $\div$ 30,000,000 $=$ 0.0015 in. per in. $\times$ 5 $=$ 0.0075). If the flanges bend toward each other 0.006 in., the remaining elongation is only 0.0015 in. in a 5-in. length, which is about 0.0003 in. per in. (0.0075 — 0.006 $\div$ 5 $=$ 0.0003), equal to a stress of 9,000 psi (0.0003 in. $\times$ 30,000,000) rather than 22,000 for the through-bolted type.

These principles can be followed to measure the stress in a bolt after a period of elevated-temperature operation. The obvious way is to measure the bolt across its ends and then loosen the nut and remeasure to find the extent of its elastic contraction. Hence, by calculation with the modulus of elasticity, the change in stress from the tight to the loosened condition can be determined. Tests of this nature yield a great deal of valuable information, but they must be interpreted with caution.

For one thing, the modulus of elasticity of steel unfortunately changes with elevated temperatures. The modulus of 30,000,000 psi at room temperature changes to something like 25,000,000 psi at 850 F, and 22,000,000 to 20,000,000 psi at 1000 F. This means that, for an elastic elongation of 0.001 in. per in. at 1000 F, the stress is from 20,000 to 22,000 psi. If the flange and bolt assembly were at a uniform temperature of 1000 F and contained elastic elongation and if the entire assembly were then cooled to room temperature, the stress in the bolt would actually increase with cooling even though the elastic elongation within it did not change.

Another confusing factor is the possibility of a thermal difference between the bolts and the flange. If the flange material is 100 F hotter than the bolt material, it will expand 0.001 in. per in. more than the bolt, thereby creating an additional stress in both bolt and flange. If we assume that half of this was transferred to the bolt in the form of elastic elongation and half to the flange in the form of elastic bending, the increase in bolt stress is about 10,000 psi at the elevated temperature. These complications demonstrate that it is difficult to estimate operating-temperature bolt stress on values obtained at room temperature.

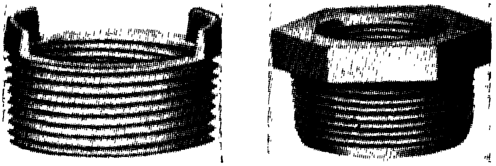


FIG. 15-16.—Select a bushing that gives a good strong pipe joint. (*Courtesy of Crane Co.*)

Pipe Bushings.—When selecting pipe bushings, remember the important distinctions between the face and the hexagon types (Fig. 15-16). If a poorly made hex bushing is screwed into an opening, the hex flange and several threads remain outside. When a pipe or other male-threaded fitting is screwed into such a bushing, the end may be just flush with the face of the original opening after the joint is tight. There is therefore no actual engagement between the original opening and the connected pipe except the thin wall of the bushing. Any excessive stress may snap the bushing at this point.

In well-made hex bushings, this fault is eliminated by holding the outside thread to the smallest code tolerance, while the inside thread is held to the code's largest permissible tolerance. The result is a pipe-lap backup of the bushing wall that is at least equal to the pipe thickness.

Contrasting with this, a face bushing screws into an opening until it is about flush with the face of it. A screwed-in pipe makes a normal thread engagement, thus forming a full-threaded lap with the original opening and getting the full benefit of that reinforcement.

Installing Valves.—Good service cannot be expected if a valve is subjected to abuse during storage or damage from poor installation practices. When shipped from the factory, they are amply protected against the entrance of foreign matter by complete wrapping or by wood or thin metal caps attached to the open ends. Keep this protection in place, and store the valve indoors until it is ready to be attached to the pipe. Storage on the ground exposed to the weather is likely to result in penetration of sand into the working parts. Valves can be easily damaged by rough handling. They should be placed where they cannot fall and where other material cannot fall on them.

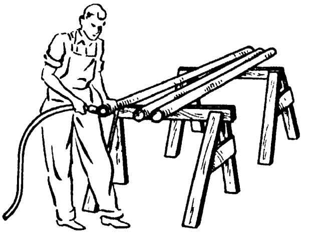


FIG. 15–17.—Clean piping and valves with compressed air before erection. (*Courtesy of Crane Co.*)

Preparing the valve for installation includes the removal of protective covering and the cleaning of all grit and dirt from the inside. Cleaning the pipe is just as important and necessary, if damage to the seat and disk is to be prevented. The valve can be blown out with compressed air or flushed with water; clean the pipe in the same manner (Fig. 15–17), or pull a swab through it to remove dirt and metal chips left from threading operations. Pipe cleaning must be done with care to ensure the removal of all scale and other loose material before the valve is attached.

Connection can be made to the line by screw threads, welding, or flanged couplings. Properly cut male and female threads are exact counterparts of each other and, when made up into a joint, form a perfect metal-to-metal contact that is tight. The secret of making a tight joint is to cut good threads rather than to fill bad threads with compound. Nicked or worn dies will not make a clean cut. Lack of or an insufficient quantity of cutting oil or improper handling of the die stock produces rough and torn threads. An extreme case is one in which short lengths of thread are torn completely away from the body. Slow die travel, plenty of cutting oil, and frequent reverse movement of the die to remove chips, when cutting by hand, produce the best job. Power-driven dies need plenty of cutting oil because of their steady, continuous speed. Cut only enough thread to make a solid joint; do not permit the pipe end to travel into the valve far enough to bear against the seat or diaphragm. Always ream the cut end of a pipe (Fig. 15–18).

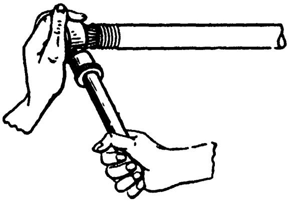


FIG. 15–18.—Ream pipe ends after cutting to remove burrs. (*Courtesy of Crane Co.*)

The only good reason for using thread compound is to furnish lubricant when making up the joint. Because of dry-metal friction, the threads will tear during engagement unless they are lubricated. Always apply the compound to the male thread (Fig. 15–19) so that it will not be forced ahead of the pipe into the valve. When making up the joint, apply the wrench to the valve end adjacent to the pipe (Fig. 15–20); attaching the wrench to the opposite end strains and twists the body. Excessive make-up of threaded joints is not necessary; the joint should not be tightened until it squeals.

Welding pipe-line and fitting joints saves time and material and produces a continuous leakproof system. When welding valves to the line, wrap them with wet rags; the appearance of steam indicates that the valve temperature is too high and that welding should be discontinued until steaming stops. Without this indication, welding might be continued until the valve temperature reached such a point that unequal expansion would throw the seat out of line.

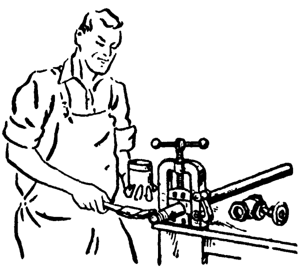


FIG. 15–19.—Put compound on male threads only, and keep it off end of pipe. (*Courtesy of Crane Co.*)

When attaching flanged valves to pipe lines, align the pipe flanges on the same center line with their faces parallel before inserting the valve. If a line is properly supported, the flange faces will remain parallel and on the same center line. In drawing up flange bolts, tighten them slowly in diametric pairs, moving around the flange, as in Fig. 15–21, until the joint is true and square around its circumference. All bolts can then be drawn up securely.

Future repair troubles can be minimized by mounting the valves properly, protecting them against outside damage, and locating them at the most suitable point in the line. Except for the split-wedge and double-disk gate valves, the majority can be mounted at any angle from the vertical position. It is always better, from the standpoint of the valve, to mount it with the stem pointing upward. The second best choice is a horizontal position, since any position lower than this brings the bonnet directly under the line of flow to serve as a pocket and catch pipe scale or other foreign matter. This soon cuts and destroys inside stem threads; or moisture trapped at this point, on lines exposed to freezing temperatures, causes frozen and burst bonnets. Even when valves are mounted

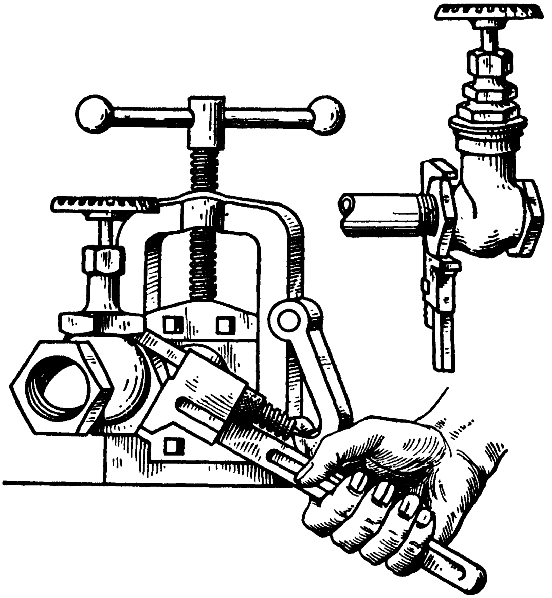


FIG. 15–20.—Apply the wrench to the pipe side of the valve to prevent twisting the body. (*Courtesy of Power.*)

upright, on lines drained during freezing weather, take the precaution of opening a drain plug in the bottom of the valve body.

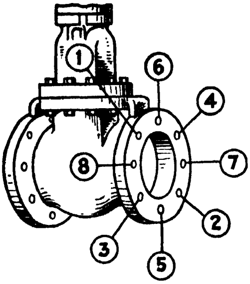


FIG. 15-21.—Tighten flange and bonnet bolts in diametrically opposite pairs. (*Courtesy of Crane Co.*)

After investigating the effects on a valve of being installed in one of several positions, consider which would be the best from an operating or inspection standpoint. If you want the valve opened and closed correctly, the operator must have reasonable and substantial access to it. When a valve is mounted high overhead, a person cannot exert any great amount of force while standing on a ladder in a precarious position. Eliminate such conditions by installing the valve horizontally, and fit it with a chain or rope drive hanging to the floor level (Fig. 15-22). Overhead valves on large lines, requiring two men to operate, are better installed vertically so men can stand on the pipe while operating the valve. Of course, power operators are much better for this kind of valve, and a working platform is better to stand on than the pipe. When valves are too large to operate with a chain pull, it is no advantage to mount them horizontally.

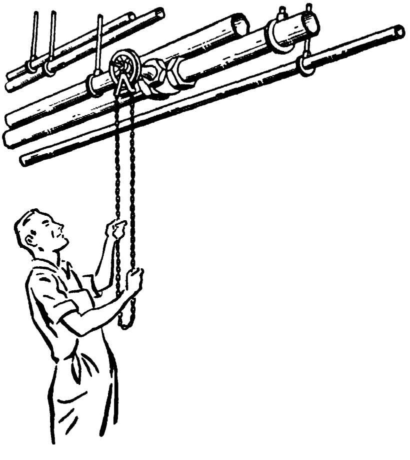


FIG. 15-22.—Install a pull chain on overhead valves. (*Courtesy of Crane Co.*)

If the valve position can be varied within reasonable limits, place it in the most accessible position for ease of operation and inspection (Fig. 15-23). Additional headroom is required for the rising stem (Fig. 15-24), but all valves need sufficient room to permit removing the bonnet and stem. It is much easier to keep the valve in place and remove only the internal parts for inspection and cleaning. Laying out a pipe along a wall that brings the valve just within fingertip reach is not good practice. In many cases, a person, rather than hunt for something to stand on, stretches just enough to reach the handwheel. The valve is therefore not closed tight and consequently soon begins to leak.

Sometimes valves and piping are installed on machines erected close to walls or other pieces of equipment. This may put the valves in a position where they cannot be reached unless a person can squeeze into the small space. Conditions in that particular crevice are going to affect the treatment that the valves receive. Many machines are of such shape and size that mounting the valves on the wall directly behind the machine places them in the most inaccessible position possible. If a workman cannot reach them from the floor and if there is no sitting or standing room on the machine, a special platform must be constructed for the repair, in many cases, of only one or two small valves. Easy access encourages good care of hand-operated as well as automatic valves.

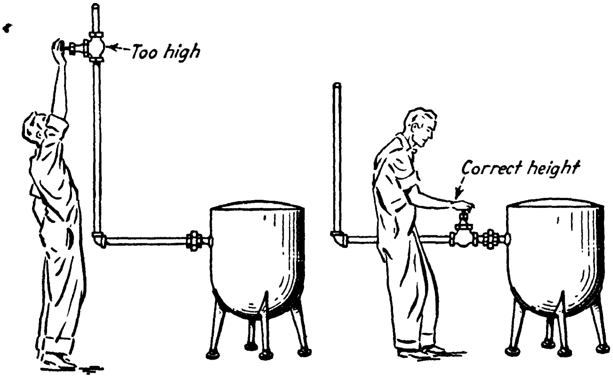


FIG. 15-23.—Locate valves where they can be operated easily. (*Courtesy of Crane Co.*)

Check valves give considerable trouble unless they are positioned in the line properly. Install swing and tilt checks with the hinge pin horizontal so that the disk will close by gravity; lift checks must be placed so that the lift is vertical. Swing check valves work satisfactorily in horizontal lines but are not fast enough for vertical risers handling liquids. In one case where a swing check was located in the foot of a 40-ft riser, the water hammer opened several pipe joints. Vibration caused horizontal sections of the pipe to swing several inches out of line. The trouble was remedied somewhat by reducing the opening travel of the disk with a setscrew threaded through the cover. This kept its outer edge in the line of flow and its center of gravity more to the closing side of the hinge pin. Quicker closing resulted in less water hammer.

Pulsating flows soon wear out swing check valves, because the disk movement is not cushioned. Spring-opposed lift check valves, when used on temperatures above 300 F, must be fitted with springs of metal able to withstand the heat. In air-compressor service, ordinary springs can be used, but the valve must be installed beyond the aftercooler.

