

A GUIDE

TO

THE APPLICATION AND INSTALLATION

OF

LARGE BOILERS

IN

HYDRONIC HEATING SYSTEMS

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40 West 40th Street
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FOREWORD

To the Field Sales Personnel

"No salesmen ever lost an order because he knew too much about his product."

This Guide has been prepared to aid field sales personnel in the application, installation and sale of American-Standard cast iron boilers, especially the G-40, G-60, A-5 and A-7.

Fundamentals of heat, pressure, combustion and electricity have been covered for background.

In a few instances complete technical and scientific accuracy have been sacrificed in the interest of simplicity and brevity. Such omissions have no practical effect upon the understanding of fundamentals in the ranges of temperature and pressure applicable to large boilers.

The technical and practical information on steam systems, hot water systems, manifolding and breeching sizing, to name a few subjects, should be helpful in guiding the contractor, the architect and the consulting engineer on boiler replacement as well as new installations.

Certain special applications are discussed, the theory explained and tables prepared to simplify the arithmetic required for an accurate engineering approach to the problem.

It is suggested that you study the Guide and use it in its present tentative form. During the next year additional Chapters may be required. Perhaps some of its present Chapters will need expanding. Whatever the action taken will depend on the Guide's usefulness to you.

Clyde A. McKeeman

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CHAPTER 1

FUNDAMENTALS OF HEAT

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CHAPTER 1

FUNDAMENTALS OF HEAT

Including several ways of looking at heat... how it is measured...and how it is transferred.

We measure the intensity of heat with a thermometer. The thermometer tells "how hot" a substance is. It does not tell us how much heat the substance contains.

Obviously two quarts of boiling water hold more heat than a cupful... but the thermometer in each case reads 212°F.

We measure the quantity of heat in btu's (British Thermal Units). This measurement is based on two factors:

1) The temperature (Fahrenheit) or intensity of the heat in a substance.

2) The weight of the substance in pounds.

We can then use the btu to measure the heating power of fuels, heat losses and gains in buildings, the heat-carrying capacity of pipes, the capacity of boilers, etc.

One btu is the quantity of heat needed to raise one pound of water one degree fahrenheit. Two btu's will raise two pounds of water one degree or one pound of water two degrees.

How many btu's are required to heat 12 pounds of water at 32°F to 180°F?

Y equals 12 (pounds of water) x 1 x (180-32)
Y equals 12 x 1 x 148
Y equals 1776 btu's

So far we have discussed btu's as a measurement of the quantity of heat needed to raise temperatures. Actually it is a measurement of the quantity of heat exchanged.

If we take one pound of water at 70°F, put it on the stove and heat it to the boiling point, we will have used 142 btu's (212-70). If we then set the one pound of boiling water on a counter, it will liberate or give off 142 btu's, as it cools back to 70°F.

This experiment demonstrates two things:

1) The law of heat exchange: heat lost = heat gained.

2) The fact that heat flows when there is a difference in temperature...from the warmer to the cooler substance.

Specific Heat

Different substances have different capacities for absorbing (and giving off) heat. This capacity depends upon the specific heat. Specific heat is the number of btu's required to raise one pound of the substance 1°F.

Water, in the temperature ranges encountered in low temperature hot water heating, has a specific heat of one.

A cubic foot of water at atmospheric pressure and 70°F weighs 62.27 lbs. This water heated to 170°F absorbs 6227 btu or approximately 3460 times the heat absorption of air.

This explains in part why warm air systems need large ducts as compared with the small pipes needed in hot water systems.

The State of a Substance

Practically all substances can exist in more than one state. Metals are solid, or, when they are melted, liquid. If heated to a high enough temperature, they will vaporize. Water, our main interest, also exists in three states: at a pressure of 29.92 inches of mercury it is solid, as ice, at 32°F and below...liquid, as water, from 32°F and through 212°F...and in a gaseous state, steam, at 212°F and above. These temperature limits will change under pressure. We'll get to that in the next chapter.

Sensible Heat (or Enthalpy) and Latent Heat

First: the new word, "enthalpy". The dictionary defines it as, "the sum of the internal and the external energies of a fluid system; thermodynamic potential at constant pressure."

Because we are more interested in the heat potential or the ability to heat than we are in the inherent heat of the liquid, enthalpy is a more exact definition. However, some engineers (usually the older ones) will use the word "heat" usually in the phrase "heat content" while some (usually the younger ones) will use the word "enthalpy". Despite the difference in dictionary definition, they are both talking about the same thing.

Sensible heat is the heat required to change the temperature of substance. It gets its name from the fact that this form of heat can be sensed or felt. In measuring the sensible heat of a substance, this measurement must be made from some base temperature. With water a base temperature of 32°F is used, since this temperature is the lowest at which water can exist in the liquid state under standard atmospheric pressure. The sensible heat or enthalpy of one pound of water at 180°F is approximately 148 btu. Two pounds of water at 180°F would contain twice 148 btu, or approximately 296 btu of sensible heat or enthalpy.

Latent heat...or enthalpy...is heat that cannot be sensed, or felt. It is heat used to change a substance from one state to another.

Boiling water must be kept boiling by the application of heat to change it into steam. But because the temperature of both the water and the steam remain the same during boiling, this heat cannot be sensed, or felt. It is hidden or latent heat.

This latent heat put into boiling the water is then recovered, in a steam system, as the steam in the radiators **condenses back into the liquid state.**

The quantity of heat required to change one pound of boiling water into steam (970.3 btu at atmospheric pressure) is called the "latent enthalpy of vaporization" or just "enthalpy of vaporization". Again, some older engineers may substitute the word "heat"!

Steam: Saturated and Superheated

During boiling, steam rises from the surface of the water at the same temperature as the water and is called saturated steam. Its enthalpy content is equal to the sensible enthalpy of the water plus the latent enthalpy of evaporation.

Superheated steam is steam that has been removed from the presence of the boiling water by the steam main and then put through a series of tubes called a superheater. The tubes are then heated...the steam absorbs the additional heat and becomes superheated.

Additional heat cannot be applied to the steam in the presence of the water because the added heat would result only in the boiling of more water.

To see the effect of heat on water, let's apply heat to one pound of water at 32°F, at atmospheric pressure.

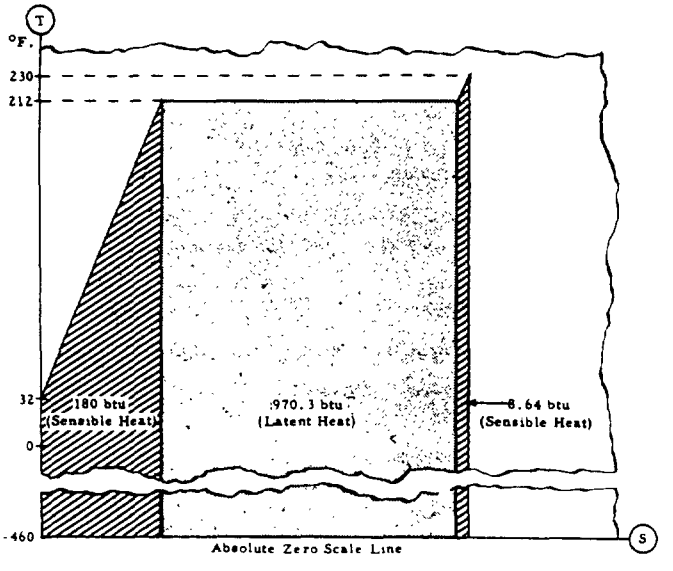


FIGURE 1-1

Raising one pound of water from 32°F to 212°F requires 180 btu of sensible heat. To boil the water completely into steam requires 970.3 btu of latent heat. Because the specific heat of superheated steam under these conditions is 0.48 only 8.64 btu of sensible heat is required. (230-212 x 1 x 0.48 = 8.64)

Thus, the total enthalpy of the saturated steam equals:

The enthalpy of the liquid	--	180.0 btu
plus the enthalpy of vaporization	--	970.3
	OR	1,150.3 btu

And the total enthalpy of the superheated steam equals:

The enthalpy of the liquid	--	180.0 btu
plus the enthalpy of vaporization	--	970.3
plus the enthalpy of superheat	--	8.64
	OR	1,158.94 btu

These figures are true at atmospheric pressure only. Variations in pressure will vary the temperature at which water boils and so, also vary the sensible, latent and total enthalpies.

Equivalent Direct Radiation (EDR) Steam

Steam heating boilers and radiation are rated in square feet of equivalent direct radiation (EDR). One EDR of steam equals 240 btu per hour.

In the early days of steam heating, the old large column radiators gave off 240 btu per hour per square foot of surface, when tested with steam at 215°F.

Modern radiation, being much more efficient, has changed the numbers but the theory remains the same.

Today's radiation is tested with steam under carefully maintained code conditions to determine its capacity in btu/hr. This figure is then divided by the old standard of 240 btu/hr. per square foot. The resultant figure is a rating in "square feet of radiation" (or EDR).

Though it has no relation to the actual square feet of surface under test, it does provide a standard to measure the capacity of various kinds and types of radiation and boilers.

Using EDR to Compute Relationship Between Radiation and Boiler

To convert one pound of boiling water to one pound of steam at 215°F requires a latent enthalpy of 968.4 btu.

The steam, as it condenses in the radiator, will of course deliver 968.4 btu through the radiation to the space to be heated.

Since one square foot of EDR equals 240 btu hr...4 square feet of EDR equals 960 btu/hr...or 4 square feet of EDR approximates the heat delivered by one pound of steam at 215°F.

EXAMPLE

How many pounds of steam must a boiler make to carry the load of 1600 square feet of radiation?

$$\frac{\text{1600 sq. ft.}}{\text{4 sq. ft./lb. of steam}}$$ equals 400 lbs. of steam/hr.

EXAMPLE

Specifications for a boiler state that it has the capacity to deliver 500 lbs. of steam per hour. What is the boiler's capacity in square feet of radiation?

500 lbs. steam/hr. x 4 sq. ft./lb. equals 2,000 sq. ft.

CHAPTER 2

FUNDAMENTALS OF PRESSURE

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FUNDAMENTALS OF PRESSURE

"Pressure", like many words in the English language, has many variations. For instance, "moral pressure", "pressure group", "pressing", "squeezing", "weight or burden", etc.

However, in this text, we are interested in the "mechanical" definition: "a force in the nature of a thrust, distributed over a surface".

In discussions of boilers, piping, radiation, etc., we can precisely measure pressure by measuring both the force and the area against which it is exerted.

Probably the most common expression of pressure is in "pounds per square inch"...or p.s.i.

Other ways to measure and state the force of a pressure are based on the weight of water and mercury.

Water: $62^{o}F$ Mercury: $32^{o}F$

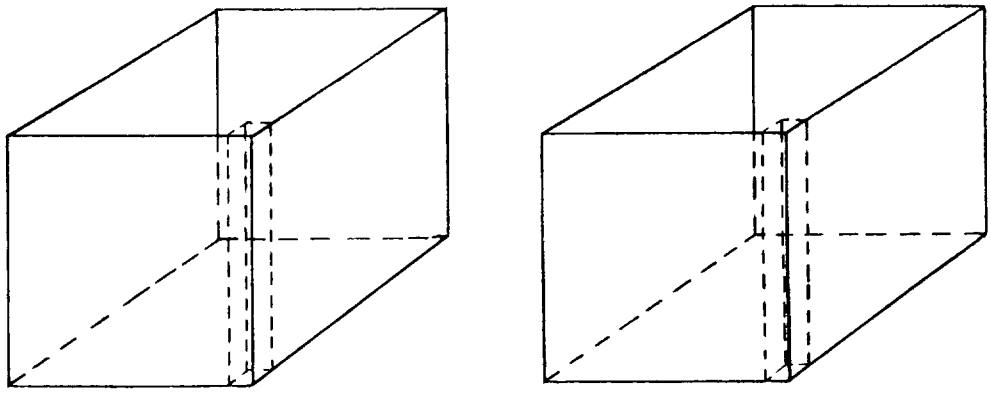


FIGURE 2-1

Assume two tanks 12" x 12" x 12"...one filled with water, one with mercury. Figure 2-1. We also assume water temperature of $62^{o}F$ and mercury temperature of $32^{o}F$. These are arbitrarily chosen standards to establish a definite relationship between weight and volume. Heated above $32^{o}F$, for instance, the mercury would expand and run over the sides of the tank. We would still have the same volume but less weight.

Water: $62^{\circ}F$		Mercury: $32^{\circ}F$
62.37 lbs.		848.2 lbs.
62.37 lbs. per sq. ft.	Total Pressure at Bottom	848.2 lbs. per sq. ft.
$\frac{62.37}{144} = .433$	Pressure lbs. per sq. in.	$\frac{848.2}{144} = 5.89$

We now see that 12 inches, or one foot column of water, exerts a pressure of .433 p.s.i. while one foot of mercury equals 5.89 p.s.i.

By computation, then, we can set up other standards:

If 1 foot of water equals .433 p.s.i., then 1 ÷ .433 or 2.31 feet (27.72") of water = 1 p.s.i.

If 1 foot of mercury equals 5.89 p.s.i., then 1 ÷ 5.89 or .17 feet (or 2.036") of mercury = 1 p.s.i.

Also 5.89 p.s.i. ÷ 12 will give the pressure 1 inch of mercury, .491 p.s.i.

We now have four different methods of expressing or defining "pressure":

1. Pounds per square inch
2. Inches of water
3. Feet of water
4. Inches of mercury

Why four? Basically, for the same reasons that space engineers discuss the dimensions of electronic components in "tenths of an inch"...the speed of supersonic aircraft in terms of "mach"...the distance between stars in terms of "light years"...to give them more practical and manageable figures to work with.

The selection of any of the four expressions for pressure is determined by the size of the force being defined.

For instance, boiler pressure, when greater than the atmosphere, is a large pressure and is expressed in "pounds per square inch", while the draft in a boiler is such an extremely small pressure, it is expressed in "inches of water", which, even then, usually turns out to be only a fraction.

The expressions "greater than atmosphere" and "less than atmosphere" are quite common. They point to the importance of the atmosphere in pressure measurements.

Atmospheric pressure is all around us. We live at the bottom of an ocean of air, which most time goes unnoticed. It seems to be a relatively light pressure, because we've grown used to it, and because it exerts equal pressure on all sides.

Atmospheric pressure is often referred to as barometric pressure, because it is measured with a barometer. The simplest barometer is a mercury barometer -- Figure 2-2

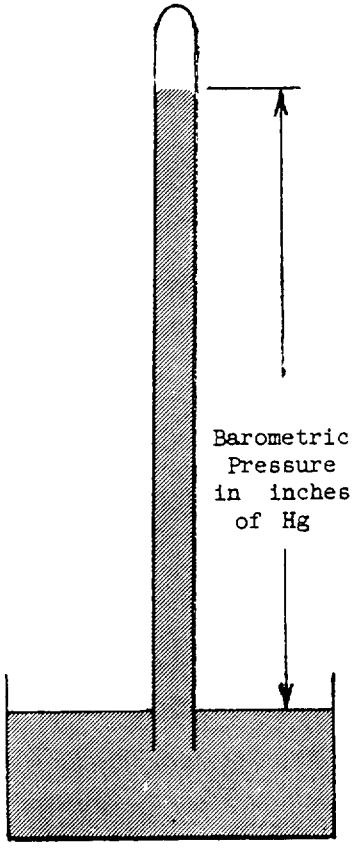


FIGURE 2-2

Take a glass tube, fill it with mercury...heat it to drive off air bubbles created in the filling process...put your finger over the open end to keep the mercury from running out...bury the open end in a pool of mercury open to the atmosphere...bring the tube to a vertical position... arrange a scale so you can read the height of the mercury in the tube above the mercury in the pool ...and you have a mercury barometer.

Some of the mercury will run out of the tube into the pool, leaving a vacuum at the top of the tube.

How much will run out? Just enough so as to leave the mercury in the column with weight enough to exert the same pressure as the pressure of the atmosphere on the surface of the pool surrounding the base of the tube.

Because stormy weather is preceded by a drop in atmospheric pressure which causes "a falling glass" and clearing weather with a rise in atmospheric pressure causing "a rising glass"...the barometer is a basic instrument in weather forcasting.

In Figure 2-3 we show 29.92 inches because that is the height of the mercury column under average conditions at sea level...and represents the equivalent pressure of 14.696 p.s.i. which is the pressure of a "standard atmosphere".