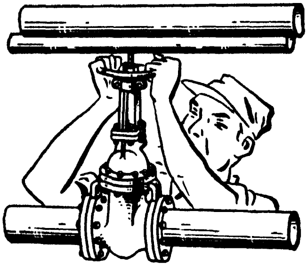


FIG. 15-24.—Give a rising stem plenty of clearance. (*Courtesy of Crane Co.*)

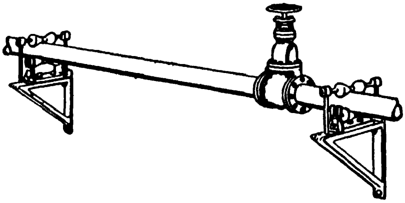


FIG. 15-25.—Place hangers so that the valve does not have to support the pipe. (*Courtesy of Crane Co.*)

For pipe lines handling sludge or other suspended matter, keep valves out of vertical lines whenever possible. Stoppage of flow allows the suspended matter to settle and choke a closed valve. This is especially troublesome on check valves. Temperature changes in rigid pipe lines set up strains and stresses in a valve body. Liquid weight, causing pipe sag, strains the valve, throws the seats out of line, and causes leakage. Support the pipe adequately and independent of the valves on wall brackets (Fig. 15-25) or ceiling hangers (Fig. 15-26). Where excessive strains exist, provide expansion joints or loops for protection (Fig. 15-27).

How to direct the flow through a globe valve can be determined only by a complete diagnosis of the system, including what would

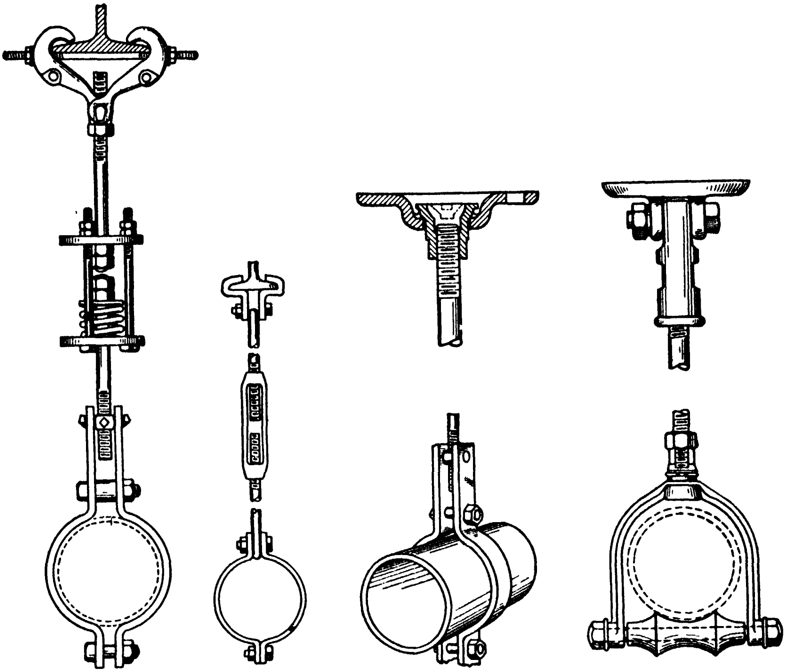


FIG. 15-26.—Select one of the many types of pipe hangers to support the pipe line adequately. (*Courtesy of Power.*)

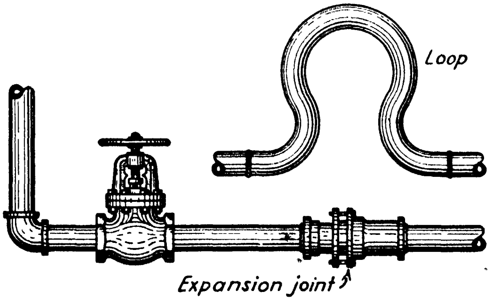


FIG. 15-27.—Install either a loop or slip joint to take care of expansion. (*Courtesy of The Wm. Powell Co.*)

happen if the disk came loose from its stem. The ASME Boiler Code requires that globe valves, installed in boiler-feed lines, have the pressure under the disk so that, if this happens, the disk will not shut off the water to the boiler.

If a valve is persistently left cracked open, reversing it and putting the pressure above the disk will assure tighter closing. When used as a drain, with pressure under the disk, the valve will vibrate open if it is not closed tightly. On the other hand, if the valve is controlling a piece of equipment that may overspeed, applying the pressure over the disk forms a check valve that will shut the machine down if the disk comes loose.

In such operations as boiler blowdowns, soot blowers, and drain legs, pressure on top of the disk is proper, since any of these services may be delayed without hazard. Therefore, unless the service clearly indicates that pressure should be under the disk, apply it on top.

Valves of large size, 12 in. and over, have disks with generous areas exposed to line pressure. Obviously, this pressure, when exerted against the disk in an unbalanced service, such as discharging from high to low pressure, makes valve operation increasingly difficult as the exposed disk area increases. This condition is less pronounced in gate than in globe valves, because the disk movement in the former is across the line of flow, whereas in the latter it is directly against flow. To minimize this condition, equip all gate valves 12 in. and over, and all globe valves over 6 in., with by-pass valves. Use only a throttling-globe valve for by-pass service.

Watch every valve installation for ways to improve and simplify maintainance work. When insulating the pipe line, never extend the insulation in a solid sheet up and over the valve bonnet. Apply the layer only up to the underside of bonnet bolts. If the bonnet must be covered, use preformed insulation that can be removed easily; otherwise internal inspection may be delayed because of hesitation in breaking a perfect-looking insulated joint. Sometimes drain valves are plugged to prevent inadvertent opening and loss of material. When done on bronze valves, use a pipe nipple and cap in preference to a solid plug. If the valve is subjected to wide variations in temperature, unequal expansion with the solid plug cracks the valve body.

A last but still important job, before considering the installation complete, is to go over the entire system, after it has reached operating temperature, and tighten all bonnet and flange bolts. This is also a good opportunity to inspect the valve-stem packing for leakage and make the necessary adjustments. Good judgment, a thorough knowledge of valve characteristics, and careful workmanship will produce a piping and valve layout that is economical, serviceable, convenient, and easy to maintain.

Valve Maintenance.—Neglecting a valve until it must be replaced or fitted with new parts not only wastes material, but the leaks are bound to show up in higher operating costs. Frequent inspection will uncover leaks before the trim is greatly damaged, reducing the repairs to that of seat refacing. Early repairs reduce the amount of metal that must be added or removed in forming a leakproof seat and prevent the conditions shown in Fig. 15–28.

A good maintenance program includes regular and systematic inspection to catch trouble before it grows serious, proper lubrication of rotating and sliding parts, replacement of stem packing when leakage or excessive friction develops, and refacing leaking seats or disks. Stem threads and thrust washers must be lubricated and disk-spacing wedges or cams kept free of corrosion, incrustation, or other foreign material. Plug cocks, with their large metal surfaces, require frequent lubrication to prevent galling and freezing of the parts.

Scored valve seats and disks can be built up by hard soldering, brazing, or welding, but the method used depends on the seat or disk metal. Rather than discard badly eroded bronze seats, build them up with hard solder or bronze rod; alloy-steel seats for temperatures below 750 F can be repaired by brazing, which, although not so resistant as the original metal, provides a serviceable valve. Building up alloy trim with supposedly identical metals can easily lead to trouble unless the kind and hardness of the parent metal are definitely known.

When new parts are needed, buy seats and disks in pairs so that only a minimum amount of grinding is needed. Salvage all good parts of damaged valves for use as spare parts to rebuild others. A knowledge of valve construction together with the ability to dismantle and reassemble one properly simplifies the job of making

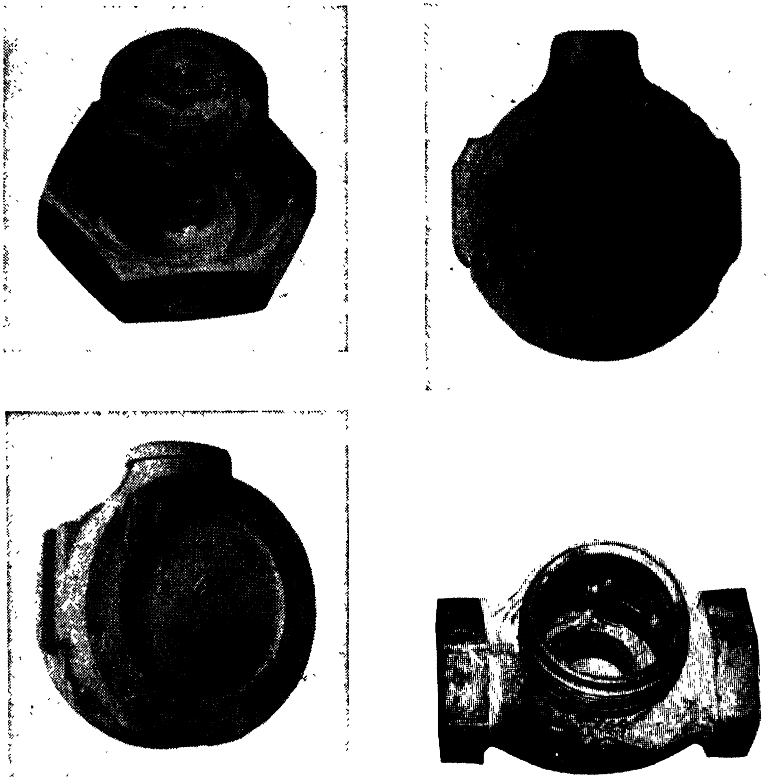


FIG. 15-28.—Proper inspection and maintenance can prevent damage of this kind. (*Courtesy of Jenkins Bros.*)

repairs. Any careless insertion of the stem packing or rough handling of the seats and the disk can cause much additional trouble.

Before starting to remove a bonnet, turn the valve to its open position so that no bending stress is placed on the stem when the bonnet is loosened. This is more pronounced in a bolted bonnet. Also put the stem in the open position before replacing the bonnet.

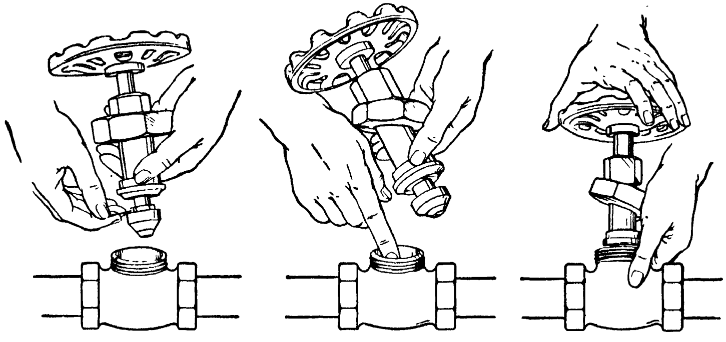
U-clamp gate-valve bodies are easily split in half by tightening the bonnet into place with the wedge in the extreme closed position. Union bonnet gate-valve seats can be sprung apart in the same manner, making the valve useless.

Proper maintenance of stuffing-box packing and correct adjustment of the packing nut are essential to satisfactory stem life. Packing that must be tightened until it is almost impossible to turn the valve stem is either not suitable for the service or has become dry and hard.

In either case, it should be discarded; otherwise, an additional burden is imposed on the stem threads that rapidly shortens their life. Another effect of excessive packing pressure is that it causes the operator to be uncertain whether the valve is fully seated. As a result, it is frequently seated, either manually or with a wrench, tighter than need be. This excessive closing effort, to ensure tight seating, also injures the stem threads. Stem packing, after being subjected to high-temperature steam and then allowed to cool, leaks a small amount when the line is again put in operation. This does not necessarily call for adjusting the stuffing nut or gland. As soon as the valve becomes hot, it will in most instances stop leaking. Expansion and contraction of the bonnet, stem, packing, gland, and stuffing nut cause this condition.

When packing a valve stem, use material that fits. Preformed rings are preferred, because they enter easily and draw up more evenly than coil packing. When winding coil packing spirally around the stem, force it to the outer edge of the stuffing box instead of wrapping it tightly to the stem. After adding the maximum number of rings, draw up the gland evenly with a wrench until the packing is forced into a snug position. Then slack off and make the nuts finger-tight. A good valve stem, sufficient packing of the right kind, and a finger-tight gland will hold on all moderate pressures. Higher pressures require a tighter gland to prevent the pressure from getting under the packing. Many valves contain a back seat that closes off the packing gland against pressure when the valve is open. This permits packing the stem while the valve is in service, but be certain of this feature before attempting to repack it under pressure.

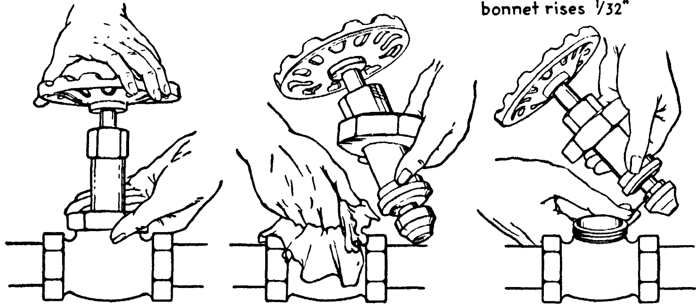
Grinding or refacing seats can be done in many ways. Sometimes only a light lapping between the seat and disk is needed, and at other times metal must be added or the surface machined. The simplest grinding operation can be performed with the union bonnet globe valve. It is only necessary to pin the stem and disk together, through a hole already drilled, and use the bonnet as a guide during the grinding operation (Fig. 15–29). The same job with a screwed bonnet is more difficult, because the bonnet cannot be used as a guide. Here the job requires a drill press or grinding



Rotate disk to line up locknut slot with hole in stem. Insert pin or nail to lock disk and stem together

Apply grinding compound to seat and disk with finger. Use fine-grain compound and keep coating thin

Set bonnet assembly in body and turn handwheel until disk touches seat. Continue turning until bonnet rises 1/32"



Screw union handtight, then back off one turn. Oscillate handwheel and bonnet: occasionally turn to a new spot

After grinding surfaces to a good seat, wipe seat and disk with a cloth wet with gasoline or other solvent

Be sure to clean away all grinding compound from seat and disk. Remove pin from hole and reassemble valve

Fig. 15–29.—Follow this method when regrinding globe valves. (*Courtesy of Power.*)

kit (Fig. 15–30). Insert the stem (after removing it from the bonnet) in the chuck. Fasten the body in the vise so that the seat is level. This can be determined by leveling the top of the bonnet edge because the two faces are parallel. Pin the disk and stem, apply compound to the disk, and lower it into place against the seat. Rotate the stem at low speed.

When lapping stainless-iron disks against similar seat rings, mix white lead and oil with the grinding compound to furnish lubricant; otherwise the metal will drag and ruin the surface. Using a compound of small grain size reduces this action. (Stainless iron has an inherent tendency to gall and seize; the fragments of metal, loosened by grinding, pile up and drag so that continued grinding only aggravates the initial roughness.) Use light strokes lifting the disk frequently to a new position, and clean the surface often. On heavy parts, suspend the valve stem from a spring. Grind stellited seat rings and disks the same way, except that, in some cases, it may be necessary to use silicon-carbide grit as the grinding agent.

FIG. 15–30.—A grinding jig for screwed-bonnet globe valves.

One disadvantage in grinding seats and disks against each other is that, where alloy parts of different hardnesses are used, the softer metal will grind away much faster than the harder one. Usually the disk metal is of harder material to prevent forming seat-impression shoulders on the disk.

A more difficult grinding operation is presented by the balanced-pressure double-seat valve. Since such valves are designed for regulating service, they should not be expected to be absolutely leakproof on shutoff and should not be depended on for tight closure from a safety standpoint. When grinding, both seats must be established at the same time. Impressions on the disk indicate which seat is touching; put compound on this disk with just a trace on the other, and grind until both seats are established uniformly. To produce a true seat under operating conditions, the valve body and stem must be kept at operating temperature while grinding.

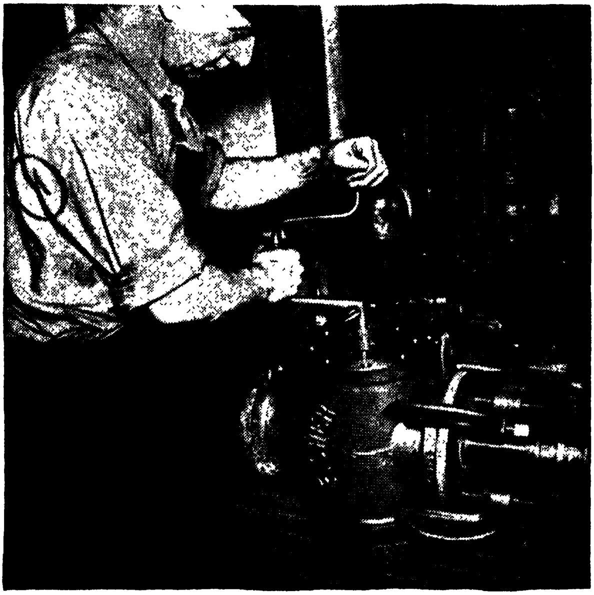


Fig. 15–31.—Hand grinding a double-seat regulating valve. (*Courtesy of **Fisher** Governor Co.*)

If the seat or disk is eroded considerably, even when the eroded areas can be built up, it will be necessary to machine the surface to eliminate prolonged grinding. On integral seat valves too small to insert soldering or brazing torches, the seat will in time be machined down to the diaphragm surface. When this happens, ream the seat opening and thread it to take a renewable seat ring; but never use stainless iron of the same hardness for both the disk and the seat ring unless the valve is to handle oil.

Repeated refacing of shoulder-type seat rings reduces the shoulder thickness to such a point that the metal contact pressure against its underside causes a concaving of the outer, or seating, surface. The only remedy is to replace the ring or increase its thickness by

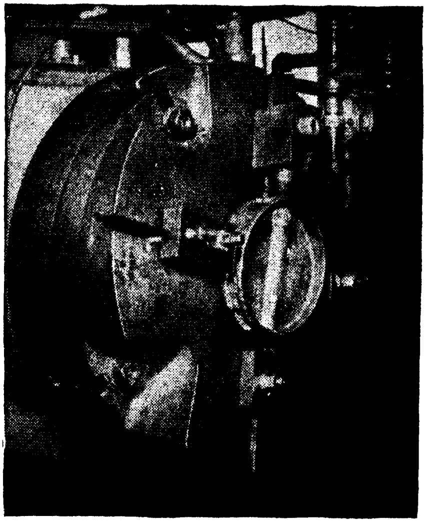


Fig. 15–32.—Special lathe faceplate for repairing gate-valve disks. (*Courtesy of Jenkins Bros.*)

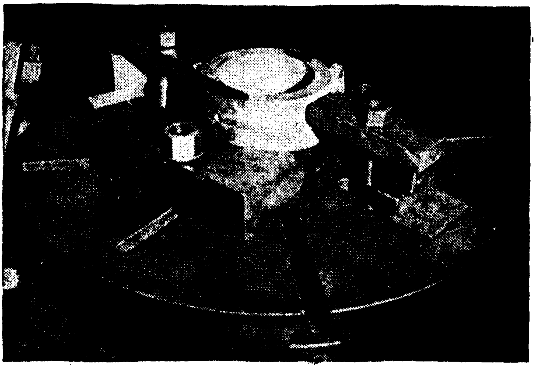


Fig. 15–33.—Homemade wedge block bolts to conventional lathe faceplate.

welding or brazing. Leakage past seat-ring threads is another trouble that occurs frequently. The leak must be repaired immediately by welding and rethreading or by reaming and threading to take an oversized ring. If the damage is too great for this, the only remedy is to weld the ring solidly in place.

Gate-valve seat rings can be refaced by placing the body in a drill press. Lathe machining, though possible on parallel-seat bodies, cannot be used for wedge-gate bodies unless special tapered faceplates (Fig. 15–32) or a wedge block (Fig. 15–33) is used. Before using a drill press for this purpose, make sure the spindle is plumb so that, when the work surface is level, the drill spindle will be perpendicular to it. Then place the body on the work table, lay a machinist's level across the seat ring, and insert wood wedges under the body flange until the seat ring is level (Fig. 15–34). A piece of emery cloth cemented to a metal disk is inserted through the bonnet opening and placed on the seat (Fig. 15–35). A square

FIG. 15–34.—A drill press can be used to reface gate-valve seats. (*Courtesy Standard Oil Co. of N. J.*)

FIG. 15–35.—Emery-cloth disk is laid on the seat and rotated by the spindle. (*Courtesy of Standard Oil Co. of N. J.*)

shank on top of this plate loosely mates with a socket held in the drill spindle and serves as a flexible driving connection; this allows the grinding plate to bear evenly on the seat ring face.

A drill press can be used to reface the wedge except that the leveling requires a tapered plate having the same surface area as the valve wedge (Fig. 15–36). Jigs milled to a tapered depth forming a snug fit with the perimeter of the wedge hold it from turning but necessitate the construction of a jig plate for every style or size of gate wedge (Fig. 15–37). One flat tapered plate can be made to serve many sizes of gate wedges and can be adapted for use on either a lathe or a drill press, because the majority of gate wedges are built with an over-all taper of 10 deg. Extreme care must be exercised in using this method, however.

Fig. 15–36.—Gate valve disk rests in a jig that lies on a rubber pad. (*Courtesy of Standard Oil Co. of N. J.*)

Fig. 15–37.—If jigs are used, it is necessary to have one for every type of gate valve.

Before clamping the gate wedge to the taper plate, inscribe a line accurately vertical across the wedge face. This line must coincide with a center line drawn lengthwise with the taper of the plate (Fig. 15–38). Any misalignment between the wedge and plate lines, while refacing, destroys the gate taper.

FIG. 15–38.—Center lines must be scribed on the valve gate to match those on the wedge block. (*Courtesy of Standard Oil Co. of N. J.*)

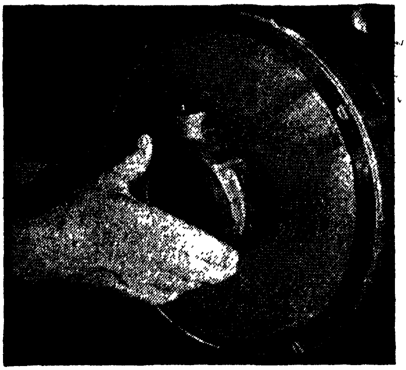


FIG. 15–39.—A disk sander can be used to resurface valve gates. (*Courtesy of Jenkins Bros.*)

FIG. 15–40.—The valve gate can be cleaned on an emery-cloth-covered surface plate. (*Courtesy of Jenkins Bros.*)

The wedge disk can also be refaced by holding the surface against a motor-driven grinding disk (Fig. 15–39). Use care to keep the wedge centered on the disk and to hold it with uniform pressure, or the taper will be ruined. Rubbing the wedge on an emery-cloth-covered surface plate (Fig. 15–40) serves the same purpose.

After completing the resurfacing operations, clean the valve body of all grit and metal cuttings by blowing it with compressed air, and then place it in a vise or other suitable holding device. Cover the gate wedge faces with a thin oil film, lower it in position, and slam it home (Fig. 15–41). Careful removal will show any uneven bearing surface between the wedge and the seats. If contact is uniform around the entire seat, reassemble the valve and add new stem packing as required.

Fig. 15–41.—To get seat impression, coat the gate with oil and drop it lightly against the seats. (*Courtesy of Standard Oil Co. of N. J.*)

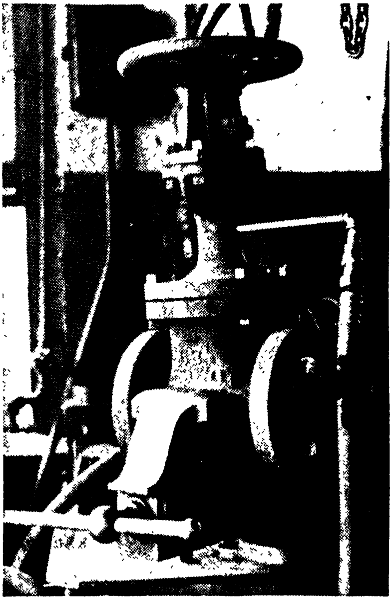


Fig. 15–42.—Admit compressed air to the bonnet to check seat leakage. (*Courtesy of Standard Oil Co. of N. J.*)

After assembly, the final testing consists in applying air pressure through a hole previously drilled in the bonnet (Fig. 15–42). Although 100 psi air pressure is used, experience demonstrates that, if a valve does not seat properly, it will leak even on low pressure. Applying the pressure at this point tests the bonnet gasket and stem packing, as well as each seat individually. This assures that the valve will be in good condition before it is delivered to the job.

When seat-ring removal is necessary, use extreme care to prevent unnecessary damage. Makeshift arrangements invariably lead to broken seat lugs or burred grooves. If conventional wrenches are not available, homemade ones will work satisfactorily if they are accurately constructed (Fig. 15–43) so that they bear uniformly on all seat lugs or grooves and are substantial enough to remove and replace the seat. Manufacturers screw the seats into place with machine power, and any oxidation makes them extremely difficult to remove unless sturdy wrenches are used. If a seat ring appears

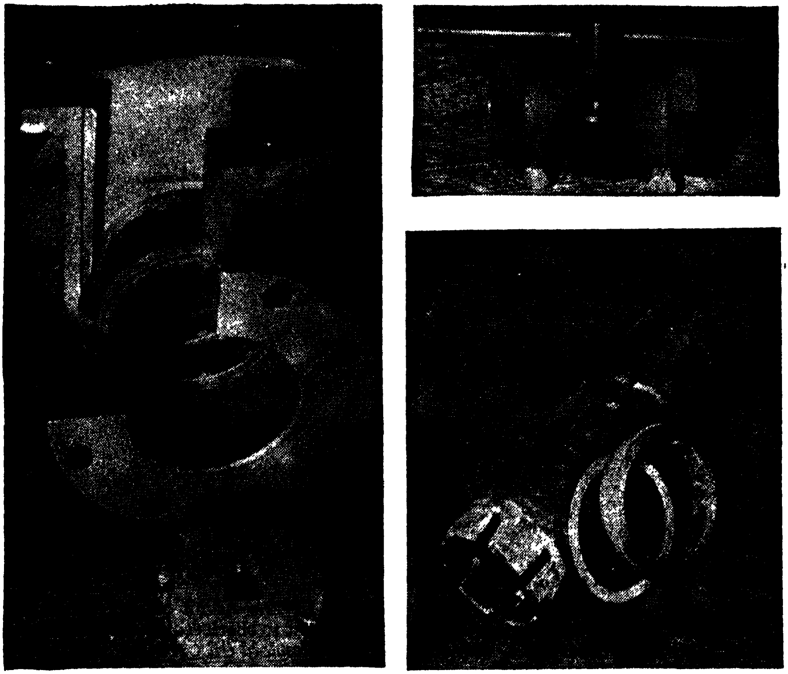


FIG. 15–43.—Use strong wrenches to remove seats. (*Courtesy of Lunkenheimer Co.*)

abnormally tight, it may be found to be soldered or tack-welded in place.

When new seat metal must be added to a wedge, the welded-on overlay is applied to one side at a time. This allows the use of the opposite side as a bearing against the tapered plate during machining to ensure maintaining the same wedge taper. After one face is welded and machined, the opposite side is completed in the same manner.

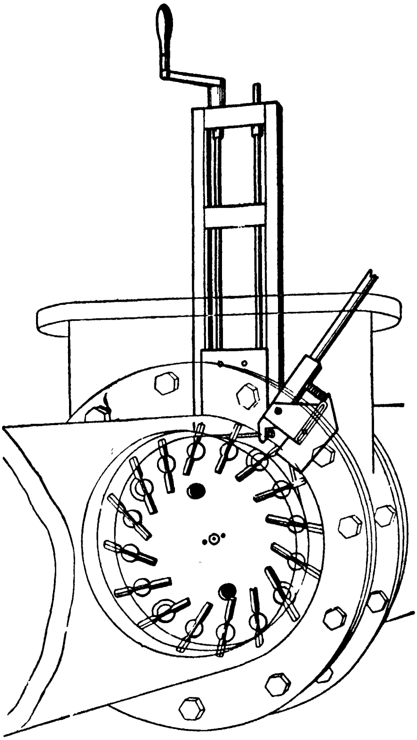


Fig. 15-44.—Tool kits are available for refinishing valve seats in place.

Valves subjected to certain corrosive conditions soon become covered with tubercles, which build up around the seat and disk rings. If allowed to increase, they soon creep over the seat edges and prevent the valve from closing tight. Cleaning by sandblasting and applying a good paint or metallizing the areas around the seats will greatly retard this growth.

Another trouble often found is pitting on the inner walls of gate valves. These cavities, similar to those found on hydraulic turbine and pump impellers, appear at the bottom between the two seats as well as on the side walls below the bonnet joint. Cavitation is suggested as being the cause of this pitting rather than corrosive action of the fluid.