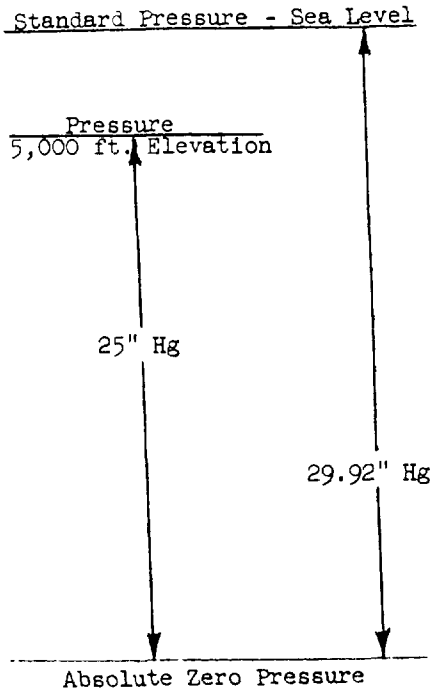


FIGURE 2-3

At higher altitudes, because the amount of air above you is less, the atmospheric pressure is less. In Denver, Colorado, for instance, under average conditions the mercury column is approximately 25 inches high.

The readings in p.s.i. are obtained by multiplying the height of the column in inches by the density of mercury (.491 lbs. per cubic inch). This chart shows an atmospheric pressure of 14.696 p.s.i. at sea level and 12.279 at Denver.

Hg is the chemical symbol for Mercury.

Both of these pressures are measurements from an absolute zero pressure line and so they are (as are all barometric readings) absolute pressures.

Gage and Vacuum Pressure

The pressures inside a heating system are limited by the design of the equipment and the temperature at which it works. They are called gage pressures because they are measured by a gage.

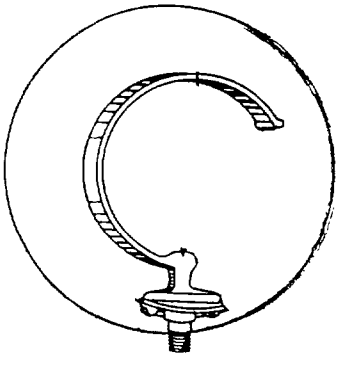
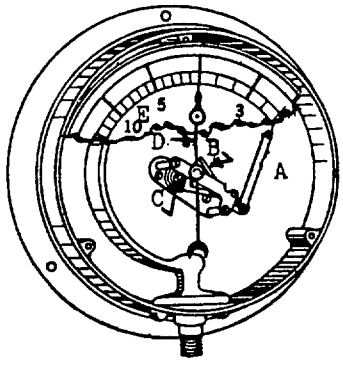


FIGURE 2-4

The most common gage is the Bourdon Tube, usually made of steel, and oval in its cross section.

Pressure from the boiler, filling the tube, tends to straighten it out, while atmospheric pressure, pushing on the outside, tends to make it curl up.

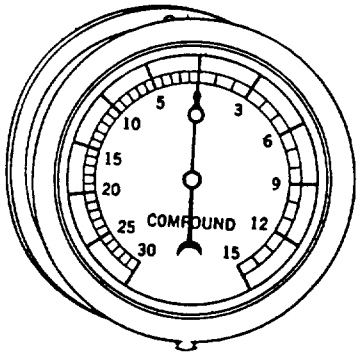
Actually, then, the tube takes a position determined by the difference between the boiler pressure and the atmospheric pressure.



By connecting the end of the Bourdon Tube in Figure 2-5 via linkage (A) to quadrant (B), which meshes with a small pinion gear (C) mounted on a shaft which carries pointer (D), we can calibrate a scale (E) to read the gage pressure.

FIGURE 2-5

In Figure 2-6 we show the indicator at zero, which means that boiler pressure is exactly compensating for the atmospheric pressure and precisely cancelling it out.



We can now finish calibrating our scale both up (when boiler pressure exceeds atmospheric pressure) and down (when boiler pressure falls below atmospheric pressure, creating a negative pressure or vacuum).

Arbitrarily, pressures above atmosphere are expressed in "pounds per square inch"...pressures below atmosphere are expressed in "inches of mercury."

FIGURE 2-6

Boiler pressure is then added to (in the case of gage pressure) or subtracted from (in the case of vacuum pressure) the existing atmospheric pressure, to arrive at the total or "absolute pressure" affecting the system.

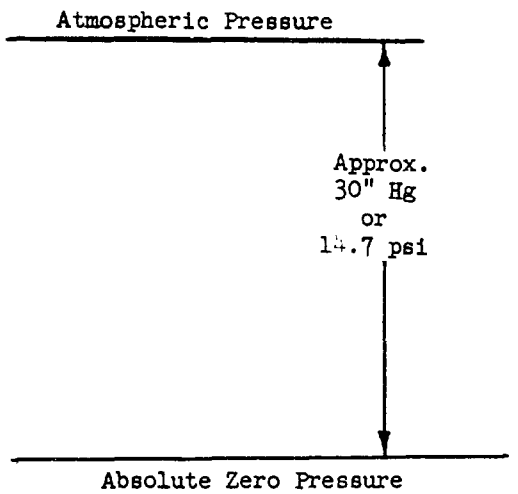


FIGURE 2-7

Starting at sea level, Figure 2-7, with no pressure added or subtracted by the functioning of the system, it is subject only to atmospheric pressure of 14.7 p.s.i. (rounding off 14.696), or 30 inches of mercury (rounding off 29.92).

To denote gage pressure we add the letter "g" to our abbreviation "p.s.i.".

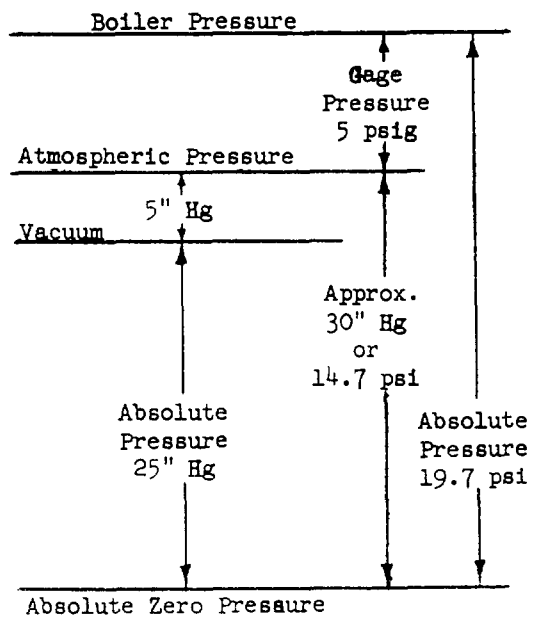


FIGURE 2-8

If the system then develops 5 p.s.i.g. of internal pressure, Figure 2-8, this gage pressure must be added to the atmospheric pressure to get the absolute pressure (19.7 p.s.i.) etc.

If the system builds up a vacuum, or negative pressure equal to the weight of 5 inches of mercury, we will have a drop in absolute pressure from 30 inches Hg to 25 inches Hg and so on.

Importance of Pressure

The knowledge of these pressures and how they are determined is important since the absolute pressure on a liquid determines its boiling temperature. This phenomenon holds true whether the liquid be water, Freon or a metallic liquid such as mercury.

For practical purposes particularly at elevations near sea level, boiling temperatures can be related to gage pressure. However, the atmospheric pressure diminishes about 4% for each 1,000 feet above sea level. Consequently, to get a reasonably accurate estimate of boiling temperatures, pressures must be corrected to absolute.

CHAPTER 3

FUNDAMENTALS OF COMBUSTION

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CHAPTER 3

FUNDAMENTALS OF COMBUSTION

Combustion...or burning...is the rapid oxidation of a substance and characteristically the result is the production of heat and light.

The word "oxidation" here means the combining of oxygen with another substance to form a third substance.

The combustion process, when methane gas is used as a fuel, can be expressed as:

$$CH_4 + 2\,O_2 \longrightarrow CO_2 + 2\,H_2O$$

These symbols mean:

CH_4 equals 1 cu. ft. of Methane (CH_4)

$2\,O_2$ equals 2 cu. ft. of Oxygen (O_2)

CO_2 equals 1 cu. ft. of Carbon Dioxide (CO_2)

$2\,H_2O$ equals 2 cu. ft. of water (H_2O)

So the equation now reads, "Combine one cubic foot of methane with 2 cubic feet of oxygen and the resultant combustion will leave one cubic foot of carbon dioxide and 2 cubic feet of water (in the form of steam) as flue gases.

Because the only practical way to get the oxygen is from the air around us, and because oxygen is only 21% of air by volume, we must, in theory, introduce 4-3/4 cubic feet of air into the system for every cubic foot of oxygen required, or 9-1/2 cubic feet of air to supply the 2 cubic feet of oxygen called for in our equation. The other 79% of the air is assumed to be nitrogen.

The left hand side of the equation now reads:

$$CH_4 + 9.5\ \text{Air}$$

But because we are primarily interested in the 2 cubic feet of oxygen, we separate it and the 7.5 cubic feet of nitrogen and write the equation this way:

$$CH_4 + 2\,O_2 + 7.5\,N_2 \longrightarrow CO_2 + 2\,H_2O + 7.5\,N_2$$

Now the formula says that when 1 cubic foot of methane (CH_4) is burned with 9.5 cubic feet of air the resultant flue gases are 1 cubic foot of CO_2, 2 cubic feet of steam and 7.5 cubic feet of nitrogen. The nitrogen does not join in the combustion process, but is heated along with the CO_2 and the H_2O. The heat that these products carry off is called stack loss.

The above formula represents perfect or theoretical combustion.

As a practical matter, theoretical perfection is impossible to attain. Therefore, in order to insure that every molecule of Methane is furnished the proper number of molecules of oxygen, excess air is introduced into the system to furnish additional oxygen.

With natural draft gas burners, approximately 50% excess air is added.

Since the excess air, representing a safety factor, is not used up in the combustion process, it is added to the flue gases...and increases the stack losses.

That's why 50% excess air is used, not more. Obviously 70% excess air would contain even more oxygen and represent an even greater safety factor.

But it would also represent extra gases to absorb more heat and create a higher chimney loss.

Adding surplus gases of any kind to the combustion products loses heat two ways:

1. In just heating more flue gases
2. All boilers have flues to hold the hot gases in the boiler so that the water jacket can absorb more of their heat before they are vented. As the quantity of flue gases increase the velocity of the gases through the flues increases. The gases moving through the flues faster have less time to give up their heat to the boiler surface and consequently go up the chimney at higher temperatures, thus increasing the stack loss. Stack temperatures from a given boiler have a direct relationship with the amount of flue gases going through it.

Now, let's look at our formula for perfect burning again,

$$CH_4 + 2\,O_2 + 7.5\,N_2 \rightarrow CO_2 + 2\,H_2O + 7.5\,N_2$$

and add the 50% air for complete practical burning.

$$CH_4 + 3\,O_2 + 11.25\,N_2 \rightarrow CO_2 + 2\,H_2O + 11.25\,N_2 + O_2$$

Now we have 15.25 cubic feet of flue gas.

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If a CO_2 analysis was made on the products of combustion for perfect combustion conditions, this analysis would show 11.8% CO_2 by volume. See the following tabulation:

Perfect Combustion

			% by Volume
	CO_2	1.0 cu. ft.	11.8%
(Steam)	H_2O	2.0 cu. ft.	
	N_2	7.5 cu. ft.	88.2%
	O_2	0.0 cu. ft.	
	TOTAL	10.5 cu. ft.	
	*Less H_2O	2.0	
Total Dry products of Combustion. . . .		8.5 cu. ft.	100.0%

*NOTE: When flue gas analyses are made, the steam condenses making it impossible to obtain its volume.

Similarly the percentage of CO_2 by volume analysis for practical or complete combustion of this gas becomes 7.5% as shown in the following tabulation:

Practical or Complete Combustion

			% by Volume
	CO_2	1.0 cu. ft.	7.55%
(Steam)	H_2O	2.0 cu. ft.	
	N_2	11.25 cu. ft.	84.90
	O_2	1.0 cu. ft.	7.55
	TOTAL	15.25	
	*Less H_2O	2.0	
Total Dry products of Combustion.....		13.25 cu. ft.	100.0%

*NOTE: When flue gas analyses are made, the steam condenses making it impossible to obtain its volume.

It is evident that as the percentage of CO_2 in the flue gases diminishes when burning a given fuel, the amount of excess air is increasing. Hence, the CO_2 percentage becomes a measure of the amount of excess air and, when combined in a formula or with curves of stack temperature, gives a basis for determining the boiler efficiency.

When analyzing flue gases the percentage of CO_2 which is obtainable under practical or complete combustion conditions varies with the kind of fuel due to the different percentages of carbon in these fuels. As the carbon content becomes larger so does the per cent of CO_2. The type of fuel burning equipment plays a part.

For natural gas, CO_2 readings for practical combustion conditions may range from 8% to 10%, for oil 9% to 11% and for coal (stoker fired) 12% to 13%.

It has been pointed out earlier in this chapter that complete combustion of all the oxidizable elements in the fuel is desirable. To accomplish this complete combustion excess air must be added.

When fuel is not burned completely, heat is lost. Should insufficient air cause the carbon in the fuel to burn to carbon monoxide (CO) not only is a poisonous gas formed but only 5,500 btu per pound of carbon are released. One pound of carbon burning to CO_2 releases 14,600 btu.

A similar result occurs with the hydrogen in the fuel. If sufficient oxygen is present to burn one pound of hydrogen to water vapor, 62,000 btu are released. When insufficient air is present to burn all the hydrogen, the unburned percentage escapes up the chimney and a proportionate amount of heat is lost.

The burning of oil efficiently presents a greater problem than burning the other two fuels. First the oil must be atomized into very small droplets. This is done by the nozzle. Secondly, the oil droplets must be vaporized. This action is accomplished by the combustion chamber walls. The air which is mixed with the vaporized oil now is able to burn it.

Materials used in combustion chambers vary from stainless steel and light-weight refractory materials to fire brick. Dense fire brick is usually employed where oil is being burned continuously.

Intermittent firing such as practiced with American-Standard boilers requires light-weight materials for combustion chambers. The quicker the combustion chamber walls become incandescent, following the starting of the burner, the quicker the combustion conditions reach maximum efficiency.

FUNDAMENTALS OF ELECTRICITY

CHAPTER 4

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CHAPTER 4

FUNDAMENTALS OF ELECTRICITY

An electric current is formed by the movement of electrons.

Because electrons move easily in metals: silver, copper, aluminum, etc., they are used to "conduct" electricity from source to load.

Because electrons do not move easily in such materials as glass, porcelain, cotton and rubber, they are used as insulation.

The flow of electric current in a circuit is analagous to the flow of water in a piping system.

The pressure or force making the current flow is supplied by a generator or a battery.

Wire size and length of the circuit determine the quantity of electricity flowing.

Large wires have less resistance to current flow than small ones.

The terminology in electricity is as follows:

1. The pressure or electromotive force (EMF) is measured in volts.
2. The quantity of electricity flowing is measured in amperes.
3. A rate of flow can be expressed as ampere hours.
4. Power equals volts X ampere hours divided by hours or watts.

A German physicist, George Simon Ohm, first studied electricity flow in a circuit in 1822. He found that in every circuit the amount of current is dirctly proportional to the pressure, or EMF, applied, and inversely proportional to the resistance in the circuit...of $I = \frac{E}{R}$

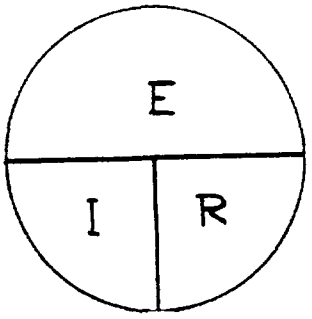
The formula, $I = \frac{E}{R}$, is called Ohm's law.

Resistance varies from circuit to circuit for three reasons.

1. Various conductors have different resistances. Silver is the best conductor and copper is almost as good (and a lot cheaper). Aluminum offers greater resistance, but because of its weight, is often used for large overhead cables.

2. The length of the circuit. The longer the wire, the greater the resistance.

3. The thickness of the conductor. The thinner the wire the greater the resistance.



A convenient way to remember Ohm's law. Cover what you're looking for and the rest of the diagram will tell you the formula. To find amps...cover amps in the diagram and the formula then is $\frac{E}{R}$. Similarly R is $\frac{E}{I}$. Covering volts leaves I x R.

FIGURE 4-1

One other important formula for measuring electricity and its effects is the relationship between E (volts), current (amperes) and power delivered (watts):

Watts equals Amperes X Volts

A 60-watt bulb connected to a 110-volt supply uses:

$60 = A \times 110$ or $\frac{60}{110} = .55$ amps

A toaster using 11.5 amps in a 110-volt supply uses:

$W = 11.5 \times 110$ or 1265 watts

Electricity is either DC (direct current) or AC (alternating current). In direct current the electrons move around the circuit always in the same direction. In the alternating current circuit, they move in one direction for a time, then reverse themselves and move in the opposite direction. This is similar to the action of a piston in a cylinder of your automobile engine.

And, as with the piston in a cylinder, one move in both directions is called a cycle. The standard in this country is 60 cycles per second, though there are some variations. Before installing any piece of electrical equipment it is very important to know the AC cycle time as well as the voltage and amperage available.

The generation of electric current is a relatively simple principle. If a wire in a circuit is moved through a magnetic field an electric current will be produced in the wire.

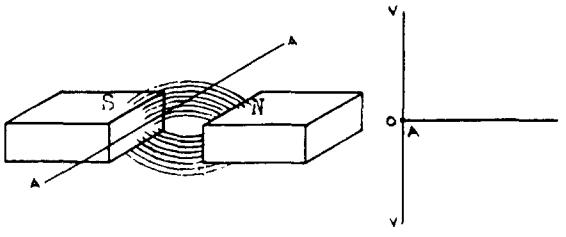


FIGURE 4-2

With the north and south poles of a magnet facing each other, a magnetic field is set up between them. A wire connected to a circuit at position "A" would have no voltage generated and no current flowing through it.

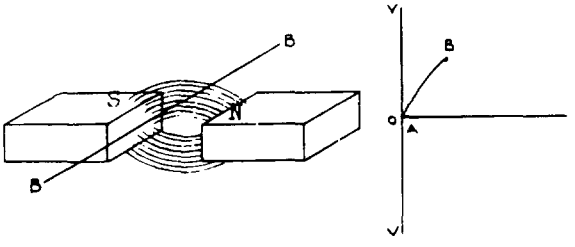


FIGURE 4-3

As the wire is moved through the magnetic field, however, voltage is created and the current starts to flow and reaches its maximum value in both volts and amperes at point "B" where the magnetic field reaches it maximum density.

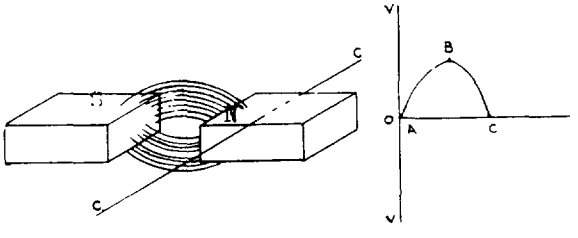
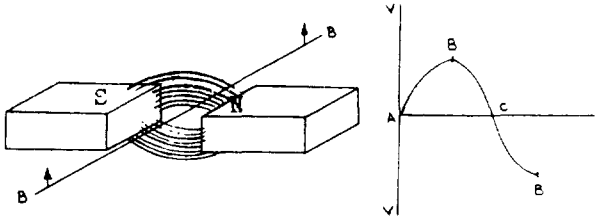


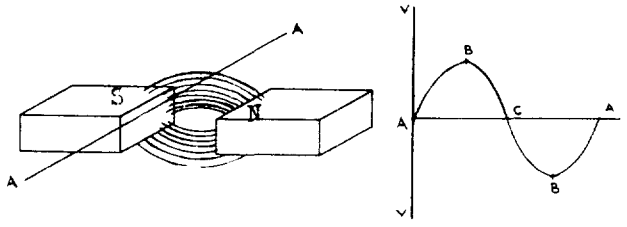
FIGURE 4-4

As the wire continues to move toward "C" the voltage and current will lessen again, until at "C", it once again reaches zero.



When the wire is then moved back through the magnetic field, voltage and current will again reach maximum value at point "B". Only, this time, because of the reversed movement, the voltage and current will be in the opposite direction.

FIGURE 4-5



Continuing back toward "A", the voltage and current will, again, lessen and, again reach zero at "A"...completing one cycle in the generation of AC current.

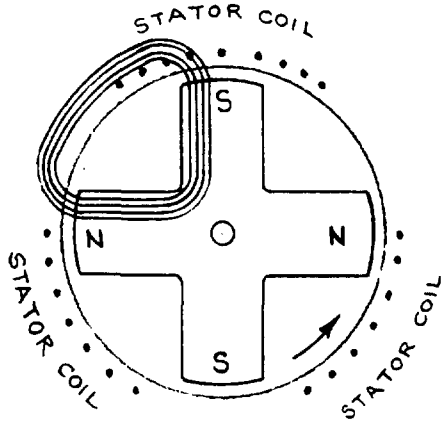
FIGURE 4-6

The amount of current depends on the strength of the magnetic field, and the size of the wire.

Obviously, if the circuit included a bulb, and the frequency of cycling was very low, the bulb would flicker. At 60 cycles per second the alternate build-up and fall-off of current have no apparent effects on the continuity of light given off by an electric bulb, since it does not have time to cool down.

Motors are roughly similar to generators. The difference is that with a generator, mechanical energy is used to turn the coil which makes it turn and deliver mechanical energy.

The same effect of alternating current can be accomplished by holding the wire stationary and moving the magnet.



To generate large amounts of power, huge electromagnets are rotated in the center of a series of coils.

As the poles of the magnet spin, their magnetic fields rotate and induce current in the coils.

FIGURE 4-7

The frequency is determined by the number of magnets and the speed with which the magnets turn...the amount of current generated is determined by the power of the magnet and the dimensions of the wire in the coils.

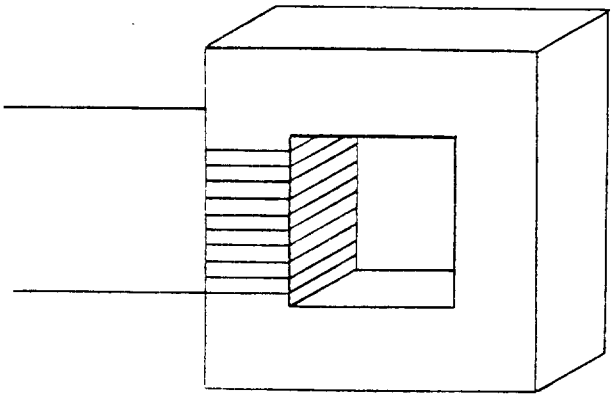
To obtain the sixty cycle current commonly in use in this country generators operate as follows:

with 2 pole magnet	---	3600 r.p.m.
4 pole magnet	---	1800 r.p.m.
6 pole magnet	---	1200 r.p.m.

The above listed r.p.m.'s are known as synchronous speeds.

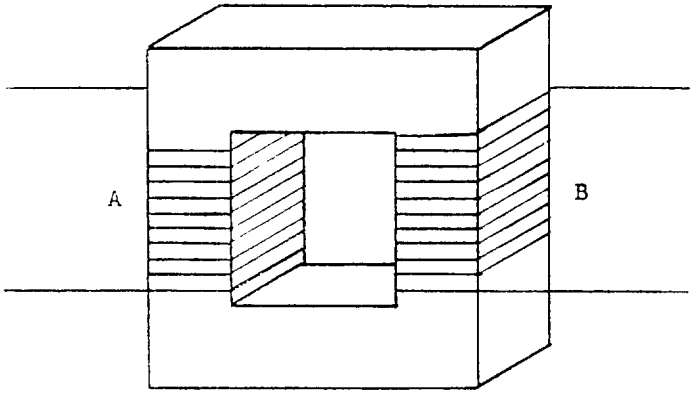
Induction motors try to attain the same synchronous speeds depending on the number of poles they have. For example an induction motor rated at 1750 r.p.m. is designed with 4 poles. The motor load slows it down to the rated speed. Similarly a 3450 r.p.m. induction motor is of two pole design.

Another use of the electromagnet is to keep the magnet stationary and move the magnetic flux inside the magnet. This is the principle used in the transformer.



When a coil around a magnet is connected to an AC source, magnetic flux is built up in, and travels around, the magnet. This flux movement corresponds to the AC current building up and then falling off... first in one direction and then the other

FIGURE 4-8



If we then wrap another, secondary coil around the magnet, the magnetic flux moving in the magnet will "induce" a current in this secondary coil.

The important features of this transformer are based on the fact that the relationship of voltage between the primary coil "A" and secondary coil "B" are determined by the number of turns in each coil. The cycles in the secondary coil will be the same as in the primary.

FIGURE 4-9

If the primary coil had double the turns of the secondary, we would have a step-down transformer and the result would be reversed: Voltage would be cut in half, and the amperage doubled.

Example: In a 115/24 volt step-down transformer the primary coil has 230 turns. How many turns in the secondary coil?

$$\text{No. of turns} = 230 \times \frac{24}{115} \quad \text{or } 48 \text{ turns}$$

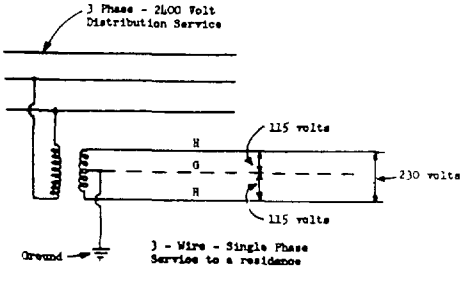
The transformer allows us to alter available voltage to suit the needs of various pieces of equipment.

The first and most important use of transformers, however, is to make it possible to transmit large amounts of electrical power over long distances, economically.

Large amperages need large diameter wires. But, because with a transformer with many turns on the secondary and relatively few on the primary we can build up the voltage, and at the same time reduce the amperage by very large amounts, power can be transmitted by relatively light, inexpensive cable. Long-Distance transmission voltages often exceed 300,000 volts.

Sub-stations may transform this voltage to something like 12,000 volts for short transmission systems with secondary sub-stations stepping it down even further to 2400 volts.

The 2400 volts, then, on its way to the consume.· (residential, commercial or industrial) is finally reduced by the transformer on the utility pole to 440, 230, 208, or 115 volts depending on need.



If your home has a 3-wire electric service offering you both 115 volts and 230 volts, this diagram shows the circuits from utility pole to the house.

FIGURE 4-10

The two wires at either end of the secondary coil are brought into the house. These are both hot wires and may carry a 230 volt current between them.

A third wire tapped into the center of the coil is brought into the house, but is also brought down the pole and grounded. This third wire, then, is neutral and carries 115 volts between it and either hot wire.

This gives us 230 volts across the two hot wires for 230 volt appliances like stoves and hot water heaters, for instance.

Also there are 115 volts for lighting and 115 volt appliances like toasters, radios, etc., between the ground wire and either hot wire.

This set-up is called a three-wire, single phase service. The three wire, single phase circuit provides greater flexibility of service, ease of installation with simple switching on the 115 volt side and safety.

For example, a light circuit in the house is shown in Figure 4-11. Note that by throwing the switch in the hot line it is safe to make repairs on the light socket since the other connection is already grounded.

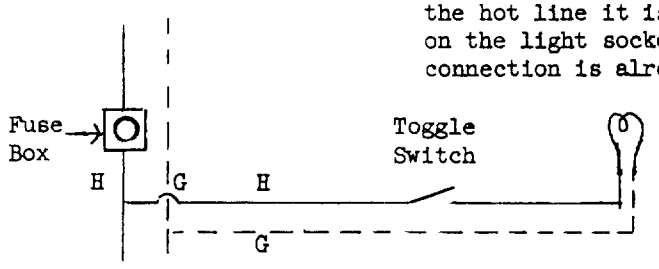


FIGURE 4-11

The same safety principles are applied for 115 volt circuits at the fuse box.

When 230 volts are involved with both wires hot, a double pole switch must be used to isolate both sides of the circuit.

For an additional understanding of alternating current problems, the term Power Factor is important.

$$\text{Power Factor} = \frac{\text{true watts}}{\text{apparent watts}}$$

or

$$\text{Power Factor} = \frac{P}{EI}$$

In electric resistance circuits such as lights, toasters and resistance electric heating the power P used equals EI. Therefore, for this type of equipment the Power Factor = $\frac{EI}{EI}$ = 1.0.

In other words, the wattage measured by the electronic meter equals exactly the volts x amperes furnished the user by the utility company.

Electric motors are a different story. The power measured by the electric meter on a motor circuit is often less than the product of volts times amperes that are supplied by the utility and the resulting power factor becomes a fraction. This condition is undesirable from the utility viewpoint. The utility is paid on the basis of power used but it must design its equipment, and size its circuits on the larger product of volts times amperes. Where large quantities of power are involved, the utility usually makes an additional charge above the meter reading when power factor conditions exist.

Some Electrical Circuit Symbols

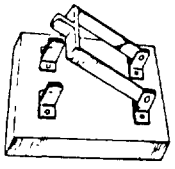


FIGURE 4-12

A double-pole, single-throw, designed to make or break both sides of the same circuit at the same time, when neither side is grounded and they are both hot.

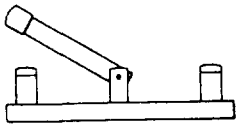


FIGURE 4-13

A single-pole, double-throw designed to make either one circuit or another or to break both of them.

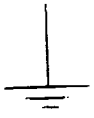


FIGURE 4-14

The symbol used to indicate a ground line.



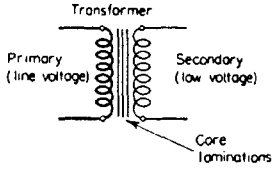
FIGURE 4-15

Resistors, used to reduce current flow.



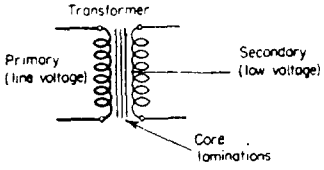
Variable resistor...or rheostat... allows the voltage to be varied between zero and line voltage.

FIGURE 4-16



An iron core transformer.

FIGURE 4-17



Iron core transformer with midtap giving us two different voltages.

FIGURE 4-18

CONTROL CIRCUITS

CHAPTER 5

CHAPTER 5

CONTROL CIRCUITS

There are two kinds of controls used in conjunction with big boilers:

(1) Operating Controls
(2) Safety Controls

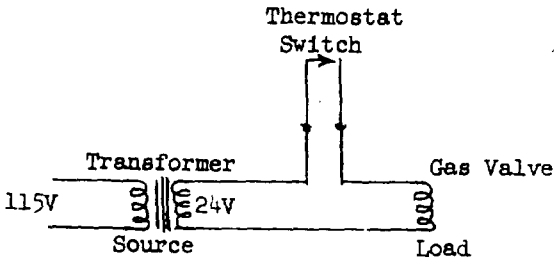
In the early heating systems "tending the furnace" was a common phrase. It was operated almost completely by hand and watched often.

The growth and refinement of controls has been and continues to be a steady, consistent process. For instance, American-Standard electronic controls, optional equipment with our large gas-fired boilers, are one of the latest control systems to be devised. But even they will continue to be researched, developed and refined.

Before we get to the actual controls, this chapter will discuss the four basic ways controls operate.

(1) Electrically
(2) Mechanically
(3) Pneumatically
(4) Any combination of these three

An Electrical Heating Control Circuit



Electrical control circuits always contain a source, a switch and a load. They can be high voltage (115-230V), low voltage (20-24V) or, in more complex controls, a combination.

The control circuit shown is a low voltage circuit controlling the main valve in a gas fired boiler.

FIGURE 5-1

While it is hooked into the main 115 volt line, it incorporates a 115/24 volt step-down transformer which is the usual voltage for the load -- a solenoid coil in ghe gas valve.

In this case, the transformer and not the main line is considered to be the source. When the switch closes a current flows through the small solenoid coil. The coil then actuates a valve mechanism causing the gas above the diaphragm of the main gas valve to be vented. The line pressure of the gas under the diaphragm will force it upward opening the main valve and permitting the gas to flow to the burners.

The switch is a thermostat which is an example of a mechanical control. Before we discuss it, points should be made about the circuit in Figure 5-1.

(1) This is the simplest kind of control circuit.
(2) It is an operating control only. It needs, as a safety device, other switches which will be discussed in Chapter 6.
(3) It shows only one of several kinds of valves, which will be covered in future chapters.