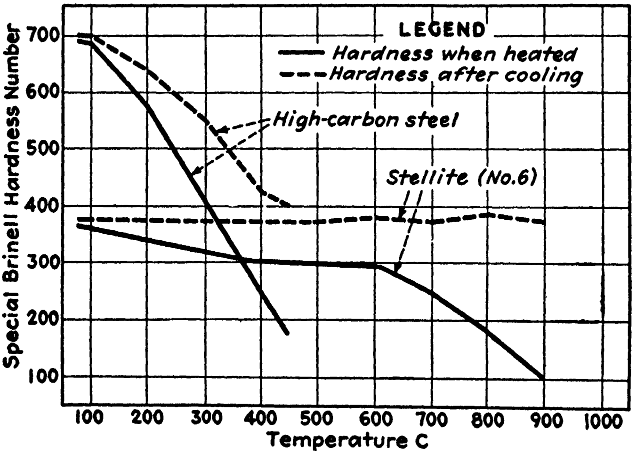


FIG. 15–45.—Hard-facing material (stellite) retains its hardness at red heat. (*Courtesy of Haynes Stellite Co.*)

All the repairs described so far require that the valve be removed from the pipe line. Valve-reseating kits are available with which the seats can be refaced without removing the body from the line (Fig. 15–44). This eliminates the need of breaking pipe joints, reduces the possibility of leaks, and saves the time and labor for removing and reinstalling the valve.

Repairs to valves, as to any other plant equipment, require skill and training. Keep in mind that a valve is a machine, although in many instances a simple one, and give it the same care that you would any other machine in the plant.

Hard-facing Valve Trim.—Hard-facing offers a useful maintenance tool for protecting steel valve parts against severe abrasive wear and wire-drawing action. Cobalt-chromium-tungsten alloys (stellite) retain their hardness at red heat (Fig. 15–45) and are therefore particularly suited to surfaces in friction and to parts exposed to high temperature.

Before applying the hard-facing alloy, prepare the part by grooving the surface to a depth of from 3/32 to ⅛ in., leaving a ridge of metal on each side of the groove. Then round off all sharp corners by filing, grinding, or machining (Fig. 15–46). This rounding off is necessary to prevent interalloying with the base metal, because sharp edges melt easily during the application of the alloy. This dilution with iron results in decreased wear resistance and frequently causes blowholes. Where the seat is too narrow for grooving, machine it flat, and round off the edges. After grooving, thoroughly clean the surface by grinding or machining until it is free from all dirt, scale, grease, or other foreign material. The surface on which the hard-facing alloy is to be deposited must be clean.

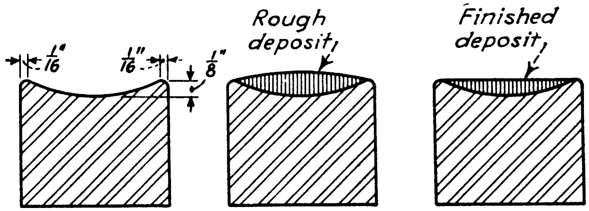


FIG. 15-46.—Groove or recess the base metal before applying the hard-facing metal. (*Courtesy of Haynes Stellite Co.*)

If the part is small, under 3 in. in diameter, and the welding flame is sufficiently large to keep it red hot during welding, no furnace preheating is necessary because the preliminary heating can be done with the flame. However, if the part is large, it must be preheated in a temporary furnace (Fig. 15–47). Bring the temperature slowly up to from 800 F to 1200 F, or just under the point where the steel begins to scale. This is a faint red heat, just visible in a dark room. If possible, deposit the metal while the valve part is in the furnace; when this is impractical, reheat the part whenever it cools to 600 F. Fixtures can be purchased or made that will rotate the work during the hard-facing, but such a device is not absolutely necessary; a helper can turn the part (Fig. 15–48).

Hard-facing requires an excess acetylene flame. Adjust the acetylene feather length, measured from the blowpipe tip, to three times the inner-cone length measured from the welding-tip end. This flame prepares the steel surface by melting an extremely thin surface layer, giving the steel a watery, glazed appearance called *sweating*. This sweating, produced only by an excess acetylene flame, is necessary for the successful application of hard-facing alloy to steel.

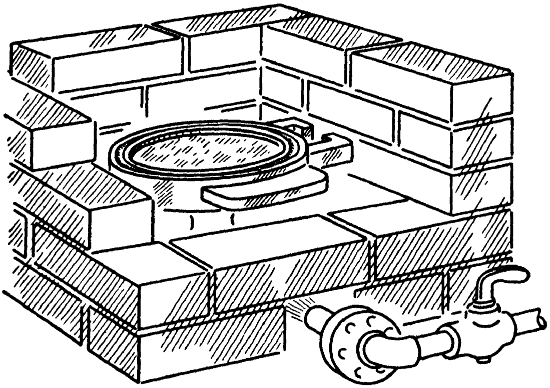


FIG. 15–47.—Build a preheating and cooling furnace to fit the part.

Hold the torch to direct the flame at a 30- to 60-deg angle to the surface; inner-cone tip about 1/8 in. from the steel. Keep this position until the steel under the flame suddenly glazes, indicating that an extremely thin surface layer has melted. The extent of this sweating area varies according to the size of the welding tip; but, for a medium-sized tip, the steel will sweat for a distance of 1/4 in. around the flame. Withdraw the torch slightly, and bring the welding rod between the inner cone of the flame and the hot steel. The tip of inner cone should almost touch the rod, and the rod should lightly touch the sweating area (Fig. 15–49). Melting of the rod forms a puddle on the steel surface. If the first few drops foam or bubble or do not spread evenly, the steel is too cold and should be brought to the recommended temperature.

Some steels foam slightly when brought to sweating heat. When this occurs, stop depositing metal until the foaming stops. If the deposit is started before the foaming is noticed, direct the torch at the foaming spot, and agitate the molten metal with the flame until the foaming stops. Depositing metal during foaming causes blowholes and produces unsatisfactory results.

Fig. 15–48.—A helper can apply the torch to keep the part hot and rotate it during welding.

To spread molten alloy over the area, remove the rod from the flame, and direct the flame into the puddle. Return the rod, and melt off more alloy as required. Now direct the flame so that it plays partly on the edge of the puddle, to keep it molten, and partly on the steel surface adjoining the puddle. As the steel approaches sweating heat, the puddle of hard-facing alloy will spread. As it spreads, bring the rod quickly into the flame again to add more metal as needed. If any dirt or scale appears on the steel or in the puddle, float it to the surface with the flame, or dislodge it with the rod. In extreme cases, use a good cast-iron welding flux.

With a little practice, the right amount of alloy can be added to make the deposit the desired thickness. It is better to do this in one application than to go back over the entire job to add another layer. During the operation, move the flame back to melt a thin surface layer of the deposit, to smooth out high spots as the work progresses. Do this quickly without letting the front edge of the puddle solidify and without interrupting the steady forward travel of the work. After completing the deposit, use the flame to smooth out the remaining rough surfaces. On this second pass, take care to melt only the surface of the hard-facing material without melting down to the base metal. This avoids bringing the iron up from the base metal into the hard-facing deposit.

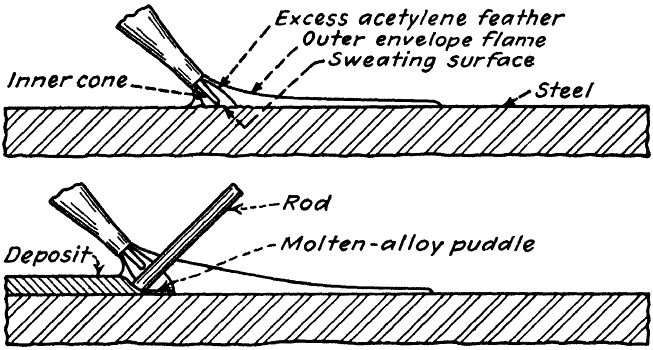


FIG. 15-49.—Hold the torch close to the work to sweat the surface. (*Courtesy of Haynes Stellite Co.*)

When the deposit reaches the desired size and thickness, remove the flame slowly to prevent the formation of shrinkage cracks and blowholes. If these occur, remelt the deposit, and remove particles of scale or oxide from the pool. If the holes still remain, grind the alloy deposit down to the steel, heat the area gradually with the flame, and deposit additional alloy. Make sure that no slag, dirt, or scale is covered or embedded in the deposit to cause pinholes.

Slow cooling is absolutely essential to produce a deposit free from cracks and internal stresses. Parts showing a strong tendency to crack, such as large gate-valve wedges and seat rings, or parts on which the deposit is circular or large in area, should be returned to the preheating furnace while still hot from welding. Bring them slowly to a low red heat, and then let them cool in the furnace. If a furnace is not available, place the part in dry, powdered lime, ashes, or other insulating material, so that at least 2 in. of material covers every part of the object. Uniform cooling is important, because it is during this process that contraction strains are set up.

Some alloy steels used for valve trim require heat-treatment in order to maintain corrosion resistance. When this is necessary, follow the steel manufacturer's instructions with but one exception: never cool hard-faced parts by quenching in water or in an air blast. This will set up strains and cause cracks in the hard-facing. If quenching is necessary, use oil.

After the hard-faced part has cooled, the excess metal must be removed. This can be done by grinding or by machining with a tungsten-carbide tool. When steel and stellite are ground simultaneously, the wheel should be dressed frequently to prevent undercutting of the steel at the edge of the stellite deposit.

TABLE 15–1. GASKET-MATERIAL SELECTION*

Gasket material	Fluid	Usual maximum temperature, F
Red rubber	Steam, air, water	250
Asbestos composition	Steam, water, oil	750
Fiber and paper	Oil	200
Synthetic rubber	Oil	200
Copper, corrugated or plain	Steam or water	600
Steel, corrugated or plain	Steam or water	1000
Stainless steel, 12–14 per cent chromium, corrugated	Steam or water	1000
Hydrogen-annealed furniture iron	Steam or water	1000
Monel, corrugated or plain	Steam or water	1000
Ingot iron, special gasket for ring-type joint	Steam, water, oil	1000

* CROCKER, S., "Piping Handbook," 4th ed., McGraw-Hill Book Company, Inc., New York 1945.

CHAPTER XVI

TRAPS AND CONDENSATE SYSTEMS

A book on automatic control would not be complete without a description of ways and means of removing condensate from a heat-exchange system, because a temperature controller is useless if the condensate is not removed as fast as it forms.

The principal purpose of a steam trap is to remove air and condensate from steam-heated equipment and distribution lines automatically. Obviously, without some means of draining, steam equipment subject to heat loss would eventually fill with condensate. Suppose we used a globe valve for a drain and opened it exactly enough to drain out all water without releasing steam. It would (1) discharge condensate as it accumulates, (2) hold back steam, and (3) vent air (always present in varying amounts).

The principal objection to the globe valve is that it must be readjusted for every change in pressure or condensation rate, or it will hold back the condensate at heavy loads and leak steam at light loads. The automatic steam trap is a mechanical device that discharges the condensate and air yet holds back the steam—all *under reasonably varying pressures and condensate loads.* Steady condensate load is a relative matter, because even the so-called "steady condensing rate" of heating lines varies with steam pressure and warm-up time. Therefore, steam-trap selection resolves itself into determining the heaviest condensate load to be experienced, the pressure under which it occurs, and which trap has sufficient capacity under these conditions.

Steam traps are of two types (Fig. 16–1): (1) Return or pumping traps used to (*a*) return condensate to the boiler against boiler pressure and (*b*) discharge condensate against a static head or pressure greater than the line being drained. (2) Nonreturn, or separating, traps that separate condensate from the steam or other

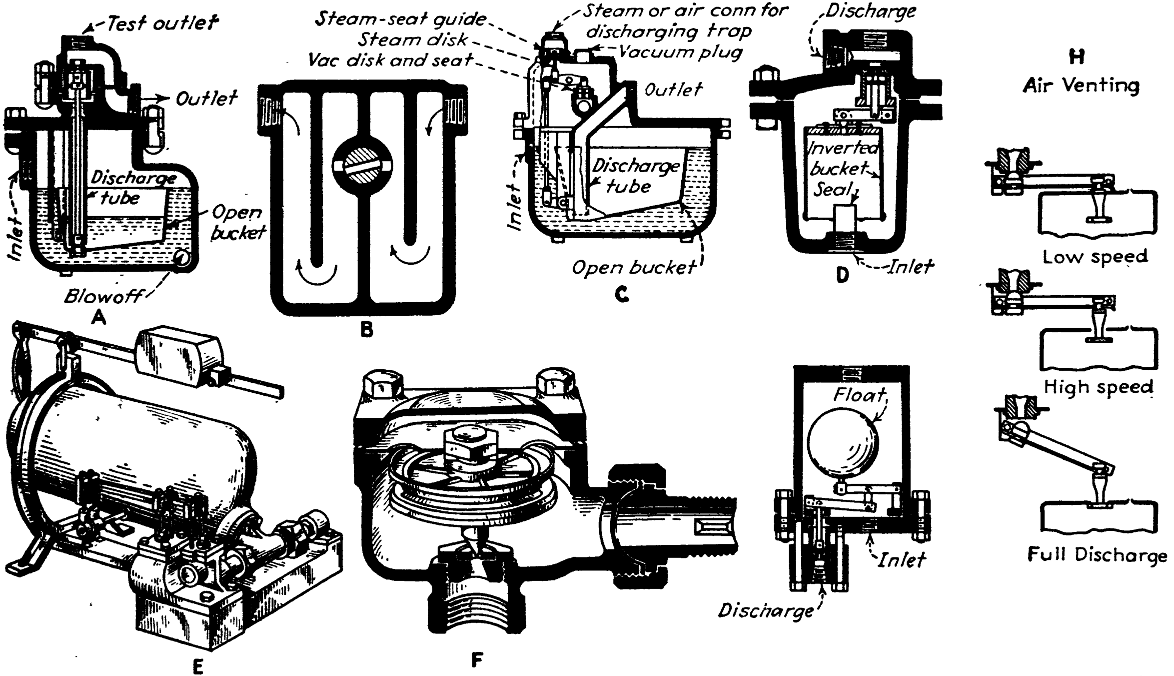


FIG. 16–1.—(*A*) Open bucket; (*B*) adjustable orifice; (*C*) open-bucket return trap; (*D*) inverted bucket; (*E*) tilting; (*F*) thermostatic; (*G*) float trap; (*H*) air vent in inverted bucket.

medium by buoyancy (ball float, open bucket, inverted bucket) tilting, thermostatic, or thermodynamic units.