A Mechanical Heating Control Circuit

Thermostats, which act as switches in heating systems, by mechanical movements, make or break electric circuits.

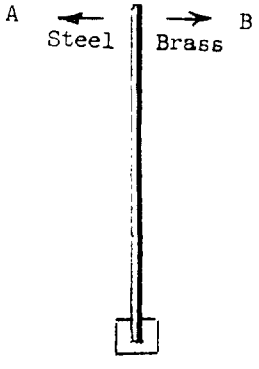


FIGURE 5-2

The simplest thermostat is a bimetal strip...two different metals welded together, usually brass or copper in conjunction with steel.

Because both copper and brass have a larger coefficient of expansion -- and contraction -- than steel, a rise in temperature will cause the bimetal strip to bend in direction of arrow A. A drop in temperature makes it bend in the direction of arrow B.



FIGURE 5-3

Two Wire Thermostat

One end of the straight bimetal strip is fastened rigidly to a base leaving the other end free to warp under temperature change. Also one wire is connected to the bimetal at the base and a contact button (A) fastened to the free moving end. The second wire to the thermostat is connected to the fixed contact (B). Wired into the circuit at the base of the thermostat is a small electric heater (C). As the room temperature approaches the control point -- say 72° F -- the

bimetal warps toward the fixed contact (B). As the contact points (A) and (B) approach each other, magnet (D) exerts a pull on the bimetal causing a positive contact of the points, thus completing the circuit and causing current to flow.

In the first thermostats there were no magnets or heaters. With the light bimetal elements employed to get sensitivity the contact points did not close or open with a positive motion. Slight vibrations in the structure would make the contacts chatter creating an electric arcing condition. This arcing causes pitting of the points. Under severe conditions the points may fuse together or freeze.

Modern thermostat design retains the light bimetal element but introduces the magnet to insure positive closing of the contacts. It then was found that an approximate $3^{\circ}F$ rise in room temperature would be required for the bimetal to develop enough warping force to overcome the magnet and open the contacts. Since a $3^{\circ}F$ swing in room temperature is not comfortable the additional temperature was created artifically under the thermostat cover by the heating element.

The thermostat closes its contacts and completes the electric circuit under room temperature conditions, but opens the circuit under an artificial temperature created under the thermostat cover.

When the thermostat closes its contacts and completes the electric circuit the thermostat heater is activated. Thus the thermostat closes under room temperature conditions but opens under an artificial temperature condition under the cover. With this arrangement a variation of room temperature from 0.5° F above the control point is achieved which is entirely satisfactory for human comfort.

The heater in the two wire thermostat presents a problem. It must be matched to the current draw of the load. An improperly sized heater causes improper control of room temperature. Too high an amperage through the heater causes the thermostat to open too quickly preventing the room from reaching the desired temperature. Too low an amperage through the heater slows the build-up of temperature under the cover, delays the opening of the contacts and causes the room to overheat.

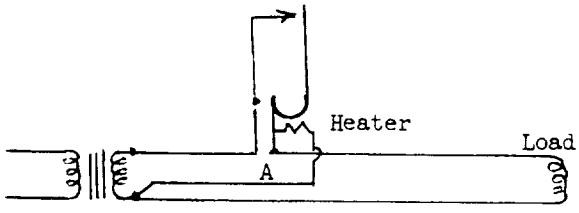


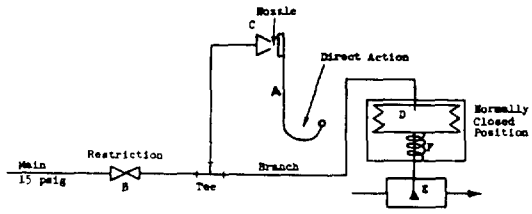
FIGURE 5-4

The three wire thermostat was designed which put the heater circuit in parallel with the load circuit thus making the heater performance independent of the amperage draw of the load. Figure 5-4 shows how the current flowing through the heater -- circuit A -- returns to the transformer without passing through the load, permitting all thermostats to be furnished with the same heater.

Heating installers had difficulty in understanding the three wire principal and thermostats were wired improperly. Consequently, the two wire thermostat is one commonly used today.

The action of the bimetal element, alone, is a mechanical circuit, where physical movement is used to make or break an electrical circuit. The total circuit is, of course, a combination of mechanical and electrical circuits.

A Pneumatic Heating Control Circuit



This diagram represents a pneumatic arrangement for handling the function of the main thermostat.

This circuit combines mechanical and pneumatic circuits. The familiar bimetal strip (A) is again used as the temperature-sensitive switch to make or break the circuit. Because this time it is a circuit of air rather than an electrical circuit we call the switch a valve, and the contact is a pad rather than an electrical contact.

FIGURE 5-5

Air from a compressor flows through the main which has an inner orifice (B), with the same air capacity as the nozzle (C) opposite the pad on the bimetal strip.

Air flow is constant, and with the switch in the open position, opening (C) will vent to the atmosphere all the air passed by orifice (B), thus preventing the build-up of pressure in bellows (D). The bellows is also held closed by the tension of spring (F).

However, when the space cools, the bimetal element will warp to the left, the pad will cover opening (C) and air coming through orifice (B) builds-up pressure in the bellows (D), overcomes the resistance of the spring and moves the plunger (E) which opens a valve.

BASIC CONTROL SYSTEMS

CHAPTER 6

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CHAPTER 6

BASIC CONTROL SYSTEMS

The modern boiler is almost completely automatic. It has a full set of both operating and safety controls -- all meshed together in a "control system."

These systems did not suddenly appear, full blown. They grew and developed; operating controls out of necessity for supplying customer comfort and convenience and safety controls out of the need for protecting equipment and property.

We have already seen how the thermostat operated by the room temperature can, through a combination electro-mechanical system, operates a gas valve to start and stop fuel flow.

This same principle can be used to operate other kinds of valves or switches. Later in this chapter we will discuss the valves used with American-Standard boilers.

We can now start and stop fuel flow. Some way to ignite the fuel is needed. For gas a pilot light is used. Gas may be supplied to the pilot through a small line tapped off the gas main.

Pilot lights are quite dependable. But they may go out for a variety of reasons. So, a safety device is required to prevent gas being turned on in the main burner when the pilot light is out.

The safety device used today on pilots is the thermocouple. It is constructed by joining two dissimilar metals -- for example, two wires -- at the ends only. In between the two junction points the wires are insulated. See Figure 6-1.

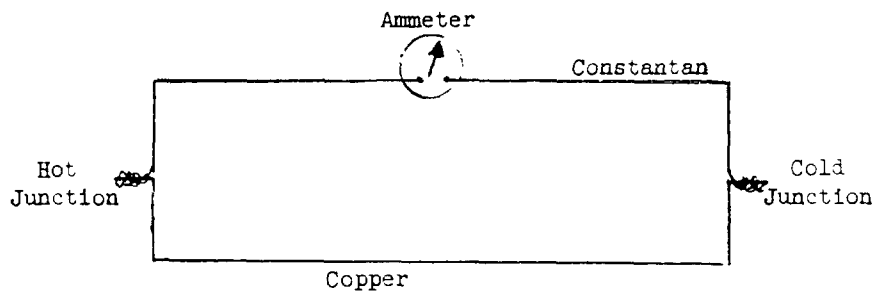


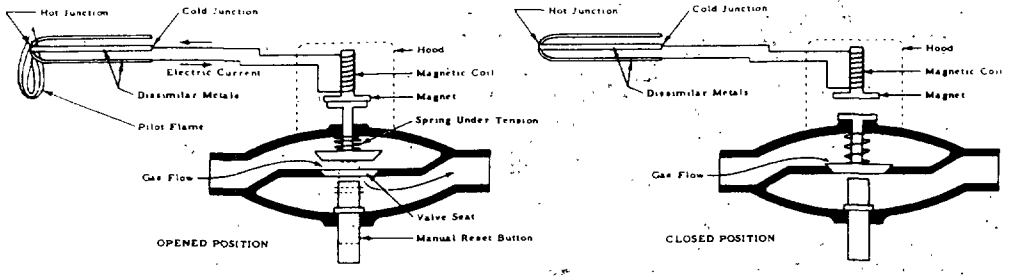
FIGURE 6-1

When one junction of the circuit is heated and the other cooled, a voltage difference is created between the two junctions and an electrical current flows. The greater the temperature difference, the greater the voltage.

This same electrical system, when a direct current from an external source is passed through it, will cause cooling of one junction and heating of the other. This phenomenon is called the Peltier Effect. It may be used to create refrigeration. Some small capacity refrigeration units have been constructed. Unfortunately, efficiencies are too low at present to permit using this principle except in very special applications.

The most common application of the thermocouple is the measurement of temperature. By measuring the voltage generated, or in a circuit of a fixed length and resistance by measuring the current flow, (the temperature difference between the hot and cold junctions) may be determined. Knowing the temperature of the cold junction, the temperature of the hot junction may be accurately determined.

We use the thermocouple in a pilot flame to generate the necessary electrical current to hold open the Baso Valve located in the main burner manifold. Failure of the pilot flame and the cooling of the thermocouple cuases the Baso Valve to close preventing the gas from reaching the main gas valve. See Figures 6-2.



FIGURES 6-2

The principle is also applied to the Baso Switch located in the electrical circuit between the thermostat and the main gas valve. The Baso Switch is held in a normally closed position by the current from the thermocouple. Failure of the pilot flame and the loss of electrical energy from the thermocouple will cause the switch to open thus preventing the main gas valve from opening.

Another safety problem is created when a thermostat, affected by a steady stream of cold air from an open door or similar condition, causes the boiler burner to stay on. This continuous burner operation, if permitted to continue, would cause the water in the water boiler to become too hot or excessive pressure in the steam boiler.

The answer is another switch in the electrical circuit to the gas valve. This switch is normally closed. It is the high limit.

In water boilers the high limit is an immersion control. When the water temperature rises above the control setting the switch opens and stops the burner.

In a steam boiler the switch is actuated by a spring loaded bellows or diaphragm which in turn is moved by the steam pressure to shut off the burner.

The thermal element in the immersion control may be a bimetal helix or a bulb charged with a temperature-responsive solid or liquid. When high limits are installed on the external surface of the boiler a light weight metal disc, which changes shape when heated, is used.

Another control used on boilers in which a minimum water temperature is required, is the operating immersion control sometimes called the "low-limit." Its function is to operate the burner when the water temperature falls below the desired level.

Still another is the reverse acting immersion control. It is called reverse acting because the switch opens with a drop in water temperature. This action is opposite to that of the immersion controls described previously.

Its application is:

1. In a water boiler with a tankless heater. It is located in the boiler so that it feels the drop in boiler water temperature when domestic hot water is being drawn from the tankless heater. It stops the circulator and the delivery of heat to the house while the domestic hot water demand is being satisfied. Since the demands for domestic hot water are of short duration this action has no noticeable effect on the house temperature.

2. When installed in boilers without tankless heaters the control may be set to operate at about 100 degrees F. It then stops the circulator when the boiler water temperature is approximately 100 degrees F. or less. This action prevents any tendency toward moisture condensation in the boiler flues and automatically stops the pump during the non-heating months.

Another control frequently used in hydronic heating systems is the relay. Its purpose is to tie together a low voltage (24 volts) control circuit with high voltage (115 volts) circuit. See Figure 6-3.

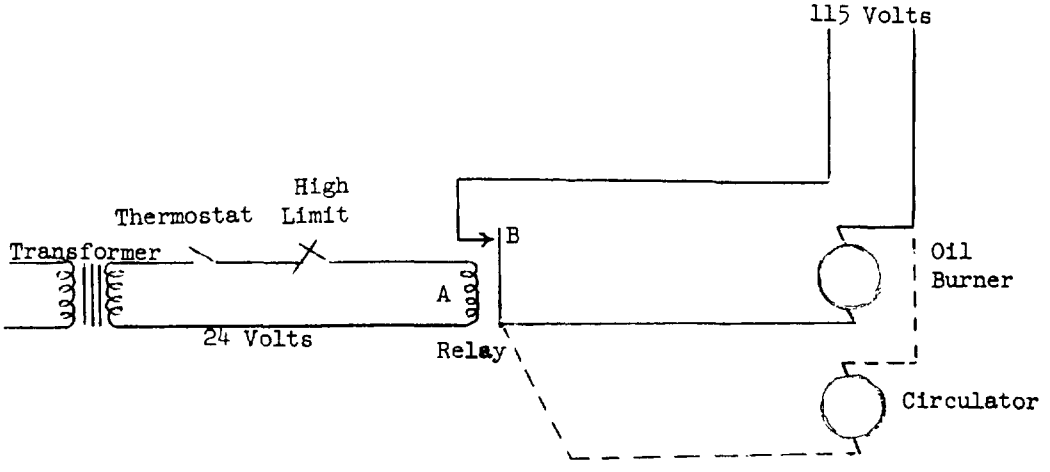


FIGURE 6-3

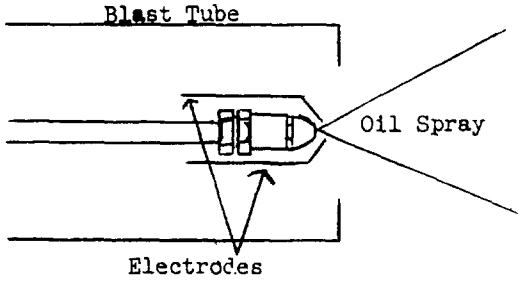
Because low voltage thermostats are inherently more sensitive and low voltage wiring does not require a licensed electrician most thermostat circuits are 24 volts. The oil burner motor is for use with 115 volts. To start the oil burner when the house needs heat requires the simplified diagram shown. When the thermostat closes, solenoid coil A is activated creating a magnetic pull on the switch B closing the switch points and completing the 115 volt circuit. When the thermostat is satisfied, the 24 volt circuit is de-activated, the contact arm B falls away and the oil burner stops. By connecting a circulator to the 115 volt circuit as shown by the dotted lines the same relay would activate the circulator. When selecting a relay in the field, always its capacity should be checked to determine if the amperage capacity is equal to or greater than the connected load.

Separate controls and their functions have been discussed. Present trends in the industry are to combine two or more of the immersion controls into a single package, which performs the individual functions, using one thermal element. For example, with the G-2 or G-40 water boiler operating with a tankless heater, a combination control is supplied that combines the high limit control, the operating control (low limit) and the reverse acting control. In addition this control includes a built-in transformer and relay.

This combining of more than one control function into a single control package has a number of advantages. It permits a more advantageous location of the control in the boiler and eliminates the need

for additional tappings thereby simplifying boiler and jacket design. Control wiring is made easier and boiler appearance is improved.

The same principles are used in controlling oil-fired boilers. The difference in fuels, however, requires a different kind of ignition and a different safety circuit.



When heat is required in an oil fired boiler, oil is forced through a small hole in the nozzle of the oil burner. This action atomizes the oil creating a mist of very tiny droplets. At the same time a high voltage, impressed on the electrodes, creates an arc. The arc passing through the oil mist ignites the fuel.

FIGURE 6-4

If the oil should not ignite, the burner must be stopped to prevent a possible explosion.

This safety control may be one of two devices.

The stack switch with a bimetal helix has been used in this safety role for a long time. As soon as the oil starts to burn, the flue gas temperature starts to rise. A 100 degrees F increase in stack temperature tells this control that the oil is now burning and that the oil burner may continue to operate until the demand for heat has been satisfied. If, after about 90 seconds of oil burner operation there is less than 100° F of increase in stack temperature, the stack switch stops the oil burner.

A more recent safety control used with oil burners is the light sensing cell. This cell or flame detector tells the primary controller that the oil spray has ignited. The primary controller then permits the oil burner to operate until the demand for heat is satisfied. Should the oil not ignite after 45 seconds of oil burner operation, no signal is received from the flame detector and the primary controller stops the burner.

The flame detector is a cell that is sensitive to the oil flame color only. It is located at the back of the burner blast tube and pointed at the flame. The cell acts as a switch. It permits an electrical current to pass through it when the oil is burning. This current tells the primary control that ignition has been established.

Ignition of the oil may be continuous or intermittent. With continuous ignition the electric arc operates as long as the burner operates.

In intermittent ignition the arc operates until the oil flame has been established.

In case of ignition failure, the stack switch will stop the oil burner. It must be manually reset.

A drop in line voltage will cause the stack switch to stop the oil burner. Automatic re-cycling of the oil burner will occur when line voltage is restored.