A *ball-float* trap uses a floating ball that rises and falls with the condensate level to open or close the discharge valve *G*. It is also called a *continuous-flow trap*, because the condensate level within the trap causes the float to accommodate itself to the inflow rate and to discharge condensate continuously. The condensate is at steam temperature. Air must be purged manually.

An *upright*, or open-bucket, trap controls the discharge-valve movement by the weight of water in an open bucket *A*. Condensate gradually fills the trap body and overflows into the bucket to produce an intermittent discharge. When the bucket becomes overbalanced, it drops and pulls the discharge valve open. The valve is closed when sufficient water is discharged from the bucket to make it buoyant again. The small amount remaining seals the discharge tube. Air must be purged manually. The condensate is at steam temperature. Operation is intermittent.

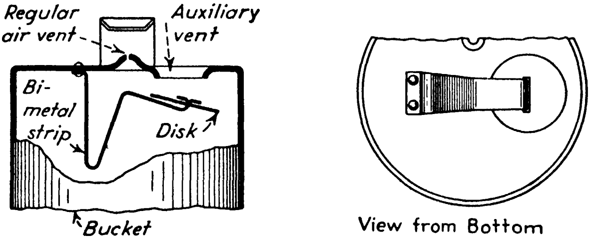


FIG. 16–2.—Bimetallic air vent in an inverted-bucket trap.

An *inverted-bucket* trap operates on the ability of the bucket to float when it is partly filled with air and steam and to sink when water displaces the steam and air, *D*. In sinking, the bucket pulls the discharge valve off its seat, and the trap discharges air and water until the entering steam causes it to become buoyant. Operation is intermittent. The condensate is at steam temperature. This trap should be filled with water when it is installed. Some inverted-bucket traps use a small hole in the bucket for low-speed air venting, and clearance around the bucket hanger for high-speed venting, *H*. Another uses the bimetallic strip (Fig. 16–2).

The inverted-bucket compound trap (Fig. 16–3) is designed to handle large volumes of condensate at any pressure. Steam and condensate reach the bucket through the pilot tube that extends from the receiving chamber down to the pilot body and up under the bucket. The inverted bucket then becomes filled with water and sinks, opening the pilot valve.

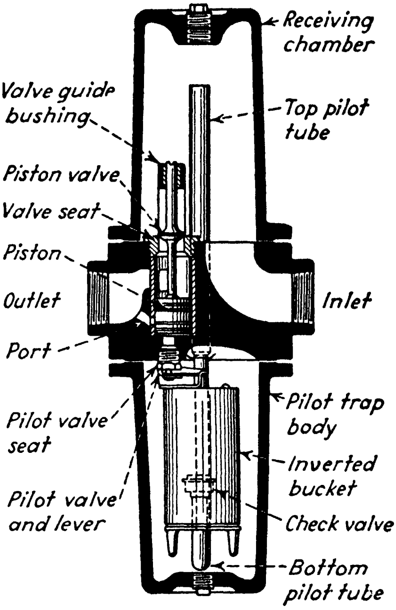


FIG. 16–3.—Inverted-bucket compound trap handles large volumes of condensate. (*Courtesy of Armstrong Machine Works.*)

The pilot valve discharges against the piston, which has an area greater than its connected main valve. Thus the piston is forced up, opening the main valve and letting water escape from the receiving chamber. As the piston moves upward, its bottom skirt uncovers a relief port, and its top skirt begins to close the main-valve discharge port. This results in restraining the piston and main valve against excessive upward movement.

When steam again enters the receiving chamber and flows down through the pilot tube, the bucket floats and closes its pilot valve and therefore the main valve closes.

A *tilting trap* operates an external discharge valve through levers. The entire trap body is counterbalanced; and, when sufficient condensate collects in the chamber, it overcomes the counterweight, and the trap tilts, thereby opening the discharge valve (Fig. 16–1*E*). After the condensate is forced out, the counterweight tilts the body back to the filling position. If the valve mechanism is arranged properly to admit live steam, this trap can be made to return condensate to the boiler. Air must be purged manually unless the tank is vented during the filling operation.

A *thermostatic* trap operates its discharge valve upon temperature changes affecting its sensitive element (Fig. 16–1*F*). The bellows is filled with a suitable liquid that expands or vaporizes when heated. Sometimes the filling agent is a mixture of water and some other liquid having a boiling point less than water. For a given temperature, the mixture exerts a greater vapor pressure than water alone. The temperature at which the condensate discharges depends upon the mixture in the element, but it is generally lower than steam temperature. Discharge is generally intermittent. When the system is first placed in service, air and cold condensate pass through the trap. As warm condensate reaches the element, it expands and starts closing the valve. The unit in Fig. 16–4 uses the expansion of a metal tube to operate the discharge valve.

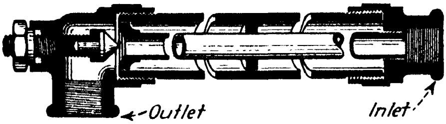


Fig. 16–4.—Trap uses the expansion of a metal tube to operate the valve.

A *thermodynamic* trap controls condensate flow by the principle that condensate temperature will regulate the flow rate through an orifice (Fig. 16–1*B*). The approach of steam into the unit heats the condensate and slows up its passage through the orifice. If air or condensate enters, the resulting lower temperature speeds up discharge through the orifice.

In the device in Fig. 16–5, the flow of condensate divides, the main part going freely out valve *A* and the remainder, called the *control flow,* by-passing continuously up into chamber *B* through the annular orifice around disk *C*. From chamber *B*, the condensate flows out through the control orifice in valve stem *D*. When the system is heating up, the condensate is not at high temperature, and the control flow through the center orifice does not change in volume. Discharge through this orifice lowers the pressure in chamber *B*, and the main valve is opened by system pressure. Thus air and full-flow condensate is discharged.

When the system comes up to heat, the condensate is at near-steam temperature, and the control flow entering chamber *B* flashes into vapor because of the reduced pressure. This flash steam increases the volume of the control flow, and discharge through the center orifice is choked. Therefore, pressure builds up in chamber *B* and closes the main valve to stop the discharge of hot condensate except for the small quantity that flows out the control orifice. The main discharge is intermittent on normal load.

Always base the selection of a separating trap for a particular installation on (1) a combination of initial and maintenance cost, (2) economy with relation to steam loss, and (3) the ability to handle air and condensate load under pressure conditions to be encountered. Trap manufacturers make specific recommendations for their products, but careful study of trap literature is the best way to learn operating characteristics and features of each particular type.

After selecting the type of trap, give careful consideration to capacity, which has little relation to pipe size or physical dimensions. One type of ½-in. trap may discharge 200 lb of condensate per hour at a given pressure, whereas another of the same size discharges 2,500 lb under the same conditions. One trap may weigh 4 lb, another 40 lb, though both have ½-in. connections. Trap capacity is determined by the unrestricted area of its discharge orifice. All reputable manufacturers furnish information on orifice sizes and capacities at various pressures.

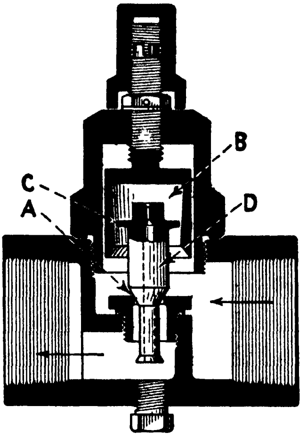


FIG. 16–5.—Operation of the impulse trap depends on pressure acting in chamber *B*. (*Courtesy of Yarnall-Waring Co.*)

Although valve-orifice area is one of the best means of comparing trap capacity, the effect of different internal designs, types of orifices, flow restrictions at the orifice, valve mechanisms, piping arrangements, air pockets, and pressure drop can easily reduce the condensate flow through a trap to a fraction of its theoretical orifice capacity.

Mechanical traps use valves and seats of different area for different pressures, because the valve must come off its seat even though subject to line pressure. The manufacturer size valves, and he must know the prescribed operating pressure. Remember that a trap will operate on differential pressures lower than the seat-orifice rating (with decreased capacity) but cannot open against higher differential pressures. Thus, a trap with a 125 psi orifice may be used on differential pressures up to this value but will not open on higher differentials. Actual pressures are determined by subtracting the discharge pressure (at the trap) from the inlet pressure (at the trap). It is this differential, or operating pressure, that controls the orifice capacity.

All buoyancy traps operate on differential pressures, and it is not important whether the discharge is to vacuum or against a back pressure. When a trap discharges into a vacuum system, differential pressure is *increased* about ½ lb for each inch of vacuum. For example, a trap with 100 psi at the inlet and discharging to a 20-in. vacuum has a differential pressure of 110 psi.

Each foot of *lift* (head) on the trap discharge *adds* about ½ lb to the back pressure. For example, a trap with 100 psi inlet pressure discharging to atmosphere, but with a 10-ft lift or head on the discharge side, has a back pressure at the trap of about 5 psi, and the differential pressure is 95 psi.

Each foot of static head on the trap inlet *adds* ½ lb to the inlet pressure, which means that, if the head on the inlet is 20 ft and the operating pressure 100 psi, the total pressure on the inlet is 110 psi. When traps discharge to atmosphere, the difference between differential pressure and inlet or operating pressure is usually so slight that only operating pressure is considered. When ordering a trap, give the manufacturer (1) inlet pressure, (2) outlet pressure—even if discharging to atmosphere, and (3) amount of condensate to be removed.

The most difficult problem is determining the amount of condensate to be removed. Usually it can only be estimated. Fortunately, most trap manufacturers provide simple rules for determining condensation rates; equipment manufacturers have reliable figures on condensing rates for their machines. Here are a few considerations frequently overlooked in checking condensation rate: (1) Are pressures constant? If not, trap capacity will change as inlet or outlet pressures vary. (2) Is the condensation rate steady? A trap may be large enough for normal operation but far too small for warm-up loads. The condensing rate for 5 min may be many times that allowed when computed on an hourly basis. Traps are rated in pounds per hour; so consider peak load when purchasing a trap. (3) Has consideration been given to entrained air? A trap may be large enough to handle the water but too small or of the wrong design to remove air.

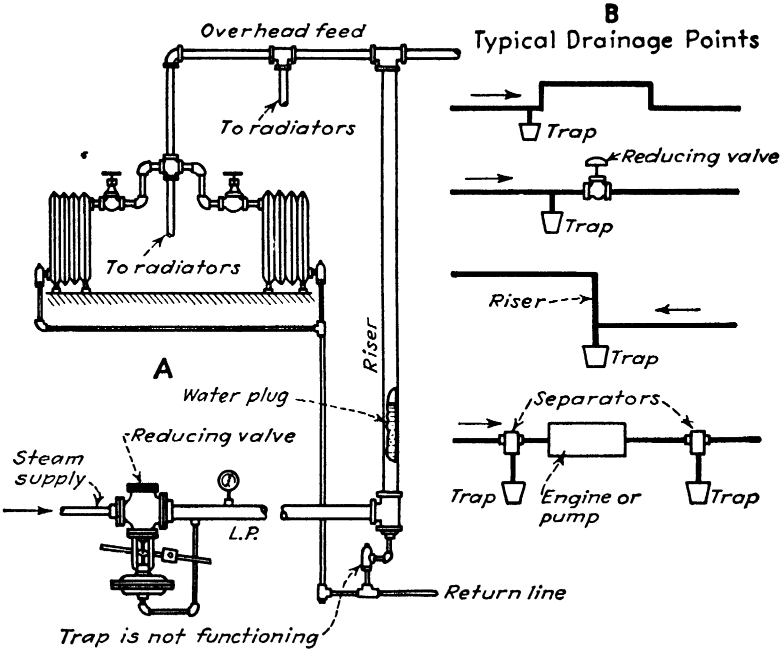


FIG. 16-6.—(*A*) If the riser trap fails, water leg builds up pressure on control-valve diaphragm, and it shuts off the steam. (*B*) Typical drainage points.

Certain traps, including the open and inverted-bucket units, discharge intermittently. This is desirable for purging lines of air and improving circulation. However, certain installations—time-temperature processes particularly—require continuous condensate flow through the trap to avoid instrument lag. Here a continuous-flow trap serves best.

To provide correct intermittent discharge and yet operate with minimum maintenance, a trap should have a capacity at least three times its normal condensate flow. At peak loads, intermittent-discharge traps may operate continuously (and they will if condensate flow is equal to or greater than trap capacity). Peak load must therefore be considered so that traps are large enough to handle the starting-up flow and at the same time be able to operate intermittently when condensate flow returns to normal.

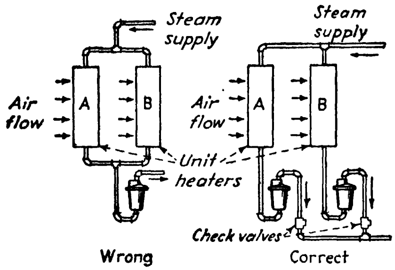


FIG. 16–7.—Each heating unit or separate parallel sections on one unit must have its own trap.

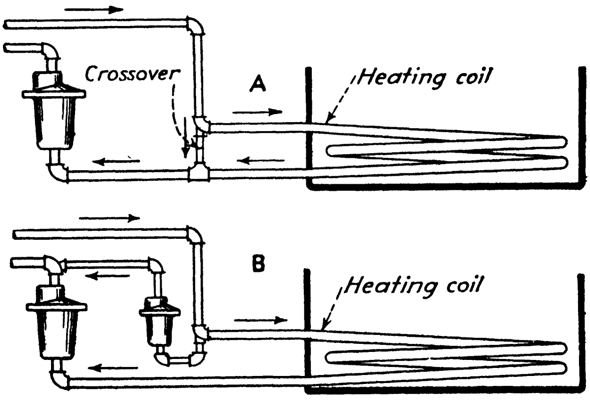


FIG. 16–8.—Never short-circuit a heating coil with a cross-over to drain the header (*A*). Use two traps (*B*).

For best results, install traps at low points (risers); at ends of mains; at intervals on long runs of pipe, ahead of equipment requiring dry steam, such as reducing valves and process units; and as a drain on each piece of steam equipment (Fig. 16–6). Install them singly, *i.e.*, one trap to a unit (Fig. 16–7), because no two pieces of equipment condense steam at exactly the same rate. Where one trap drains several units, air and condensate are likely to accumulate in one machine. This short-circuiting results in lower efficiency, longer heating time, uneven rate of heat, and increased rate of radiation loss. This means increased steam consumption and the higher labor cost of production slow-up. Never short-circuit a heating coil with a cross-over to drain a header. Use a separate trap (Fig. 16–8).

Never discharge condensate from high-pressure traps directly to the sewer, even though the condensate is not to return to the boiler. Such practice wastes fuel. When discharging to a lower pressure, a certain percentage of the condensate flashes into steam. Flash steam from medium and high pressures can be recovered by passing the discharged condensate through a heat exchanger, water heater, or feed-water preheater or into a low-pressure heating main. The amount of reflash is easily computed. For example, when 125 psi condensate is discharged into a 15 psi system, 11.2 per cent of the condensate flashes into steam.

TABLE 16–1.—USING REFLASH STEAM SAVES MONEY

High-pressure condensate, lb per hr	Reflash steam 11.2%, lb	Saving		
		Per hr	Per day	Per year
1,000	112	$0.04	$1.08	$ 388.80
2,000	224	0.09	2.16	777.60
4,000	448	0.18	4.32	1,555.20
5,000	560	0.22	5.40	1,944.00
10,000	1120	0.45	$10.80	3,888.00

Savings are based on a steam cost of 40 cents per 1,000 lb.

$$\begin{array}{lll} 125 \text{ psi } (353 \text{ F}) & = 324 \text{ Btu} & \text{(heat in liquid)} \\ 15 \text{ psi } (250 \text{ F}) & = 218 \text{ Btu} & \text{(heat in liquid)} \\ \hline \text{Difference} & \quad 106 \text{ Btu} & \end{array}$$

Latent heat at 15 psi = 945 Btu per lb

$$\frac{106}{945} = 0.112 \text{ or } 11.2 \text{ per cent}$$

This means that, if steam costs 40 cents per 1,000 lb, using the reflash from 5,000 lb of condensate an hour saves more than $1,944 a year (Table 16–1).

Follow these pointers when installing traps:

1. Connect them close to and preferably below the unit they drain so that the condensate reaches the trap by gravity (Fig. 16–9).

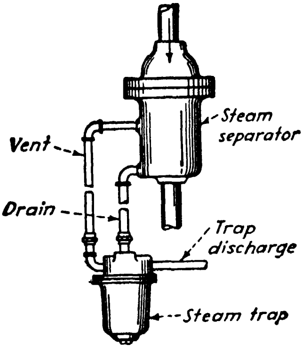


FIG. 16–9.—Vent the trap back to the object being drained to prevent air and steam binding.

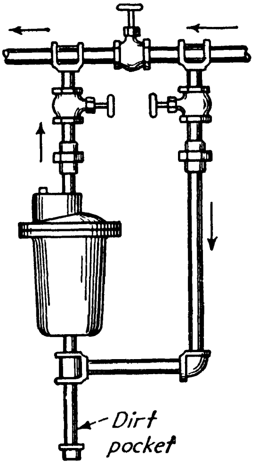


FIG. 16–10.—A dirt pocket is better than no protection if it is cleaned often, but it is a poor substitute for a strainer.

2. Make them accessible for easy maintenance.
3. Install them in an upright position.
4. Connect inlet and outlet correctly.
5. Use shutoff valves and unions on both inlet and outlet to make removal for maintenance easy (Fig. 16–10).
6. Good engineering practice requires strainers ahead of all traps. This is particularly true of high-pressure units.
7. Second best practice is to install a dirt pocket below the trap (Fig. 16–10).
8. Blow out the line before hooking up a trap.
9. Avoid unnecessary U bends; they obstruct free flow and cause steam binding (Fig. 16–11). A slug of water flowing toward the trap immediately

after discharge lies in the pocket until steam, remaining beyond the pocket and in the trap chamber, condenses.

10. Bucket traps must have water in the body at all times and should be primed when installed, because there may not be enough accumulated condensate in the system to float the bucket (catch the prime). If insufficient water reaches the trap to prime it, the bucket does not become buoyant, and the trap blows steam.

11. A trap operates when placed above the system being drained, provided that it has sufficient capacity at the differential pressure available and that there is enough pressure to lift the condensate to the trap. When locating a bottom inlet, or inverted-bucket trap above the unit being drained, install a check valve on the inlet line (close to the trap) to prevent backflow and loss of water seal (Fig. 16–12). A water seal drains the lower coil. A small pipe length screwed into the body of a bottom-inlet trap (about 2 in.) provides an effective seal (Fig. 16–1*D*).

12. Where traps discharge into a common line, install a check valve in each trap discharge (Fig. 16–7) to prevent backflow from other units or drainage back to an idle unit. This is especially troublesome with overhead return lines.

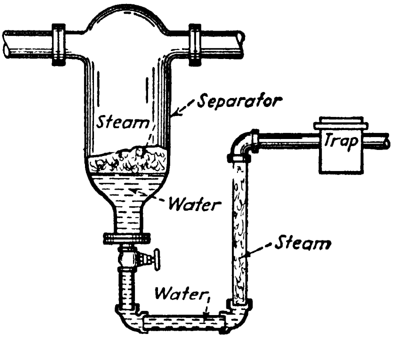


Fig. 16–11.—Unnecessary bends and long lines between trap and vessel being drained cause steam binding.

13. When the condensate flows vertically downward to a trap, a heavy rush of water may choke the line and prevent the backward escape of trapped steam. Thus the steam must condense before water can enter the trap. For conditions such as this, and especially with air, vent the trap chamber back to the line (Fig. 16–9). Naturally, this will work on air only when the trap is lower than the equipment being drained.

Here are the more common trap troubles:

1. If it takes too long to bring steam-heated equipment to the desired temperature, the trap may be too small or have insufficient air-handling capacity. Check this by opening the by-pass or blowoff to see whether the condensate has backed up in the machine.

2. Check the pressure on heating and process units at regular intervals. Too high pressures may damage thermostatic traps. Mechanical ones may not be able to open against higher differential pressure. If the steam pressure is too low, the unit does not get enough steam, and the trap capacity may be inadequate at the new differential pressure.

3. On a manually controlled drain valve or on traps with manually adjusted orifices, make frequent checks to determine whether the setting is such that no steam escapes but all condensate is removed. Such traps usually carry a gage glass or indicator to show condensate level.

4. When discharging to a vacuum system, see that all parts of thermostatic or bucket traps, including the union and cover, are tight against air leakage.

5. If a bucket trap fails to discharge, be certain that the condensate actually reaches it. The trap may be clogged with dirt. Be sure that the differential pressure (inlet minus outlet) is the same or less than valve-and-seat rating stamped on the body.

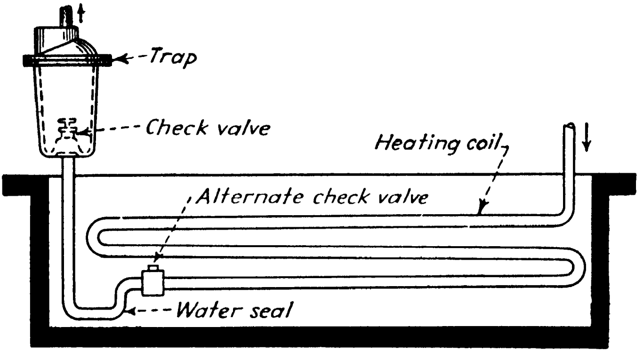


FIG. 16–12.—When a trap is above the heating coil, install suitable check valves to prevent reverse flow.

6. When a bucket trap discharges continuously, it may be too small or clogged with dirt; or perhaps back pressure on the outlet reduces the differential pressure so much that the trap capacity is inadequate. For example, a trap orifice may be rated at 50 psi for use on 50 psi inlet pressure with discharge to atmosphere. Should the back pressure build up to 20 or 30 psi, the differential pressure would be so reduced that the trap capacity would be negligible. Therefore, it would naturally discharge continuously.

7. Frequent trouble occurs when traps are purchased for certain operating pressures and later installed on higher pressure systems; or sometimes the system pressure may be raised. Nearly all buoyancy traps have valves and seats that can be removed and replaced with ones of correct area for the new pressure.

8. Steam lines frequently contain dirt, scale, packing, or gasket material that finds its way to the trap and lodges between the valve and seat. Use a strainer to prevent this.

9. When traps drain units of different pressures and discharge into a common return line, the common back pressure may be considerably higher than the inlet pressure on a trap of low inlet pressure. Of course, steam and condensate back up through this low-pressure trap and flood the heating unit.

Equip all traps except small radiator ones with valves on the discharge side for periodic testing for leaks. If there is no test outlet on the trap, insert a tee and plug in the discharge line. When the back pressure in the return is also high, insert a check or shutoff valve in the discharge line to prevent flash steam from passing back through the test valve.

If traps are equipped with a by-pass, be sure that the valve is kept tightly closed. An open or even a leaking valve can waste large amounts of steam. Install without a by-pass if feasible. When the condensate returning to the hot well is abnormally hot, one or more traps may be leaking. Trace out the return lines until the hot branch is found, and follow it up to the leaking trap.

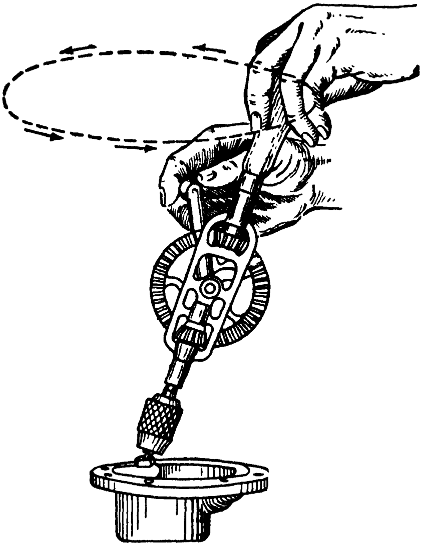


FIG. 16–13.—When grinding ball disks, turn the crank while moving hand drill along dotted line. (*Courtesy of Armstrong Machine Works.*)

Trap maintenance usually involves cleaning to remove foreign matter that might interfere with valve or linkage action, reseating valves when necessary, removing lost motion from linkage, and renewing body gaskets. Normal wear of some trap valves gradually enlarges the seat area at the point of disk contact. This increases the surface acted on by differential pressure across the valve, making it more difficult to operate. The area may increase to such an extent that the operating device is no longer powerful enough to move the valve. When this happens, it must be refaced to regain its original area. When grinding ball-disk valves, turn the crank while moving the hand drill along the dotted line shown in Fig. 16–13.

Whenever a trap fails to operate and the reason is not readily apparent, observe the discharge by opening a test cock or breaking the outlet connection. Live steam usually indicates a leaking valve; it may be caused by the trap's losing prime. Failure to discharge can be caused by a leaking by-pass valve, inlet piping or a trap obstructed by sediment and scale, a return line that is too small, or an obstructed outlet.

Valve leakage represents a common cause of trouble; worn seats are not so much to blame as are foreign particles that prevent the valve from seating. Other trap troubles include rusting and sticking of the mechanism, lost motion in the linkage, gasket and connection leaks, float leaks, and bent lever arms. Wear in levers and pins of a continuous-discharge trap cause it to operate intermittently. When a trap valve does not seat properly yet appears to be in good condition, check the linkage and stem length. Repeated regrinding may have shortened the stem. Adjust the stem length when it is at full operating temperature.

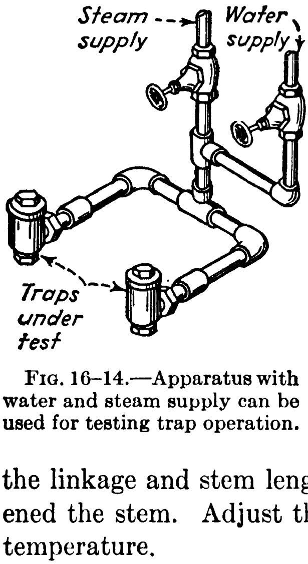
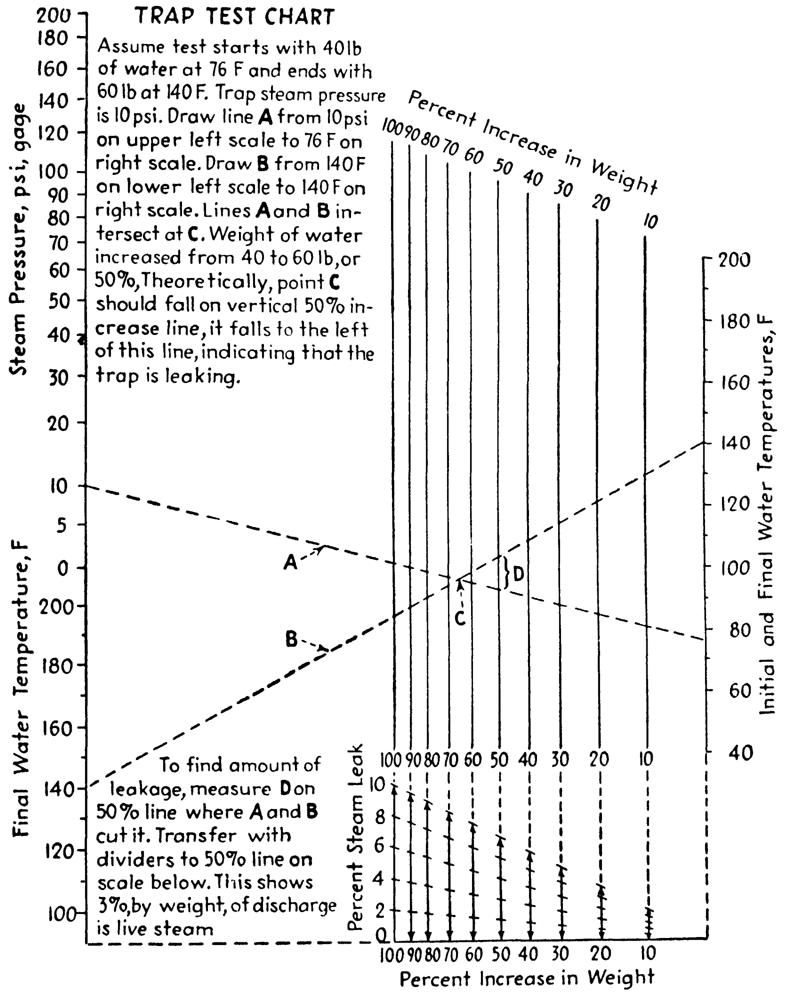


FIG. 16–14.—Apparatus with water and steam supply can be used for testing trap operation.