When ignition fails, the primary control used with the flame sensing cell also stops the burner. It must be manually reset. Similar to the stack switch the primary control stops the burner when the line voltage drops to 90 volts or less. On full power resumption the burner automatically re-cycles.

Electronic boiler controls have enjoyed increased application in recent years particularly on large gas boiler installations. They have greater sensitivity, fast action and a high degree of reliability.

These electronic controls are required on both the G-40 and the G-60 boilers when specifications call for the requirements of either the Factory Mutual Association (F. M.) or the Factory Insurance Association (F. I. A.).

Factory Mutual Association (F. M.)

The electronic control system specifically designed for the Factory Mutual Association requires electric ignition on all pilots (auxiliary and safety) and electronic control of all safety pilots. All pilots must have 100% shut off. A protector relay is necessary for each combustion chamber. The American-Standard EG-0 or the DPG-0 control packages meet these requirements.

In addition to the electronic controls F. M. requires the following valves on each gas manifold. See Figure 5-6.

1. Two manual shut off valves.
2. One gas pressure regulator.
3. One vent cock inserted in a special pipe nipple tapped 1/8 inch.
4. One General Controls line voltage H-0 Hydromotor Valve with iron body and bronze trim. Normally closed.

Factory Insurance Association (F. I. A.)

The Factory Insurance Association specifies the same electronic controls (EG-0 or EPG-0) as Factory Mutual. In addition F. I. A. requires the following valves and switches on each gas manifold. See Figure 5-7.

1. Two manual shut off valves.

2. One line voltage mercoid type PR high pressure semi-automatic switch set to open automatically at a pressure of 5" of water column. It must be manually reset. The pressure range of the switch is 1" to 30" water column pressure.
3. One line voltage mercoid type PRL-3 low pressure semi-automatic switch set to open when the manifold pressure drops to 2" of water column pressure. Manual reset is required. The pressure range of the switch is 1" to 30" water column pressure.
4. One line voltage General Controls H-O Hydromotor valve with iron body and bronze trim.

   NOTE: Both the K-3E 621 solenoid and the H-O valves are normally closed. They open on the electrical impulse received from the American-Standard pre-wired panel or the MG 890 E relay. This action occurs only after all pilots have been ignited.
5. One line voltage General Controls 3/4 inch iron pipe size vent valve No. K-10 CA 443. This valve is normally open when the solenoid (K-3E) and the H-O valve is closed. The vent valve closes when these two valves open.

Details of the General Controls valves are to be found in General Controls literature.

H-O Valve	Form No. SDI-H-OV-1
K-3 Valve	Form No. SDI-K-3-9
K-10 Valve	Form No. SDI-K-10-4

For the electronic controls details see Electronic Control Systems for G-40 and G-60 boilers, Form No. BP-15095, Revised 12-62.

The foregoing lists and arrangements satisfy the national requirements of both the Factory Mutual Association and the Factory Insurance Association. Local inspectors may request changes at times in the control equipment, and the order in which the components are installed on the gas supply piping.

CONTROL ARRANGEMENT FACTORY MUTUAL APPROVED

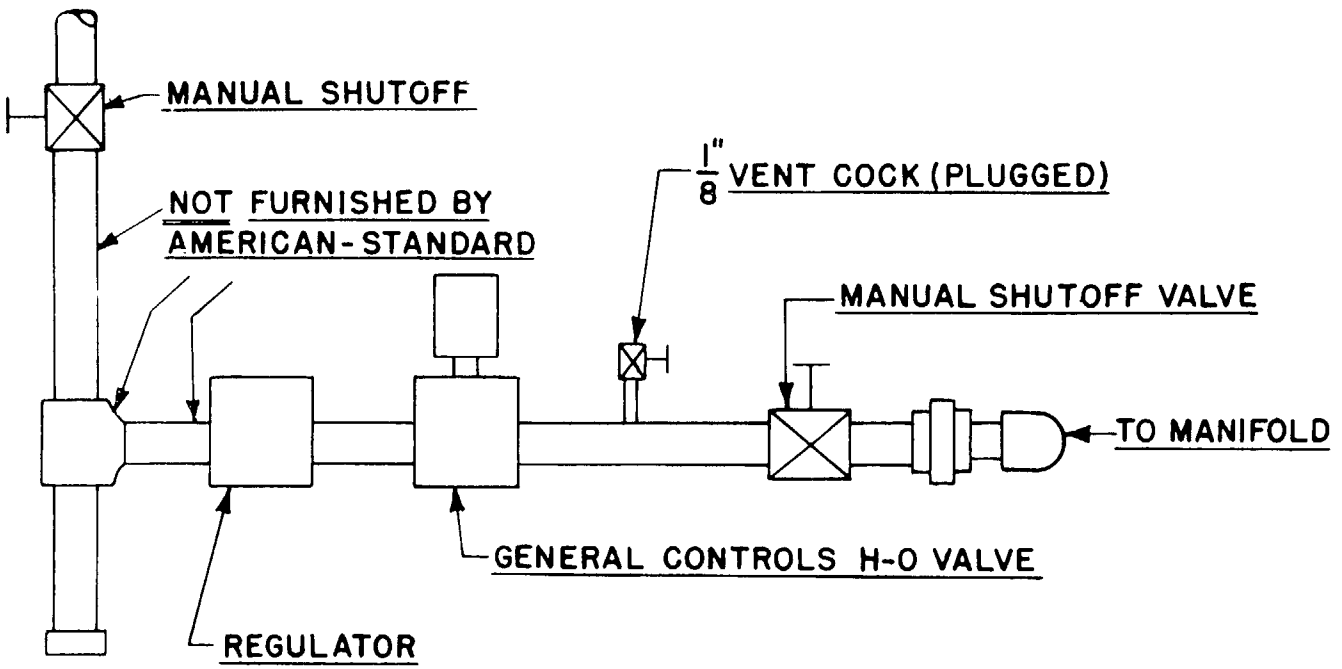


FIGURE 5-5

G-40 AND G-60 GAS BOILERS

CONTROL ARRANGEMENT FACTORY INSURANCE ASSOCIATION

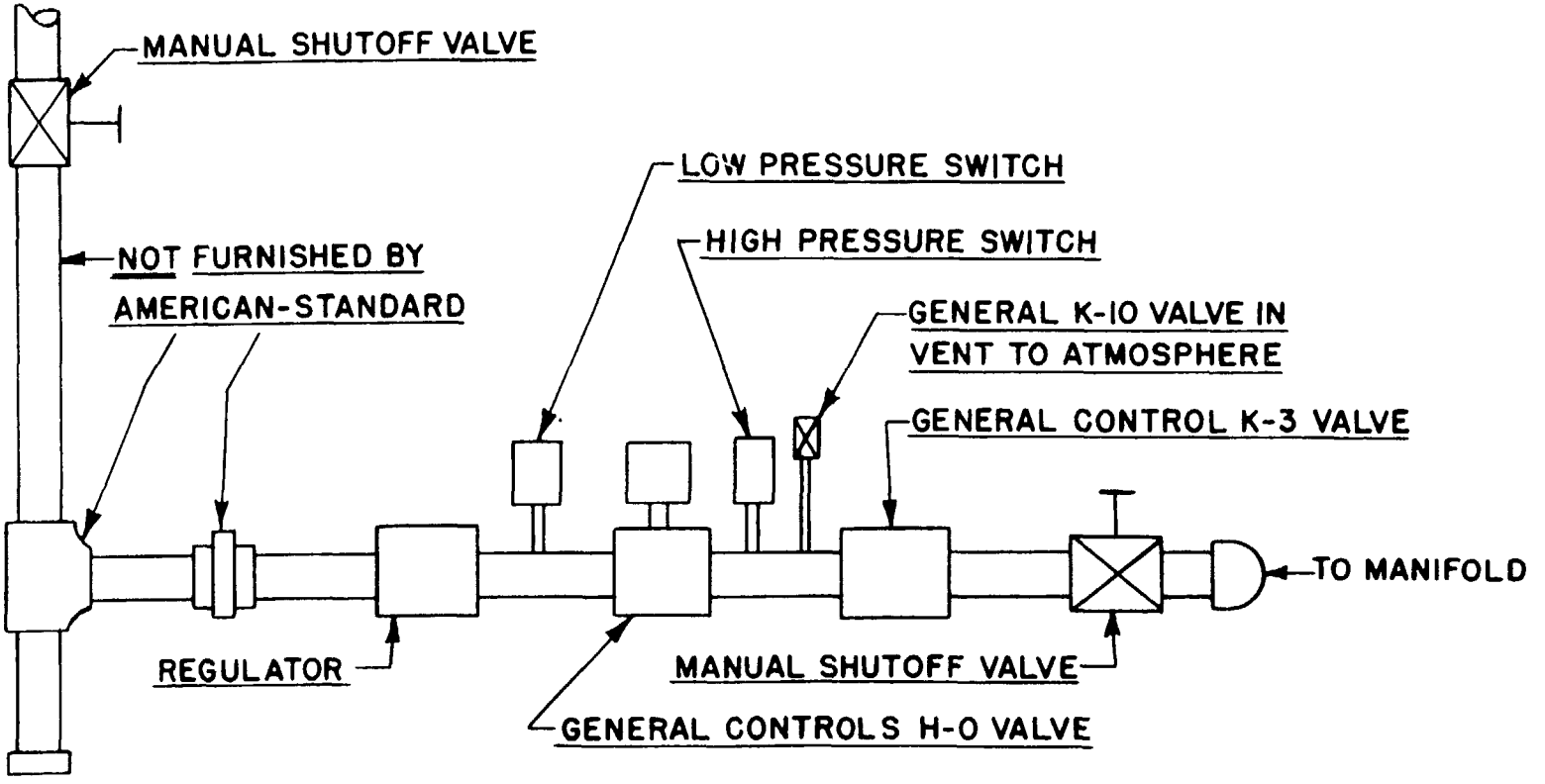


FIGURE 6-6

G-40 AND G-60 GAS BOILERS

HEATING CONTROL SYSTEMS— LARGE BUILDINGS

CHAPTER 7

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CHAPTER 7

HEATING CONTROL SYSTEMS -- LARGE BUILDINGS

The control of large heating systems involve the same principles as the control of small systems.

However, because budgets are larger and savings on larger amounts of fuel become important, more expensive control systems are normally justified.

Instead of the "off-on" control system, continuous but modulated heating is frequently employed. The pneumatic control system as briefly described in Chapter 5 is of the modulating type.

Selection of radiation should be coordinated with the type of heating control. Non-ferrous finned radiation works best with continuous but modulated heat delivery. Cast iron works equally as well with modulated heat input, but is preferred in intermittently heated space because of its heat-holding characteristics.

This chapter will discuss the control of both hot water and steam systems. single zone and multiple zone...continuous and intermittent heat delivery.

Hot Water...Intermittent...Single Zone

The boiler is operated at a fixed water temperature to provide domestic hot water with an immersion control in the boiler water controlling the burner. Heat is delivered intermittently when the space thermostat starts the circulator.

Hot Water...Intermittent...Multiple Zone

The boiler is operated the same as in a single zone system. Heat delivery to each zone may be accomplished as follows:

1. Each zone is served by a separate circulator under the control of the zone thermostat. With this type of control the circuit to each zone has a lift check valve installed to prevent a circulator on one zone creating water circulation in a zone not calling for heat.

2. A single pump can be used for all zones with each zone having a motorized valve under the control of the zone thermostat. On large systems with heavy circulators a by-pass is usually provided between the supply and return mains to permit water circulation through the boiler and circulator when all zone valves are closed. Water flow through the by-pass may be controlled by a throttling hand valve or by a pressure operated valve, which opens automatically with the rise in pump head occurring as the system water flow diminishes.

Hot Water...Continuous...Single Zone

The boiler is operated at a fixed water temperature to provide domestic hot water. An immersion control operates the burner.

Continuous heat delivery to the space is achieved by operating the circulator continuously. A by-pass around the boiler with a three-way mixing valve automatically modulates water temperatures with the severity of the weather. An outdoor bulb controls the three-way valve.

The circulator is controlled by an indoor thermostat set two or three degrees higher than the normal indoor temperature maintained by the outdoor control. Should indoor temperatures, for any reason, rise above the normal control point the thermostat stops the circulator and all heat delivery to the space.

Hot Water...Continuous...Multiple Zone

The boiler is operated the same as in the single zone system.

Three-way automatically controlled mixing valves modulate the water temperature to each zone under the control of an outdoor bulb located to reflect the heat loss of the zone.

A single circulator for the system or individual circulators for each zone may be used.

When zones require no heat it is possible to provide auxiliary switches on the three-way valves to stop the circulator.

On large hot water systems with multiple zones, piping may be designed for using a secondary pumping arrangement.

Steam...Intermittent...Single Zone

There are two basic control arrangements.

The first is simple, inexpensive but somewhat slow in response. The space thermostat operates the burner. Steam pressure is developed to force air from the system and steam into the radiation.

A variation to this method is to use an outdoor thermostat. Here the occupants of the space have no control of space temperature.

The second method requires the maintenance of steam pressure during the heating season. A combination low limit and high limit pressure-stat is employed. The space thermostat operates a motorized valve. This is a more expensive control arrangement but the response to a call for heat is quicker.

An outdoor thermostat may be used with the motorized valve.

Steam...Intermittent...Multi-Zone

In the multi-zone steam system the boiler operates at a fixed pressure with intermittent heat delivery to each zone controlled by motorized valves operated by zone thermostats, either in the space or outdoors. In large two pipe systems each piece of radiation may have orifices installed to provide for uniform steam delivery to each radiator.

Steam...Continuous...Single Zone

In large house and small commercial installations, single zone steam systems accomplish continuous heat input by operating as a vapor system.

With this type of system, steam is maintained continuously in the radiation but under a vacuum that varies with the weather conditions. As the vacuum becomes greater, the steam temperature becomes less and the heat output from the radiation diminishes.

When the space cools, the thermostat starts the burner, increases the pressure and steam temperature -- vacuum becomes less -- and more heat is obtained from the radiation. The normal range of steam temperatures in the vapor system is from 170°F to 215°F. Response to a demand for heat is rapdi and uniform throughout the system.

Steam...Continuous...Multiple Zone

To provide continuous but modulated heat delivery in a multiple zone steam system, a two pipe vacuum is employed. Boilers are operated at pressures above atmosphere.

Thermostatic control of system pressure is accomplished through operation of motorized valves on the steam main and high vacuum pumps on the return mains provide modulated and continuous heat delivery. A number of proprietary systems with the necessary steam specialties have been developed for this type of operation.

Outdoor or indoor thermostats may be used.

All radiation is installed with orifices.

CHAPTER 8

CAST IRON vs. STEEL BOILERS

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CHAPTER 8

CAST IRON vs STEEL BOILERS

There are a number of differences between cast iron and steel boilers which every buyer and seller should understand.

Steel boilers may be designed for greater pressures that are necessary for process steam used in manufacturing and power generation.

Low pressure cast iron and steel boilers for heating are designed and constructed in accordance with the American Society of Mechanical Engineers Code for Low Pressure Heating Boilers.

By this ASME Code, working pressures are limited to 15 psig for steam heating boilers and 160 psig for hot water heating boilers with a maximum temperature limit of 250° F.

American-Standard cast iron steam boilers are therefore limited by the ASME Code to 15 psig working pressure and 250° F temperature.

All American-Standard cast iron water boilers were originally good for 30 psig working pressure. Recently the working pressure has been raised to 50 psig and, on special order, boilers may be obtained for 80 psig. Refer to current catalog information for boilers available at these higher pressures.

Regardless of pressure, hot water heating boilers are limited to 250° F. working temperature.

Within those limits...and for all normal heating requirements...cast iron boilers have a definite edge over steel.

As far as installation is concerned there is no clear-cut advantage of one over the other. Cast iron boilers delivered in sections require installation time to seal and bolt the sections together. Steel boilers are often delivered complete, ready to hook up. However, because of their size they need expensive rigging to handle and jockey them into position.

Steel boilers also suffer from the cost of replacement. In the case of cast-iron, replacement can often be made on a section-by-section basis...whereas with steel, damage to the shell often involves replacement of the entire boiler.

And, when a steel boiler is installed in a new building with no provision for replacement, the operation requires the demolition and reconstruction of a considerable wall area.

The biggest advantage in cast iron is inherent in the metal itself. Cast iron is much less susceptible to corrosion than steel to begin with...and corrosion, when it does start, has a much more serious effect on steel.

Rust will attack steel much quicker than cast iron and, when it does, it tends to bore through the metal in a small area. This has the effect of making the metal thinner and therefore weaker.

In the case of cast-iron, it resists rusting longer and rust, when it does start, spreads across the surface and does not eat into the iron.

In addition to giving much longer life, cast iron has one other big, inherent advantage over steel...it can be poured into molds, while steel must be formed and shaped in dies and presses, with various parts joined by welding.

Internal tubing on a steel boiler, for instance, must be rolled into the plates (tube sheets) supporting them much like the copper gas line is flared at the connection to the carburetor in your car.

This joint represents a weak point in the steel boiler and, as weak points often do, is liable to let go just when it is being pushed hard in severe weather when breakdown will cause the most inconvenience.

Another advantage of molding cast-iron is that piping connections can be more flexible. In the case of steel, the practicalities of fabrication rather than desire govern the placement of connections.

Cast-iron boilers are expandable within limits merely through adding sections. Many communities, for instance, are adding wings to existing schools. Where cast iron boilers are used, adding sections may take care of the new wing. If the expansion goes beyond the capabilities of a steel boiler, the entire boiler must be replaced, or a second boiler added...if there is room enough.

Cast-iron boilers take up less space both in square feet of floor space and in total cubic feet, for the same capacity. This, too, is a result of pouring iron into the most ideal shape.

Finally, because cast iron uses a number of smaller combustion chambers, rather than one huge flame, controls can be simpler and less costly. And simplicity means lower maintenance cost and less down time.

To Sum Up

At pressures above 15 p. s. i. g. steam and 80 p. s. i. g. water...only steel can do the job.

Installation cost is a stand-off.

Cast Iron Advantages

1. Less susceptible to corrosion.
2. Longer internal-connection life.
3. More flexible piping connections.
4. Readily expandable to handle larger loads.
5. Easier, less expensive replacement.
6. Smaller space requirements.
7. Simpler, less expensive controls.

CHAPTER 9

STOKER-FIRED BOILERS

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CHAPTER 9

STOKER-FIRED BOILERS

A mechanical stoker is a device to provide automatic fuel feed in a coal furnace, to provide a supply of air under automatic control for burning the fuel, and in some cases, to provide an automatic means of removing ash and other refuse of combustion.

Mechanical stokers burn coal much more efficiently than hand-fired furnaces because the fuel feed is more uniform and more perfectly distributed and because it provides more positive control of the oxygen necessary to combustion.

Stokers may be roughly separated into four classes:

Class 1: burning 10 to 100 pounds of coal an hour are used primarily for home heating.

Class 2: burning 100 to 300 pounds of coal an hour are usually underfeed motor-driven screw-type stokers and are used extensively in apartment houses, schools, hotels and motels.

Class 3: burning 300 to 1,200 pounds an hour and...

Class 4: burning over 1,200 pounds an hour.

With American-Standard cast-iron boilers it is unnecessary to go beyond the Class 2 stoker.

Coal firing rates for American-Standard boilers are based upon coal having a calorific value of 13,000 btu per pound.

Two measurements are sometimes confused: headroom and combustion space.

Headroom is the distance between the top of the retort (the stoker basket that holds the burning fuel) and the boiler crownsheet.

The minimum headroom should be governed by this table for the Severn, Exbrook, No. 3 Redflash and A-7 boilers.

Firing Rate pounds per hour	Minimum Distance
0- 20	18"
21- 30	20"
31- 40	24"
40-100	28"
101-200	30"
201-300	36"

This headroom is necessary when burning bituminous coal because of the tendency of coke "cones" to grow out of the fire bed, like stalagmites in underground caves. If they build high enough to touch the crownsheet, they topple over, carrying with them the coke that has formed at the level of the fire bed.

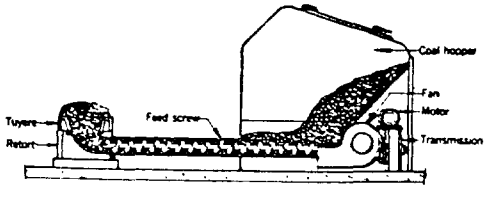
This results in a thinning of the fire bed and the exposure of white hot coals. This thinner portion of the bed offers less resistance to the air-flow through the fuel bed, creating an intense blast condition. The intense local radiant heat generated is sufficient to cause breakage of the crown-sheet directly over it.

American-Standard boilers should not be stoker fired at fuel burning rates greater than those specified. To do so aggravates coke cone conditions and creates risk of boiler damage.

Combustion space is the number of cubic feet required for complete combustion of the fuel. Combustion space must be large enough to avoid too high a btu release per cubic foot per hour.

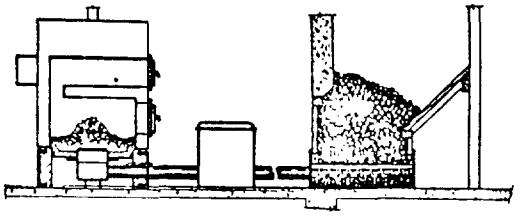
If too high a release for the combustion space is attempted we are faced with what is known as a bottled heating condition. This excessive heat for the space results in excessively hot front doors, warping boiler fronts, liquifying refractory brick and damage to the walls of the stoker retort.

The firing rates of stoker fired American-Standard boilers are satisfied by either Class 1 or Class 2 stokers. In most instances, the stokers are of the underfeed type using a screw to convey the coal from a hopper or bin to the stoker retort.



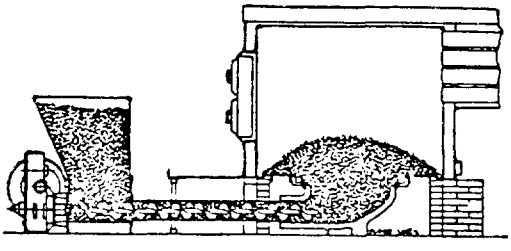
A hopper type Class 1 underfeed stoker, with a round retort.

FIGURE 9-1



A Class 1 bin feed type of underfeed stoker with round retort.

FIGURE 9-2



A Class 2 hopper type of underfeed stoker with rectangular retort.

FIGURE 9-3

These stokers may burn either anthracite coal (pea, buckwheat, rice or barley size) or bituminous coal, usually pea size.

When burning anthracite there is less tendency for the ash to clinker. As the coal burns the dry ash spills over the edge of the retort collecting in a well or on a hearth from which it may be removed manually or automatically.

The burning of bituminous coal presents a different problem. The ash collects principally on the hearth around the retort. As the ash layer gets thicker the heat from the fire melts it and causes a clinker to form. This clinker is removed manually with clinker tongs designed for the purpose. While a few Class 1 or Class 2 underfeed stokers are designed for automatic ash removal the major portion of them require manual removal of the clinkered ash.

For best efficiency the quantity of air fed to the fuel by the fan must be kept in the proper proportion to the rate of coal feed. This may be done by setting a damper in the air duct but the wide variance in fuel bed conditions (thickness and porosity) make this type of control unsatisfactory. Most modern stokers are provided with automatic regulation of the air.

A draft regulator in the smoke pipe is used to control the over-fire draft within reasonable limits for best results.

As with other fuel burning equipment, there must be an ample supply of air made available to the stoker if efficient combustion is to be obtained. A stoker burning 100 pounds per hour of coal will require an air supply of 1,500 pounds per hour or approximately 375 cubic feet per minute. Usually a direct opening between the boiler room and outdoors is needed.

CHAPTER 10

OIL-FIRED BOILERS

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CHAPTER 10

OIL FIRED BOILERS

Originally, crude oil was put through a straight run distillation process to separate it into various petroleum products; naptha, kerosene, fuel oil, gasoline and lubricating oils. The quantity of each from a barrel of crude oil depended entirely on the field producing the crude. Because of these fixed quantities a high demand for one of the products -- gasoline for instance -- would result in over production of the others.

To overcome this problem the oil companies developed a thermal-cracking process by which the natural arrangement of the carbon atoms and hydrogen atoms in the oil is changed. This process made possible the production of varying quantities of products from the same crude oil. For example, larger percentages of gasoline could be produced in the summer. During the heating season the production of fuel oil would be increased and less gasoline produced.

In the late 1930's the catalytic cracking process was developed. It has almost replaced the thermal cracking process since the same results are obtained, but at lower temperatures and pressures.

The development of cat-cracking gave the heating industry some problems. While cat-cracked #2 heating oil can be produced chemically identical to distilled #2 heating oil, it has different burning characteristics...it causes smoking, pulsing and noise in burners.

This trouble, however, has been minimizied because burners are now designed to handle both distilled and cat-cracked oils or blends.

Oil is distributed in several grades.

One of the lightest oils (No. 2) is almost always used in residential burners. Number 6 is the heaviest and is used primarily in the larger boilers. The heavier the oil, the higher the btu value.

Common terminology refers to light, medium and heavy oils. Actually the numbers 1, 2, 4, 5 and 6 as grades of heating oil and 10, 20, 30 as grades of oil in your car's crankcase indicate the relative viscosity.

Viscosity is a measurement of the resistance to flow in a substance. The higher the viscosity, the higher the resistance to flow. The heavier the oil, the higher the viscosity.

This table shows the approximate weight and heating values of standard grades of fuel oil.

Commercial Standard No.	Weight lbs./gal.	Heat Value btu/gal. (000's)
1	6.951-6.675	137.0-132.0
2	7.296-6.870	141.8-135.8
4	8.212-7.206	153.3-140.6
5	8.488-7.778	155.7-148.1
6	8.571-7.882	157.3-149.4

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The basic or standard viscosity of any oil is affected by temperature. As temperature falls, the oil becomes more viscous...develops more resistance to flow, becomes more sluggish.

Obviously, above certain levels of viscosity, flow to the burner nozzle and, much more importantly, nozzle ability to atomize the oil would suffer.

Because of their inherently low viscosity, temperature presents no problems in the case of the two lighter grades.

In very severe weather, however, No. 4 oil must be heated to at least 0°F to burn efficiently.

No. 5 must be heated to at least 60°F -- No. 6 oil to 120°F.

There are three basic types of burners:

1) High pressure atomizing
2) Low pressure atomizing
3) Rotary Cup

The "High pressure atomizing" is the most common, least expensive and simplest in design. It uses the lightest of oils. It is supplied in capacities up to about 30 gallons per hour, and makes no provision for modulating oil flow. The oil is forced through the nozzle under heavy pressure (from 75 to 140 psig) so that it breaks up, or atomizes, as it leaves the nozzle. American-Standard burners are set at the factory for 100 psig.

The "low pressure atomizing" burner (from 2 - 7 psig) is made in about the same oil-burning capacity as the pressure atomizing type, but because it can handle heavier oils (up to No. 5) with greater btu values it can supply more heat at any given firing rate in gallons per hour.

Here the oil and air are mixed before they reach the nozzle. Because the oil is emulsified before it reaches the nozzle, this type of burner requires a larger nozzle which does not clog as readily.

The "rotary cup" burner handles up to No. 6 fuel oil in capacities up to 150 gallons per hour...with the flow modulated as required by the load.

A rotary cup, in the form of a cone, replaces the nozzle. Spinning at 3600 RPM, the cup throws off a thin film of oil mixed with small amounts of air, called primary air and equal to 10 or 15% of the total necessary for complete combustion.

The additional 85% to 90% of air, called secondary air, is brought through a secondary air door by either natural or forced draft.

To review the four steps in oil-combustion, after the combustion chamber is warmed up:

1) The oil is atomized
2) It is vaporized or gasified by the radiant heat of the combustion chamber
3) It combines with oxygen in the air
4) It burns.

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The efficiency of burning is largely determined by the efficiency of vaporization. This factor makes it necessary to concentrate the radiant heat of the walls of the combustion chamber on the oil spray as it leaves the nozzle.

For this reason, nozzles are designed to give a variety of spray patterns so that the spray can be tailor-made to fit the dimensions of the combustion chamber. Wide angle nozzles deliver a "sunflower" flame in short, wide combustion chambers and an infinite variety of narrow angle nozzles deliver longer, narrower "horsetail" flames depending on the ratio of length to width in longer, narrower combustion chambers.

Oil spray should not be allowed to hit either walls or floor of the combustion chamber for two reasons:

1. It builds up carbon on the walls and floor of the chamber which shortens its life

2. It creates excessive smoke which deposits heavy soot layers on flue surfaces, lowering heat transfer efficiency and necessitating frequent cleaning.

The short, wide "sunflower" flame is quieter when small quantities of oil are used.

In intermittently-fired boilers it is essential that combustion chamber walls heat up rapidly so as to reach maximum efficiency of vaporization as soon as possible in each firing cycle.

Combustion chamber walls should be high enough to protect the lower portion of the boiler's water legs. Should an accumulation of sludge and scale exist in the water legs, restriction of normal water circulation may occur and dissipation of heat at that point restricted. Boiler damage could result.

The following tables show typical combustion chamber sizes for the various burning rates in gallons-per-hour of low and high pressure atomizing "gun-type" burners.

COMBUSTION CHAMBERS

DIMENSION TABLE: HIGH PRESSURE ATOMIZING BURNERS

Burning Rate GPH	Length Inches	Width Inches	Height Inches	Flame Center to Floor	Floor Area Sq.In.
4	21	17	17	8	288
5	23	18	18	8	396
6	24	20	18	9	480
7	26	22	19	10	572
8	28	23	19	11	644
10	33	24	19	12	792
12	36	26	20	13	936
14	39	28	21	14	1092
16	42	30	22	15	1260
18	44	32	23	16	1408
20	46	34	24	17	1564
22	48	34	26	17	1612
24	50	36	28	18	1800
26	52	36	30	18	1872
28	54	38	32	19	2052
30	56	40	34	19	2240

DIMENSION TABLE: LOW PRESSURE ATOMIZING BURNERS

Burning Rate GPH	Length Inches		Width Inches		Height Inches		Flame Center to Floor		Floor Area Sq. Inches	
	Rec.	Min.	Rec.	Min.	Rec.	Min.	Rec.	Min.	Rec.	Min.
Up to 8	34	32	21	19	18	16	10	9	715	640
10	38	36	22	21	19	17	11	10	840	755
12	45	40	24	22	20	18	12	10	1080	975
14	48	44	26	24	22	20	12	10	1250	1125
16	54	48	28	26	22	20	13	11	1525	1380
16-20	56	50	32	30	24	21	15	12	1780	1620
20-25	60	54	34	30	30	26	15	12	2040	1840

This table shows both recommended and minimum dimensions. If the minimum width is used, the recommended length should be used. In all cases length has more effect and, therefore, more importance than width.

DIMENSION TABLE: ROTARY CUP BURNERS

Burning Rate GPH	Length Inches	Width Inches	Height Inches	Flame Center to Floor	Floor Area
9	39	20	30	10	780
16	49	24	30	10	1176
25	57	27	32	12	1539
35	77	39	37	15	3003
50	90	45	38	18	4050
69	105	54	38	22	5670
76	112	60	38	26	6720
100	118	65	45	29	7670
125	130	70	46	32	9100

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Large oil burning systems commonly use automatic draft regulators to produce a draft pressure over the fire of .02 to .025 inches of water.

All oil burners should be installed in American-Standard boilers according to the specifications of the burner manufacturer.

INDIRECT DOMESTIC HOT WATER HEATERS

CHAPTER 11

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CHAPTER 11

INDIRECT DOMESTIC HOT WATER HEATERS

The necessity for heating water, in either a hot-water or a steam heating system, naturally enough, gave somebody ideas of using the same boiler to provide domestic hot water.

In the hot-water boiler the immersion control actuates the burner to maintain the desired boiler water temperature, summer and winter.

Steam heating systems have a different problem. In the heating season, a thermostat or pressure-stat is the normal burner control. But in the non-heating season it is wasteful of fuel to maintain a steam pressure just to heat domestic hot water. For the time when steam is not required, an immersion control is installed below the water line of the steam boiler.

By keeping boiler water at or near a specified temperature, domestic hot water is provided even when the space thermostat is not operating.

This means that one unit can provide both heat and hot water, eliminating the need of extra expense for separate water storage and water heating equipment..saving both space, and , during the heating season, fuel.