There are numerous ways of checking trap operation. The test arrangement in Fig. 16–14 can be used to check any type of trap. In service, a slight temperature difference between the inlet and outlet indicates a working trap, provided that the inlet temperature is up to normal. No difference in temperature indicates a leaking trap; a large difference or a cold trap indicates that no condensate is passing. Intermittent-discharge traps produce a light clicking sound at each operation; constant-discharge ones can be checked with a listening rod or stethoscope applied to the trap body.

When visible discharge is not satisfactory evidence, passing the trap discharge into a vessel of water forms a positive test. Weigh



Assume test starts with 40 lb of water at 76 F and ends with 60 lb at 140 F. Trap steam pressure is 10 psi. Draw line **A** from 10 psi on upper left scale to 76 F on right scale. Draw **B** from 140 F on lower left scale to 140 F on right scale. Lines **A** and **B** intersect at **C**. Weight of water increased from 40 to 60 lb, or 50%. Theoretically, point **C** should fall on vertical 50% increase line, it falls to the left of this line, indicating that the trap is leaking.

To find amount of leakage, measure **D** on 50% line where **A** and **B** cut it. Transfer with dividers to 50% line on scale below. This shows 3%, by weight, of discharge is live steam

FIG. 16–15.—Chart simplifies computations of trap steam leakages, eliminating the use of steam tables. (*Courtesy of Power.*)

the original quantity of water, and measure its temperature. After discharge, weigh the water, and measure its temperature. The heat given up by trap discharge in falling to the final temperature equals the heat gained by the original water quantity rising to the same temperature. Figure 16–15 simplifies the computations, and directions for its use are contained in the chart.

Inspect, clean, and repair traps at least yearly. The valve and seat on intermittent-discharge traps should have a bright shiny mark all around the contact area. Inspect seats for wire drawing. Always replace the valve and seat in sets. Replace leaking ball floats and cracked buckets, and worn or broken parts in thermostatic traps. Bellows, diaphragms, or thermal strips may be checked for operation by submerging them in hot water.

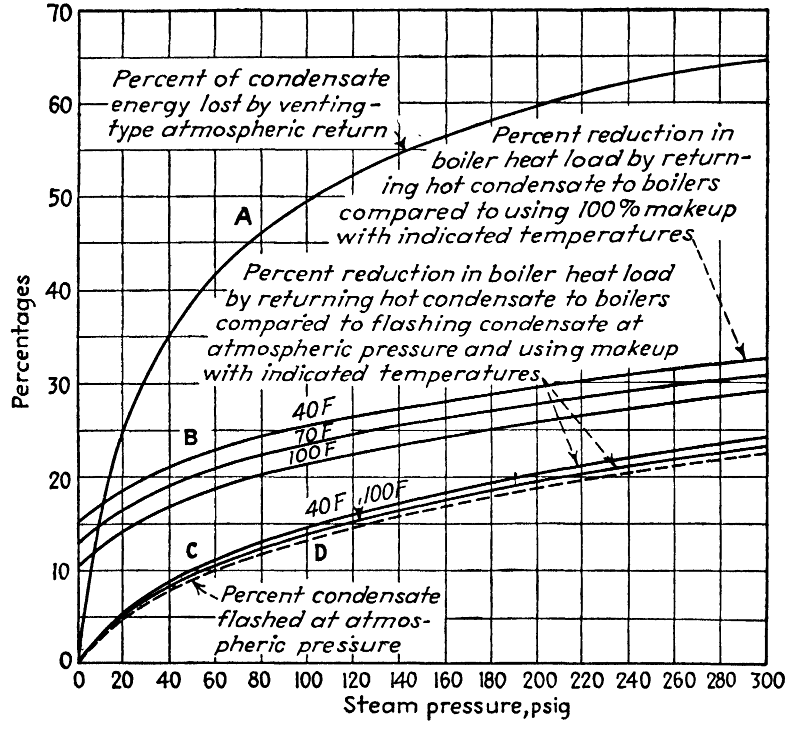


FIG. 16–16.—Returning hot condensate as feed water reduces the boiler load. (*Courtesy of David Stout & Sons.*)

What becomes of the condensate after leaving the trap is important from the standpoint of the remaining heat it contains and of whether it can be used again for feed water. If the condensate must be discharged to waste, economy demands that it be run through a heat exchanger so that its heat serves a useful purpose. On the other hand, if the condensate can be used in the boiler, fuel will be saved if the condensate is returned without flash loss to the atmosphere. Besides saving valuable heat units, hot-condensate return markedly reduces the boiler load and releases capacity for additional uses, especially at higher steam pressures (Fig. 16–16).

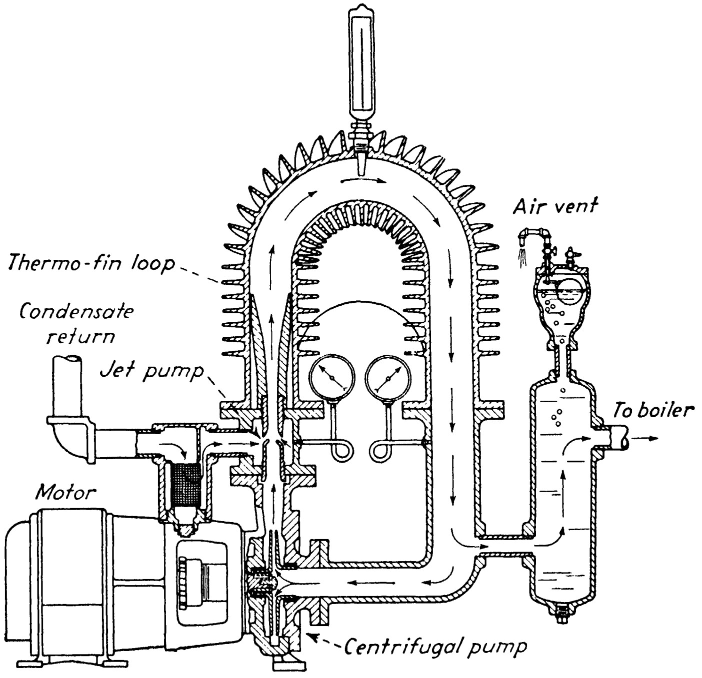


FIG. 16–17.—Apparatus that returns condensate to the boiler under pressure, thereby preventing flash loss. (*Courtesy of Cochrane Corp.*)

Curve *D* shows the proportion of condensate flashed to steam when throttled to atmospheric pressure. If the flashed steam cannot be utilized, it is wasted, and the proportion of energy lost is indicated by curve *A*. The practical effect of such throttling and waste is given by curves *C*, showing the increase in boiler load caused by replacing the lost flashed steam with make-up at temperatures of 40 F and 100 F and assuming the remaining condensate is returned at 212 F.

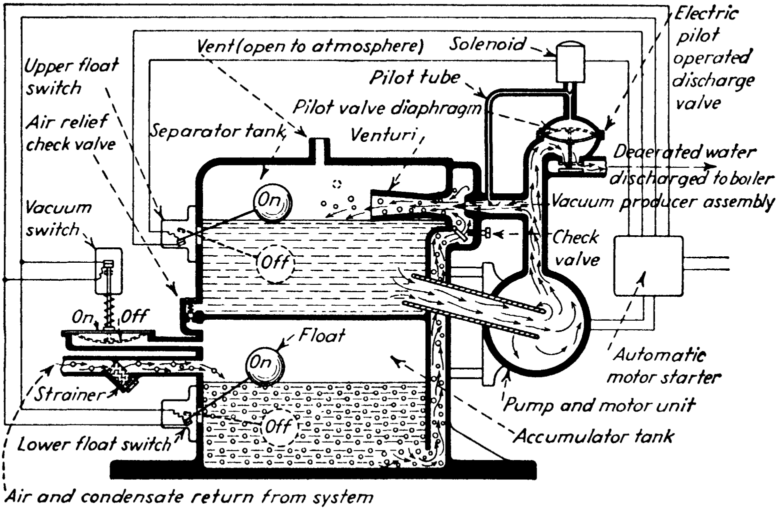


FIG. 16–18.—Apparatus that maintains a partial vacuum on condensate lines but vents condensate to the atmosphere. (*Courtesy of Sterling, Inc.*)

The curve family *B* compares the reduction in boiler load effected by returning the hot condensate under pressure with the condition of wasting all the condensate and replacing it with raw make-up to the boiler at inlet temperatures of 40, 70, and 100 F.

A device that gets the condensate back to the boiler under pressure is shown in Fig. 16–17. Here a centrifugal pump draws water from the thermofin priming loop and discharges it as a high-velocity jet through the jet-pump nozzle. This jet induces condensate flow from the incoming line through the venturi-shaped mixing tube and into the thermofin priming loop. Introducing this additional volume of water into the constantly filled loop results in the discharge of a like volume through the air separator to the boiler.

Hot condensate and air are drawn from the processing equipment by the jet pump. Air is eliminated under pressure at the separator, and condensate goes to the boiler at maximum pressure and temperature. Thus the circuit from the boiler to the process equipment and back to the boiler is completed in a closed system without flash loss or substantial drop in temperature.

A conventional vacuum pump can return condensate to the boiler, but considerable flash loss will occur unless heat is extracted from the condensate before it reaches the pump chamber. In operation, the condensate enters the lower chamber (Fig. 16–18). A centrifugal pump taking suction from the upper chamber forces a jet of water through the venturi nozzle. The jet induces water flow from the lower to the upper chamber, where the entrained air is liberated to pass out the vent. The diaphragm valve on the pump discharge operates to feed the boiler as needed.

APPENDIX

APPENDIX

AUTOMATIC-CONTROL TERMS[1]

The need for a consistent and usable set of terms applying to industrial-process control has been evident for a number of years. Suppliers and users alike frequently find it difficult to discuss the subject without misunderstandings that result from the lack of a common language. Further, in the writing of specifications for new equipment and of directions for its use, it is desirable to have at hand a standard set of terms. The list of automatic-control terms and definitions which follows was prepared by the Terminology Committee of the Industrial Instruments and Regulators Division of the American Society of Mechanical Engineers. An attempt has been made to adhere to best current usage, with such modifications as may be required to make the list consistent and comprehensive. In its present form, the list embodies revisions based on comments and suggestions that were received in response to the publication of a preliminary list.[2]

This list covers most of the terms commonly employed in the field of automatic control for industrial processes. The definitions are directed primarily to this field but in many cases are sufficiently broad to have wider application. The order in which the terms appear is merely one of convenience. For a clear and comprehensive understanding of any particular term, it will often be necessary to consider associated terms elsewhere in the list. In all cases, the definitions apply to mechanisms and elements that perform their designed functions perfectly.

Automatic controllers are here considered to be only those devices which are constructed and capable of being used in such a way that the complete installation, including the process under control, constitutes a closed loop of action and counteraction operating without human aid.

Some phenomenon of the process, called the *controlled variable,* produces an initial effect upon the measuring means of the controller. If this effect does not correspond with the value of the controlled variable that the automatic controller operates to maintain, the measuring means institutes corrective action. The corrective action restores, or tends to restore, the value of the controlled variable to the desired value.

Unless this entire loop of initial action and corrective action is present, automatic control is not realized. Anything less than this constitutes some form of automatic operation, rather than automatic control.

---

[1] Reprinted from *Mechanical Engineering* for February, 1946.

[2] Process Control Terms, *Mechanical Engineering*, vol. 66, pp. 205–206, 1944.

CLASSIFIED LIST OF AUTOMATIC-CONTROL TERMS AND DEFINITIONS

100. AUTOMATIC CONTROLLERS AND CONTROL SYSTEMS

101. An *automatic controller* is a mechanism that measures the value of a variable quantity or condition and operates to correct or limit deviation of this measured value from a selected reference. It includes both the measuring means and the controlling means. (*Automatic regulator* is a synonymous term.)

102. A *self-operated controller* is one in which all the energy necessary to operate the final control element is derived from the controlled medium through the primary element.

This type of automatic controller must have both self-operated measuring means and self-operated controlling means.

103. A *relay-operated controller* is one in which the energy transmitted through the primary element is either supplemented or amplified for operating the final control element by employing energy from another source.

This type of automatic controller may have either a self-operated measuring means and a relay-operated controlling means, or a relay-operated measuring means and a self-operated controlling means, or a relay-operated measuring means and a relay-operated controlling means.

104. The *desired value* is the value of the controlled variable which it is desired to maintain.

105. The *set point* is the position to which the control-point-setting mechanism is set.

Where the automatic controller possesses a set-point scale the set point is the scale reading translated into units of the controlled variable. Where a setting scale is not provided the set point is the position of the control-point-setting mechanism translated into units of the controlled variable.

In some types of automatic controllers, *e.g.*, those with a two-position differential gap, floating with neutral or proportional-position action, the set point is related to the position of a range of values of the controlled variable. The set point is then generally selected as the center of this range of values.

The set point may be varied manually or by automatic means, such as in time-schedule or ratio control.

106. The *control point* is the value of the controlled variable that, at any instant, the automatic controller operates to maintain.

In some types of automatic controllers, *e.g.*, those with a two-position differential gap or floating with neutral controller action, the control point becomes a control range of values of the controlled variable rather than a single value.

In positioning-type controller action, the control point may lie anywhere within a predetermined range of values of the controlled variable. The control point may then differ from the set point by the amount of offset.

In floating with zero neutral zone controller action, the control point and the set point coincide.

200. BASIC CHARACTERISTICS

Delaying or retarding effects associated with industrial process control are caused by capacitance, resistance, and dead time (either separately or in combination) and have often been designated as various forms of "lag." These three terms cover the basic concepts involved and, in the interest of clarity, should be used in place of the less exact term "lag."

201. *Capacity* is a measure of the maximum quantity of energy or material that can be stored. It is measured in units of quantity.

The volume capacity of an open tank, for example, is the maximum volume of liquid it will hold without overflowing. The weight capacity of a compressed-air tank is the maximum weight of air it will hold without exceeding safe pressure.

202. *Capacitance* is the change in quantity contained per unit of change in some reference variable. It is measured in units of quantity, divided by the reference variable.