Four types of domestic hot water heaters are used with boilers :

1. Internal Tankless (instantaneous coil type)
2. External Tankless (instantaneous coil type)
3. Internal Coil with storage tank
4. External Coil with storage tank

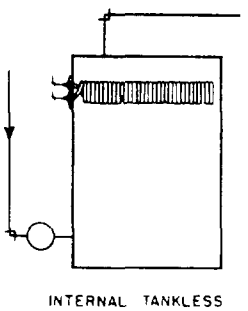


FIGURE 11-1

The internal tankless heater uses a finned copper coil submerged in the boiler water. To extract the most heat, the coil is fixed near the top of a hot water boiler and just below the surface in a steam boiler, where boiler water temperature is highest.

The fins are usually an integral part of the coil and never loosen. They are scientifically sized to minimize the amount of coil necessary to develop water heater capacity.

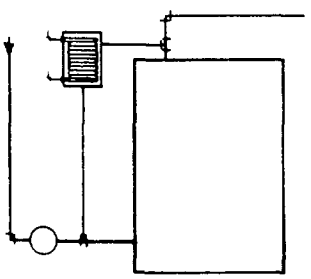
The coil is designed to give maximum length without, however, restricting the flow of boiler water over and around the coil.

Controls must maintain boiler water at the proper temperature during the non-heating season...usually between 180° F and 200° F.

To insure quick response to a demand for hot water, the operating immersion control in the boiler water is usually located just under the cold water inlet to the heater, so it will sense the temperature drop almost as soon as the domestic hot water draw occurs. Immediately the main burner is switched on to keep the boiler water at the proper temperature.

With American-Standard coil type heaters to provide maximum sensitivity, the control is placed inside the loops of the coil where boiler water cools the quickest when a hot water demand draws cold water into the coil.

During the heating season, controls in a hot water system lock out the circulator pump until any call for domestic hot water is satisfied. The call for hot water is usually completed in a few minutes and has no noticeable effect on heating the space.

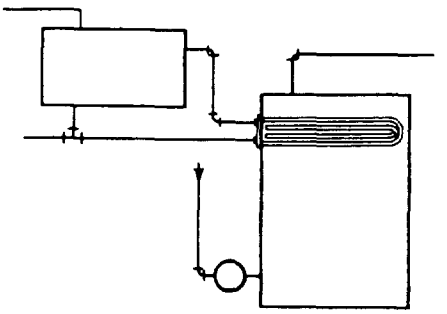


EXTERNAL TANKLESS

The external tankless heater carries its coil in a separate small "tank" of its own. Hot water from the main boiler is circulated through this coil tank, usually by gravity. The operating control is again placed adjacent to the coil inlet.

FIGURE 11-2

In the case of steam, the coil and its tank are dropped down so that a horizontal upper pipe enters the boiler just below the surface of the boiler water.



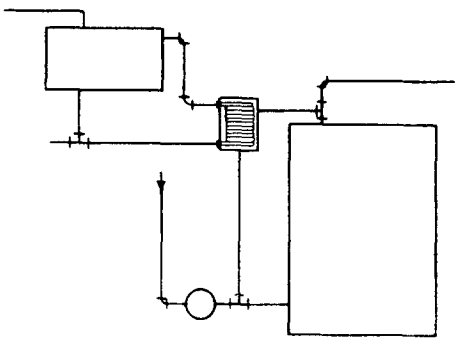
INTERNAL TANK TYPE

FIGURE 11-3

The internal tank type heater is a specially designed internal heater with a hot water storage tank added so that an initial supply of hot water is always available. Instead of a long lengths of small size copper tubing, it uses a relatively short lengths of large size tubing to allow for flow by gravity.

The hot water in the coil and tank is constantly circulated by gravity, out of the upper outlet of the coil to the top of the storage tank, then out of the bottom of the tank, and back to the lower coil inlet.

In some cases a mixing valve is installed joining a cold water line to the output piping of the storage tank where the requirement is for water cooler than 180°F.



EXTERNAL TANK TYPE

FIGURE 11-4

The coil of the external tank type heater is designed in the same manner as the internal tank type. It is used when the boiler design does not provide for its installation in the boiler.

Again hot water is circulated through coil and storage tank by gravity and again, in the case of a steam boiler, the heater is lowered to allow for piping just below the surface of the boiler water.

Tankless Heater Performance - I-W-H Rated

After extensive research and investigation on the rating of tankless heaters, the Institute of Boiler and Radiator Manufacturers announced in 1956 the Indirect Water Heater (I-W-H) Code. This code stipulates that the tankless heater must be tested and rated in the boiler in which it is installed.

The I-W-H Code takes into account all the variables affecting tankless heater performance: water temperature rise through the heating coil (40°F-140°F); water flow through this heater coil (G.P.M.); boiler water temperature; water content of the boiler and boiler-burner input. By testing the tankless heaters installed in the particular boiler with which it is used, the code assures accurate, dependable ratings.

Not all tankless water heaters are rated according to the I-W-H Code. Some domestic water heaters of the tankless type are rated by the manufacturer of the water heater in terms of gallons per minute of water draw at 100°F temperature rise (40°F - 140°F)...and on the basis of the heating surface of the coil that is in contact with the boiler water, plus the temperature of the boiler water.

Usually, these ratings are obtained by laboratory tests conducted with the tankless heater installed in a large hot water tank where boiler water temperatures are simulated.

Advantages of Indirect Water Heaters

1. Initial investment costs are lower for the boiler with tankless heater.

2. Servicing and maintenance is reduced.

3. Life expectancy of summer-winter installations is greater for boilers which are not shut down in summer -- corrosion and pilot problems during summer months are minimized.

4. Space requirements are reduced to a minimum.

5. The copper tubing in tankless water heaters is highly resistant to corrosion.

6. In damp climates, summer-winter installations help keep basements dry.

7. Domestic hot water is generated instantaneously.

NOTE: Where excessive lime exists in the water, tankless type water heaters are not recommended unless provision is made for regular cleaning. Tank type heaters are recommended in hard water areas.

HOW THE STEAM SYSTEM WORKS

CHAPTER 12

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CHAPTER 12

HOW THE STEAM SYSTEM WORKS

Basically, any heating system, like any electrical circuit, has three components:

1. The generator or boiler.
2. Circuitry or piping.
3. A load...the radiation.

Providing each of the three components is in good working order and connected to each other with tight joints, a steam heating system needs only the removal of air and free return of condensate to function properly.

System air is vented to the atmosphere and condensate is returned to the boiler to maintain the proper boiler water level, and to go through the next cycle of boiling to steam and condensing in the radiation as it heats the space.

Air and condensate are removed from the system by a series of vents and traps covered in the next chapter (13).

A steam boiler is not completely filled with water. When heat is applied to the water it boils and creates steam. As discussed earlier, it will boil at varying temperatures depending on the vapor pressure on the surface of the water.

At one pound gage pressure, one pound of water (1/62nd of a cubic foot) will create 25 cubic feet of steam. This represents an approximate 1,500 times increase in volume.

This tremendous increase in volume builds up a pressure greater than the pressure in the piping and radiation. Since steam flows in the direction of a lower pressure it flows through the piping to the radiation.

The piping normally, is insulated to prevent excessive condensation of the steam and loss of heat until the steam reaches the radiation.

A steam system transfers heat from boiler to radiation in the form of latent enthalpy of vaporization. A hot water system transfers heat in the form of sensible enthalpy of the water.

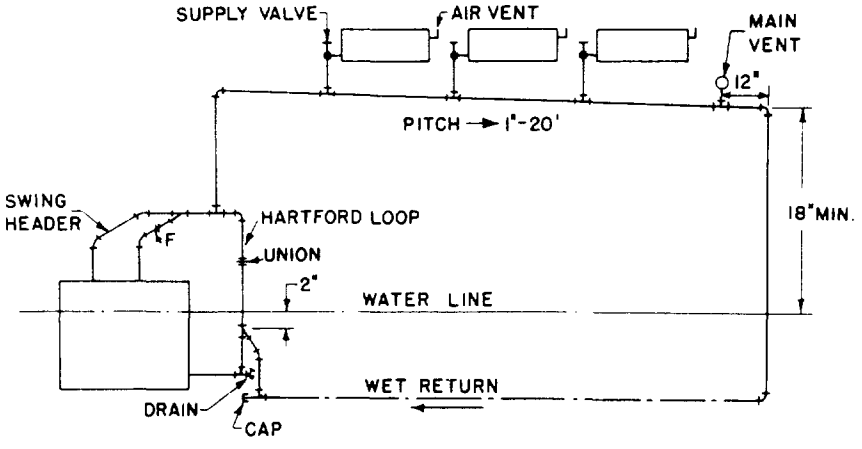


FIGURE 12-1

In Figure 12-1 the diagram shows a simple one-pipe, up-feed system. The supply main is pitched downward to facilitate the gravity flow of condensate.

Some definitions: The supply main carries steam to all radiators. After the last piece of radiation the piping back to the boiler is called the return. At the end of the pitched supply main a drop-leg carries condensate back to the boiler room level. Piping from drop-leg to boiler is called a wet return because it is below the boiler water level and completely filled with water.

Actually the water level in the drop-leg is always a little higher than in the boiler, during firing cycles. Resistance to flow in the supply main acts to decrease the pressure on the water surface in the drop-leg which causes the water level to rise. And still more height of water in the drop-leg is necessary to overcome the resistance in the return. For gravity return of condensate, the minimum clearance between the boiler and the water level and the end of the supply main is 18"

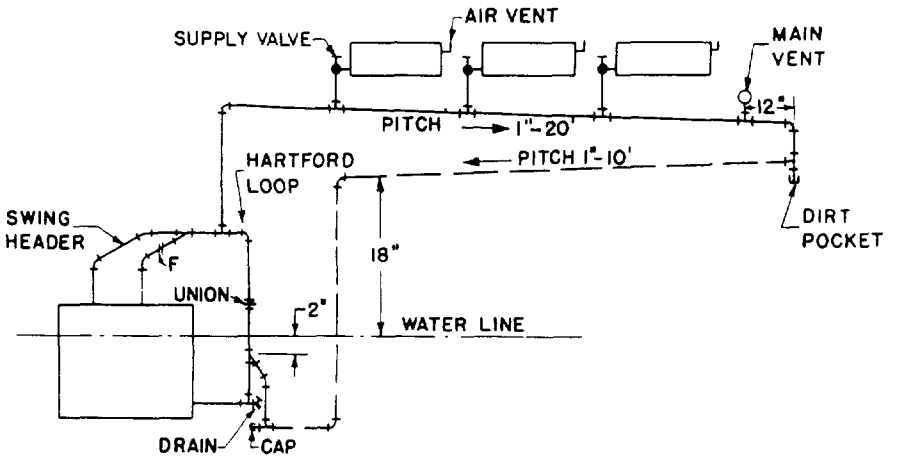


FIGURE 12-2

Frequently in one-pipe systems it may not be practical to run a wet return from the end of the steam main back to the boiler. In such instances a dry return is installed near the ceiling. As shown in Figure 12-2 with the drop-leg located near the boiler.

The radiator runout creates a possible noise problem if not properly installed since the condensate must flow back through the runout against the flow of the steam. The noise problem is most acute during the heating-up period. Cold radiators condense the steam very rapidly. This means greater steam velocities flowing against greater quantities of condensate.

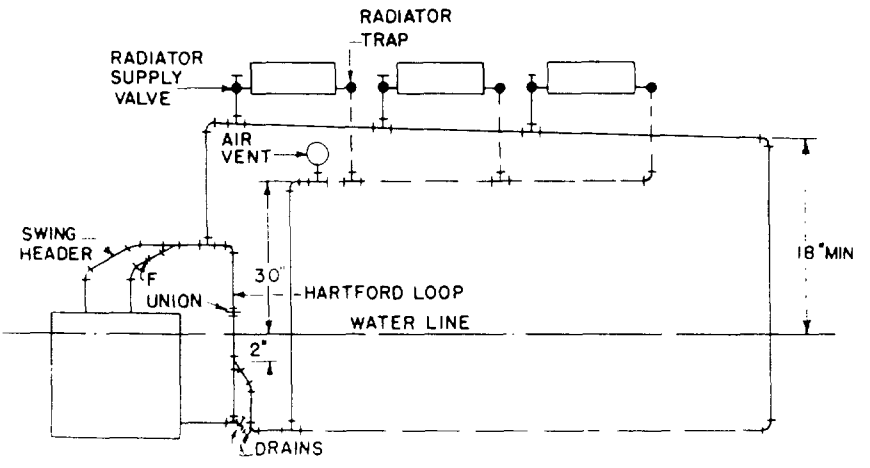


FIGURE 12-3

A quieter system, though more expensive, is the two-pipe up-feed layout. Figure 12-3.

Here the condensate from the radiation is carried out the other end of the radiator and flows through a separate condensate return back to the boiler. Condensate and steam, then, flow in the same direction, eliminating water hammer.

Note that the return pipe is above boiler water level and is not completely filled with water. It is, therefore, called a dry return.

The two pipe system uses a trap at the outlet end of each radiator to allow air and condensate to escape but holds steam in the radiator. (see Chapter 13).

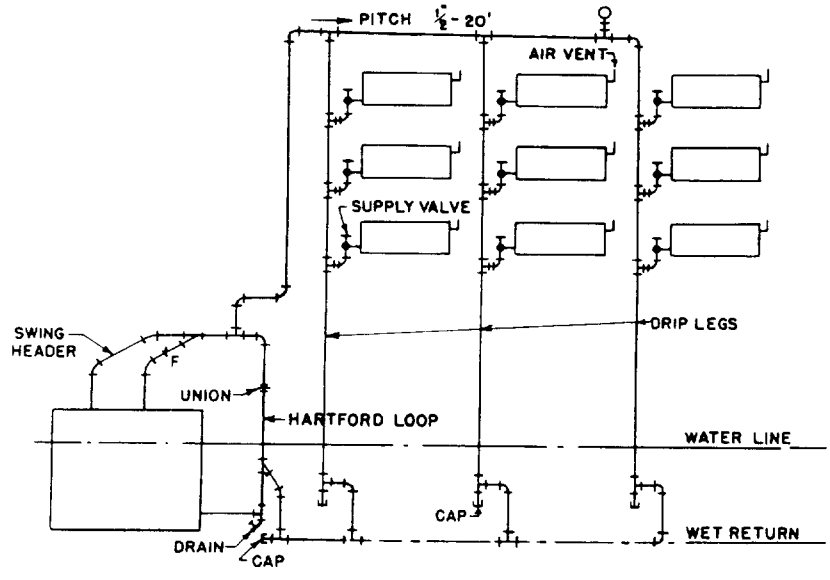


FIGURE 12-4

Downfeed -- Figure 12-4 -- where the steam main is above the radiation, are used on larger installations, primarily to eliminate counterflow of steam and condensate in one-pipe systems and because of building geography in two-pipe systems.

The dripping of downfeed risers also provides drier steam which puts a smaller load on condensate traps in the radiation.

However, the upfeed system, if dripped properly, can do as good a job as the more costly downfeed system.

All four of these piping arrangements are possible in a gravity system ...where the condensate is returned to the boiler by gravity alone.

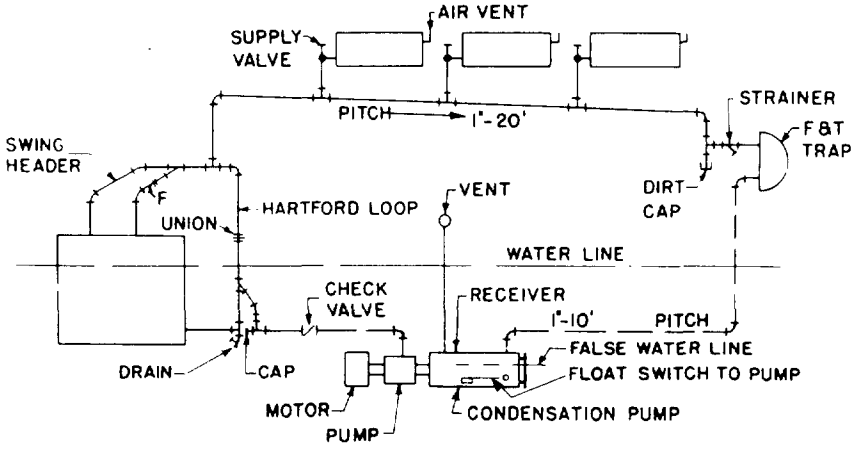


FIGURE 12-5

It is sometimes necessary, as shown in Figure 12-5, to install a condensate pump to pump condensate back to the boiler:

1) When radiation is below the water level on the boiler.