The energy or material being contained and the reference variable determine the type of capacitance. Process capacitance may involve different quantities and reference variables, and several types may exist together in one process.

The volume capacitance of an open tank with respect to head is the change of volume of stored liquid per unit change of head, which is equivalent in value to the area of the liquid surface. It should be noted that, if the shape of the tank causes the liquid-surface area to vary with change of head, the capacitance will likewise vary with head.

The weight capacitance of a gas-filled tank with respect to pressure is the change of weight of stored gas per unit change of pressure.

203. *Resistance* is opposition to flow. It is measured in units of potential change required to produce unit change in flow.

204. *Dead time* is any definite delay period between two related actions. It is measured in units of time.

300. PROCESSES, THEIR ELEMENTS AND CHARACTERISTICS

301. A *process* comprises the collective functions performed in and by the equipment in which a variable is to be controlled.

"Equipment," as embodied in this definition, should be understood not to include any automatic-control equipment.

302. *Self-regulation* is a sustained reaction inherent in the process that assists or opposes the estabishment of equilibrium.

303. The *controlled variable* is that quantity or condition which is measured and controlled.

The controlled variable is a condition or characteristic of the controlled medium. For example, where temperature of water in a tank is automatically controlled, the controlled variable is temperature, and the controlled medium is water,

304. The *controlled medium* is that process, energy, or material in which a variable is controlled. See the example for 303.

305. The *manipulated variable* is that quantity or condition which is varied by the automatic controller so as to affect the value of the controlled variable.

The manipulated variable is a condition or characteristic of the control agent. For example, where a final control element changes the rate of fuel-gas flow to a burner, the manipulated variable is rate of flow, and the control agent is fuel gas.

306. The *control agent* is that process, energy, or material of which the manipulated variable is a condition or characteristic. See the example for 305.

400. CHARACTERISTICS OF AUTOMATIC CONTROL

401. *Error* is the difference between the instantaneous value and the desired value of the controlled variable.

402. *Deviation* is the difference between the instantaneous value of the controlled variable and the value of the controlled variable corresponding with the set point.

403. *Offset* is a sustained deviation due to an inherent characteristic of positioning-controller action and is the difference existing at any time between the control point and the value of the controlled variable corresponding with the set point.

404. *Corrective action* is predetermined variation of the manipulated variable initiated by a deviation.

405. *Cycling* is a periodic change of the controlled variable from one value to another. ("Oscillation" is a synonymous term.)

There are three types of cycling, *i.e.*, cycling in which the amplitude gradually decreases, cycling in which the amplitude is constant, and cycling in which the amplitude gradually increases.

500. TYPES OF AUTOMATIC-CONTROLLER ACTION

For simplicity, the definitions that follow are stated in terms relating controller action to "position of a final control element." However, these definitions apply equally to equivalent controller action related to (1) "value of the manipulated variable," and (2) value of the set point of another controller."

For types of automatic-controller actions, which are defined as having a linear relation between a function of the controlled variable and the position or rate of motion of the final control element, it is assumed that the linear relation may be referred to either "motion or force of the last element in the measuring means," as well as to "value of the controlled variable."

It is assumed that the automatic controller operates ideally, *i.e.*, it is capable of detecting infinitesimal variations of the controlled variable and responds instantaneously in accordance with its predetermined action.

501. *Positioning action* is that in which there is a predetermined relation between the value of the controlled variable and the position of a final control element.

501*a*. *Two-position action* is that in which a final control element is moved from one of two fixed positions to the other. ("Open and shut action" and "on-off action" are synonymous terms.)

501*aa*. *Two-position differential-gap action* is that in which a final control element is moved from one of two fixed positions to the other when the controlled variable reaches a predetermined value from one direction, and subsequently is moved to the other position only after the variable has passed in the opposite direction through a range of values to a second predetermined value.

501*ab*. *Two-position single-point action* is that in which a final control element is moved from one of two fixed positions to the other at a single value of the controlled variable.

The differential gap of this type of two-position action is zero. Such controller action may also be considered as proportional-position action in which the proportional band is zero, or floating action with zero neutral zone and with infinite floating speed.

501*b*. *Multiposition action* is that in which a final control element is moved to one of three or more predetermined positions, each corresponding to a definite range of values of the controlled variable.

501*c*. *Proportional-position action* is that in which there is a continuous linear relation between the value of the controlled variable and the position of a final control element.

501*d*. *Average-position action* is that in which there is a predetermined relation between the value of the controlled variable and the time-average position of a final control element that is moved periodically from one of two fixed positions to the other.

This controller action is similar to two-position action in which the percentage "time on" of the final control element is dependent upon the value of the controlled variable. The percentage "time on" may have either a fixed or an infinite number of values to correspond to any one of the other positioning-controller actions defined previously.

502. *Integral action* is that in which there is a predetermined relation between an integral function of the controlled variable and the position of a final control element.

502*a*. *Floating action* is that in which there is a predetermined relation between the value of the controlled variable and the rate of motion of a final control element.

A neutral zone, in which no motion of the final control element occurs, is often employed in floating-controller action.

502*aa*. *Single-speed floating action* is that in which a final control element is moved at a single rate.

502*ab*. *Multispeed floating action* is that in which a final control element is moved at two or more rates, each corresponding to a definite range of values of the controlled variable.

502*ac*. *Proportional-speed floating action* is that in which there is a continuous linear relation between the value of the controlled variable and the rate of motion of a final control element.

502*ad*. *Floating average-position action* is that in which there is a predetermined relation between the value of the controlled variable and the rate of change of the time-average position of a final control element that is moved periodically from one of two fixed positions to the other.

This controller action is similar to two-position action in which the percentage "time on" of the final control element is gradually changed at a rate dependent upon the value of the controlled variable. The rate of change of the percentage "time on" may have either a fixed or an infinite number of values to correspond to any one of the other floating controller actions defined previously.

503. *Derivative action* is that in which there is a predetermined relation between a derivative function of the controlled variable and the position of a final control element.

503*a*. *Rate action* is that in which there is a continuous linear relation between rate of change of the controlled variable and the position of a final control element.

This controller action maintains a linear relation between the first derivative or rate of change of the controlled variable and the position of a final control element. This identical controller action may also be considered as maintaining a linear relation between the second derivative or rate of the rate of change of the controlled variable and the rate of motion of the final control element.

504. *Multiple action* is that in which two or more controller actions are combined.

504*a*. *Proportional plus floating action* is that in which proportional-position action and floating action are combined.

504*aa*. *Proportional plus reset action* is that in which proportional-position action and proportional-speed floating action are combined.

504*b*. *Proportional plus derivative action* is that in which proportional-position action and derivative action are combined.

504*c*. *Proportional plus floating plus derivative action* is that in which proportional-position action, proportional-speed floating action and derivative action are combined.

504ca. *Proportional plus reset plus rate action* is that in which proportional-position action, proportional-speed floating action, and rate action are combined.

600. ADJUSTMENTS OF AUTOMATIC-CONTROLLER ACTION

601. *Neutral zone* is a predetermined range of values of the controlled variable in which no corrective action occurs.

Neutral zone is commonly expressed in percentage of controller-scale range. A neutral zone is employed in some types of floating-controller action.

602. *Differential gap,* applying to two-position controller action, is the smallest range of values through which the controlled variable must pass in order to move the final control element in succession to both of its fixed positions.

Differential gap is commonly expressed in percentage of controller-scale range.

603. *Proportional band,* applying to proportional-position controller action, is the range of values of the controlled variable that corresponds to the full operating range of the final control element.

Proportional band is commonly expressed in percentage of controller-scale range or, particularly in the absence of a controller scale, in units of the controlled variable.

604. *Floating speed,* applying to single or multispeed floating-controller action, is the rate of motion of the final control element.

Floating speed is commonly expressed in percentage of full-range motion per minute.

605. *Floating rate,* applying to proportional-speed floating-controller action, is the rate of motion of the final control element corresponding to a specified deviation.

Floating rate is commonly expressed in percentage of full-range motion per minute per percent deviation.

606. *Reset rate,* applying to proportional plus reset-controller action and proportional plus reset plus rate-controller action, is the number of times per minute that the effect of the proportional-position action upon the final control element is repeated by the proportional-speed floating action.

Reset rate is commonly expressed as a number of "repeats" per minute. It is determined by dividing (1) the travel of the final control element in 1 min due to the effect of proportional-speed floating action by (2) the travel due to the effect of proportional-position action, with the same deviation in both cases.

In automatic controllers having proportional plus reset action and a reset-rate adjustment, the proportional-band adjustment simultaneously affects the proportional-speed floating action in such a manner that the reset rate remains substantially constant at its set value.

Similarly, in automatic controllers having proportional plus reset plus rate action and a reset-rate adjustment, the proportional band may be adjusted without affecting the set value of the reset rate.

607. *Rate time,* applying to proportional plus rate-controller action and proportional plus reset plus rate-controller action, is the time interval by which the rate action advances the effect of the proportional-position action upon the final control element.

Rate time is commonly expressed in minutes. It is determined by subtracting (1) the time required for a selected motion of the final control element, due to the combined effect of proportional-position plus rate actions, from (2) the time required for the same motion due to the effect of proportional-position action alone, with the same rate of change of the controlled variable in both cases.

In automatic controllers having proportional plus rate action and a rate-time adjustment, the proportional-band adjustment simultaneously affects the rate action in such a manner that the rate time remains substantially constant at its set value.

Similarly, in automatic controllers having proportional plus reset plus rate action and a rate-time adjustment, the proportional band may be adjusted without affecting the set value of the rate time.

700. ELEMENTS AND CHARACTERISTICS OF AUTOMATIC CONTROLLERS

701. The *measuring means* consists of those elements of an automatic controller which are involved in ascertaining and communicating to the controlling means the value of the controlled variable, the error, or the deviation.

701*a*. The *primary element* is that portion of the measuring means which first either utilizes or transforms energy from the controlled medium to produce an effect in response to change in the value of the controlled variable. The effect produced by the primary element may be a change of pressure, force, position, electrical potential, or resistance.

701*b*. A *self-operated measuring means* is one in which all the energy necessary to actuate the controlling means of an automatic controller is derived from the controlled medium through the primary element.

701*c*. A *relay-operated measuring means* is one in which the energy transmitted through the primary element is either supplemented or amplified for actuating the controlling means of an automatic controller by employing additional energy.

702. The *controlling means* consists of those elements of an automatic controller which are involved in producing a corrective action.

702*a*. A *power unit* is a portion of the controlling means that applies power for operating the final control element.

702*b*. The *final control element* is that portion of the controlling means which directly changes the value of the manipulated variable.

702*c*. A *self-operated controlling means* is one in which all the energy necessary to operate the final control element is derived from the measuring means.

702*d*. A *relay-operated controlling means* is one in which the energy transmitted from the measuring means is either supplemented or amplified for operating the final control element by employing additional energy.

INDEX OF NONSTANDARD TERMS

Recognizing that many terms now in use are not found in the list of standard terms, the committee has included this section for the convenience of those who wish to become acquainted with the proper terms. It is hoped that the standard term will be used in place of a nonstandard term.

These cross references are inexact, however, since different meanings exist for the same term, as well as different nonstandard terms for the same meaning. Many nonstandard terms may still be used properly to express meanings that are not covered by standard terms or not in conflict with the latter.

Nonstandard Terms	*Standard Terms*
Anticipatory control	Rate action
Automatic reset	Proportional-speed floating action
Booster response	Rate action
Control band	Proportional band
Control effect	Corrective action
Control-index setting	Set point
Control instrument	Automatic controller
Control-point setting	Set point
Control response	Corrective action
Control setting	Set point
Controller function	Corrective action
Controller response	Corrective action
Conversion response	Proportional-position action
Corresponding control	Positioning action
Damping control	Rate action
Dead neutral	Neutral zone
Dead-period lag	Dead time
Dead spot	Differential gap or neutral zone
Dead zone	Differential gap or neutral zone
Deflection	Deviation
Delay	Dead time
Departure	Deviation
Desired condition	Desired value
Differentiating control	Rate action
Direct-operated controller	Self-operated controller
Displacement	Deviation and offset
Distance-velocity lag	Dead time
Drift	Offset

Nonstandard Terms	*Standard Terms*
Drift compensation	Proportional-speed floating action
Droop	Offset
Droop correction	Proportional-speed floating action
Elastic follow-up	Floating action
Finite time lag	Dead time
Floating component	Floating action
Floating response	Floating action
Floating sensitivity	Floating rate
Floating time	Reset rate
Flow-lag	Dead time
High-low control	Two-position control
Hunting	Cycling
Inactive neutral	Differential gap
Index setting	Set point
Integral of deviation	Integral action
Integrating control	Integral action
Interval	Dead time
Inverse minutes	Reset rate
Kicker	Rate action
Lapse	Dead time
Load error	Offset
Loss of control point	Offset
Measuring element	Measuring means and primary element
Modulating control	Proportional-position action
Noncorresponding	Floating action
Normal	Desired value
On-off action	Two-position action
Open and shut action	Two-position action
Oscillating	Cycling
Overshooting	Cycling
Per-rate response	Rate action
Per-time response	Floating action
Pilot-operated controller	Relay-operated controller
Rate-component control	Rate action
Rate-of-change method	Rate action
Rate-of-departure component	Rate action
Rate of droop correction	Reset rate
Rate response	Rate action
Regulation	Offset
Regulator controller	Automatic controller
Reset constant	Reset rate
Reset control	Proportional-speed floating action
Reset response	Proportional-speed floating action

Nonstandard Terms	*Standard Terms*
Reset sensitivity	Reset rate
Reset speed	Reset rate
Reset time	Reset rate
Response characteristic	Corrective action
Response delay	Dead time
Second derivative	Rate action
Self-acting controller	Self-operated controller
Self-actuated controller	Self-operated controller
Sensing element	Primary element
Sensitive element	Primary element and measuring means
Sensitivity	Proportional band
Servo	Power unit
Servo-operated controller	Relay-operated controller
Setting	Set point
Set value	Set point
Speed of reset	Reset rate
Swinging	Cycling
Throttling action	Proportional-position action
Throttling band	Proportional band
Throttling range	Proportional band
Time response	Rate action
Transportation lag	Dead time
Variable-speed reset	Proportional-speed floating action
Velocity-distance lag	Dead time

INDEX