2) When the system develops great quantities of condensate... e.g. a system using unit heaters (see Chapter 29) where fans blowing cold air over the steam coil condense it very rapidly, particularly at the beginning of the daily heating cycle.

3) When the geography of the building defeats easy gravity flow.
   (a) Radiation considerably removed from the boiler when it is uneconomical to enlarge pipe size sufficiently, and there is not room enough to pitch the pipe severely.
   (b) When the condensate return must be high to avoid doorways, windows, etc., and also to keep floor-space free.

4) Or any combination of these.

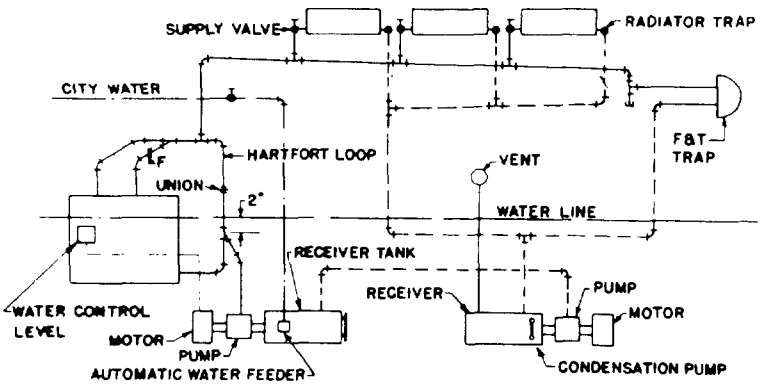


FIGURE 12-6

In large steam systems it is usually necessary to have a receiver tank to hold extra water for the system. This tank receives the condensate from the condensation or vacuum pumps in the system. See Figure 12-6. The boiler feed pump draws water from the receiver and delivers it to the boiler through a feed water regulator.

When the condensate in the receiver is insufficient to satisfy the boiler demand, a float operated valve automatically adds make-up water to the tank.

A receiver must be used when the boiler has insufficient water holding capacity to fill the piping and the radiation with steam.

To repeat: Only two things are essential to the proper functioning of a steam system:

1. Removal of air

2. Removal of condensate

This chapter has discussed four different piping arrangements for steam heating systems. The next three chapters, covering accessories piping, will occasionally refer to system pressure.

The four different kinds of pressure systems, above atmospheric pressure, subatmospheric, vapor and vacuum systems will be covered in detail in Chapters 16 and 17.

STEAM ACCESSORIES

CHAPTER 13

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CHAPTER 13

STEAM ACCESSORIES

A full range of steam accessories is necessary in all but the smallest of domestic systems.

Accessories accomplish just two things:

1. Removal of air.
2. Easy return of condensate.

By accomplishing these two things, they assure proper transmission of steam through the system.

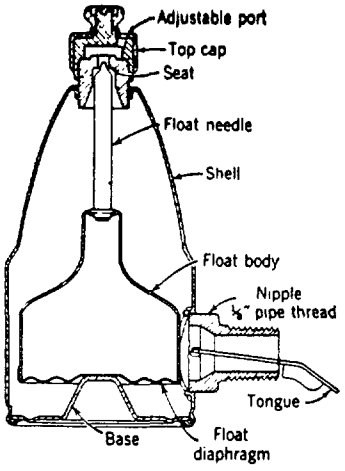


FIGURE 13-1

Radiation Air Vent

This is a typical air vent in one pipe low pressure systems. It is located as low as practicable on the opposite end of the radiator from the supply.

The float body, with a flexible diaphragm base, has a volatile liquid charge which vaporizes easily under the steam temperature.

At the beginning of any heating cycle, as steam enters the radiator, it pushes the air ahead of it. The air is vented through the port.

When the steam has filled the entire radiator, having pushed all the air through the vent, it also fills the shell of the air vent. The liquid in the float body vaporizes, the diaphragm base expands, pushes against the base, raises the float body and the float needle which seats in the top of the chamber preventing the steam from escaping.

With the burner off, the steam condenses, the vapor within the float body cools and condenses, the diaphragm collapses and the float needle is withdrawn from its seat, allowing air to re-enter the radiation.

The air vent shown has an adjustable port to regulate the venting rate ...an important feature when several radiators in the same enclosed space do not heat up uniformly. By adjusting the venting rate they can all be brought to full steam capacity by approximately the same time, providing a better balanced system.

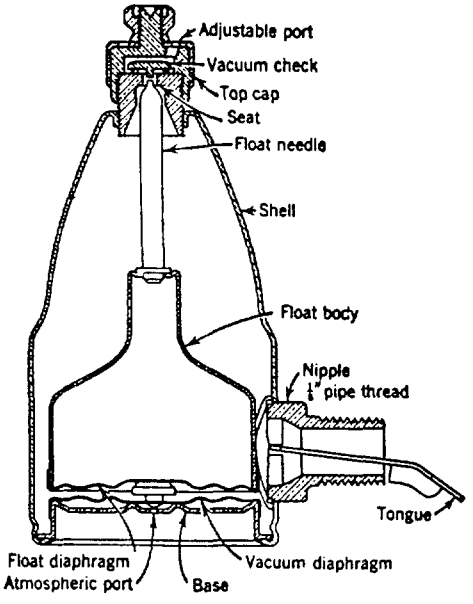


FIGURE 13-2

Vacuum Vent

The vacuum vent is very similar to the air vent (13-1). The basic difference is the vacuum check valve just above the port.

When the float needle is withdrawn from its seat as the steam condenses, this check valve closes, preventing re-entry of air into the system.

These valves are used on one pipe vapor systems...one on each radiator.

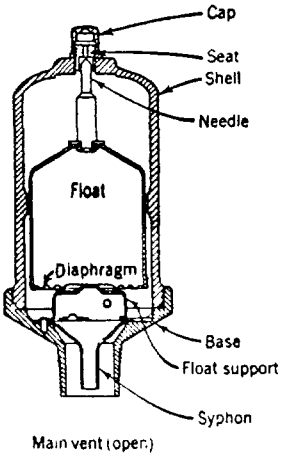


FIGURE 13-3

Main Air Vent

This larger capacity air vent is used at the end of steam mains and dry returns -- about 12 inches back from the drop leg.

Its operation is identical to that of the radiator valve described in Figure 13-1. It is used on pressure systems.

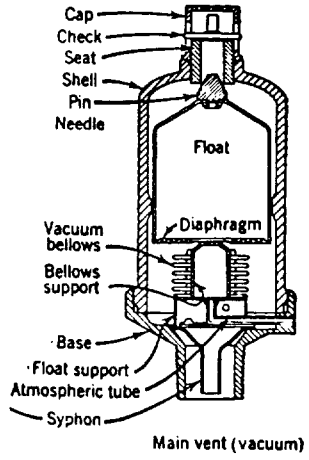
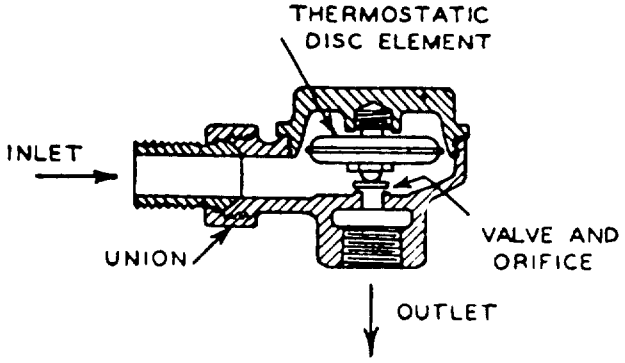


FIGURE 13-4

The vacuum vent valve illustrated in Figure 13-4 is applicable to mains and dry returns in vapor steam systems.

It vents quantities of air easily and quickly. The check valve prevents air from re-entering the piping when the system cools down.



Disc Type
FIGURE 13-5

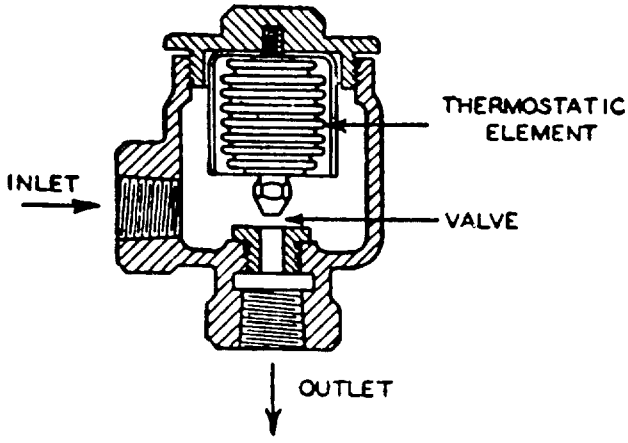
Thermal Trap

The most efficient air vent location in steam radiation is at the bottom of the radiator. Unfortunately, in one-pipe systems it is impractical to put air vents so close to the floor, since they may be damaged.

Two-pipe steam systems are more advantageous. The thermal trap which is located at the radiator discharge passes both air and condensate.

Thermal traps are used on all two pipe systems; pressure vapor and vacuum.

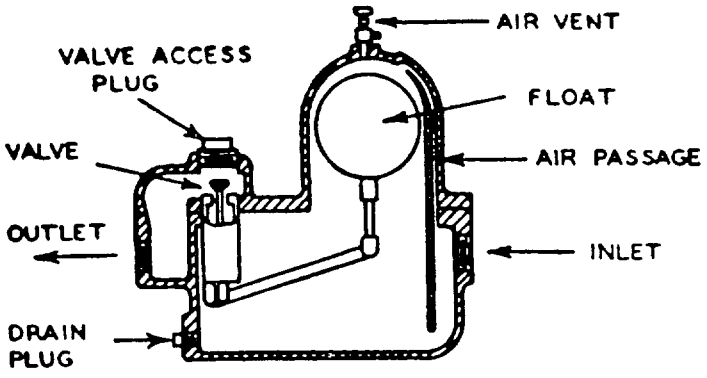
There are two common types: disc and bellows. In each the thermostatic element is charged with a volatile liquid.



Bellows Type
FIGURE 13-6

Air and condensate enter the thermostatic trap at the connection marked "inlet". If the condensate is cold, the thermostatic element remains contracted and the air and condensate pass through the orifice to the return main. As the condensate gets hot the pressure generated by the volatile liquid expands the bellows and closes the port. The trap will not re-open until the condensate cools. At this time the cycle of performance is repeated.

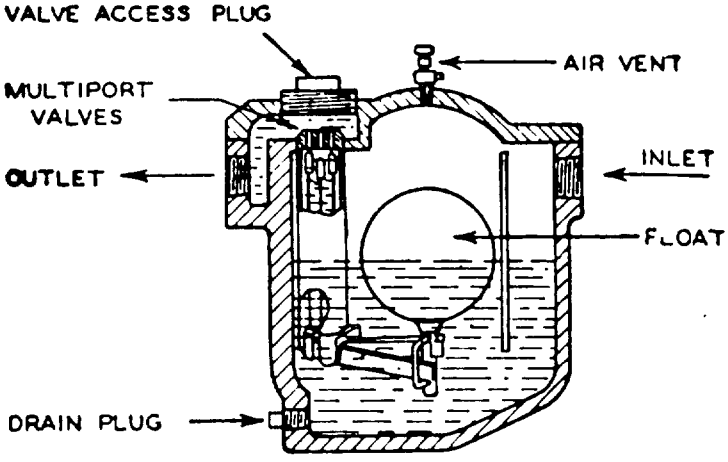
Exactly the same result is achieved with the bellows type trap - Figure 13-6



Single Port
FIGURE 13-7

Float Trap

As condensate builds up in the trap, the float rises, opening the outlet valve.



Multiport
FIGURE 13-8

The steam in the chamber forces condensate out of the trap allowing the float to fall and close the valve, before any steam can escape.

The multiport float trap is indicated where the amount of condensate to be handled varies.

The float, depending on its height, will open one or more ports.

These float traps are used for dripping heels of up-feed systems varying in pressure from vacuum to 200 psig.

In the case of low pressure systems (vacuum - 15 psig.), the trap should be equipped with a thermostatic air vent.

Float traps are better suited to systems where condensate removal is continuous. They should be well-made. False economy in float trap purchases will result in steam leakage and sticking of moving parts.

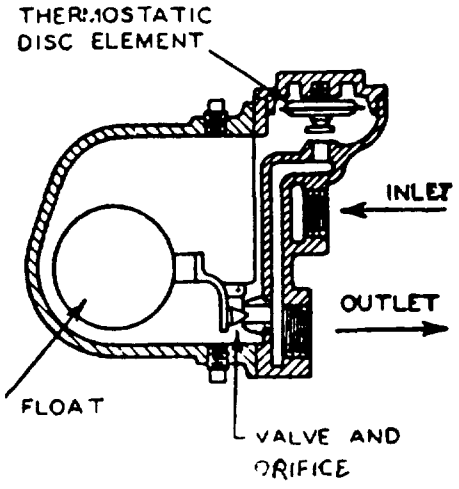


FIGURE 13-9

Combination Float and Thermostatic Trap

This trap is merely a combination of float and thermal traps.

Air is vented to atmosphere ahead of the steam and condensate because the thermostatic disc element, in its relaxed position, holds the port open.

When steam or hot condensate gets to the trap, the disc expands and closes the port. As the condensate level rises, the float opens the outlet valve and steam pressure forces the condensate into the return main.

Float and thermostatic traps are used for dripping mains and supply risers in systems varying from vacuum to 40 psig where large quantities of condensate must be returned.

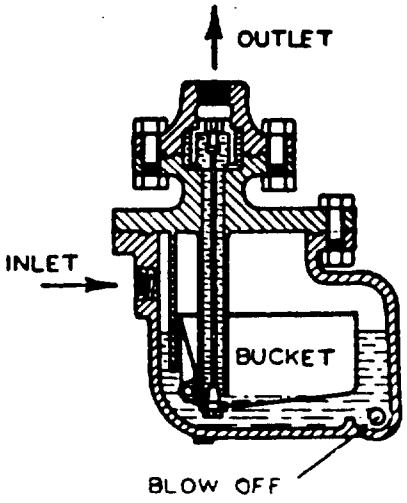


FIGURE 13-10

Upright Bucket Trap

This diagram shows a typical upright bucket trap. Condensate enters the inlet. As it rises the bucket floats and closes the outlet valve.

Eventually, the outer lip of the bucket runs into the baffle of the body of the trap and is held stationary. The condensate continues to rise, flows over the sides of the bucket and sinks it, opening the outlet valve.

Condensate is discharged from inside the bucket until the bucket regains its buoyancy and closes valve.

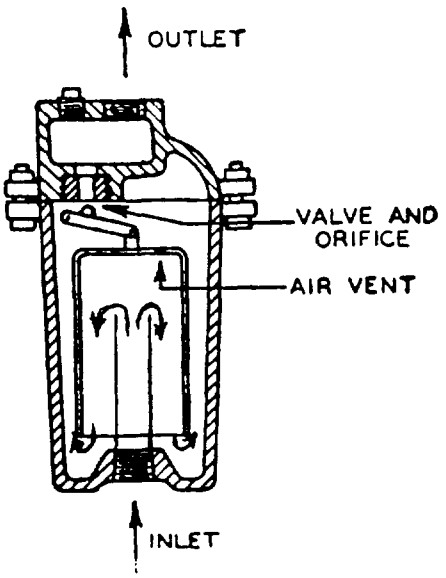


FIGURE 13-11

Inverted Bucket Trap

The inverted bucket trap illustrated in Figure 13-11 is discharging. Pressure at the condensate inlet is forcing the water out of the bucket and into the trap body. As the water leaves the bucket it is replaced by air and steam, the bucket becomes buoyant, rises and closes the orifice.

The escape of air from inside the bucket through small bleed hole, marked air vent, permits more condensate to enter. Shortly, the bucket loses buoyancy, sinks and the discharge portion of the cycle is repeated.

Air is automatically eliminated through the small air vent at the top of the inverted bucket. They are particularly suited to systems handling abnormal amounts of both air and condensate.

They come in sizes from $\frac{1}{2}$" to 3" and for pressures varying from vacuum to 2,400 psig. They are particularly suited to high pressure systems.

Both types of bucket traps are used for draining air and condensate from blast coils, unit heaters and steam mains.

They both use large size exhaust ports. They are designed for use in systems where high pressure might collapse a float.

Combination Bucket-Thermal Traps

With blast coils and unit heaters in low pressure systems, bucket traps are equipped with bellows or disc type thermal traps for venting air. The bucket element returns the condensate.

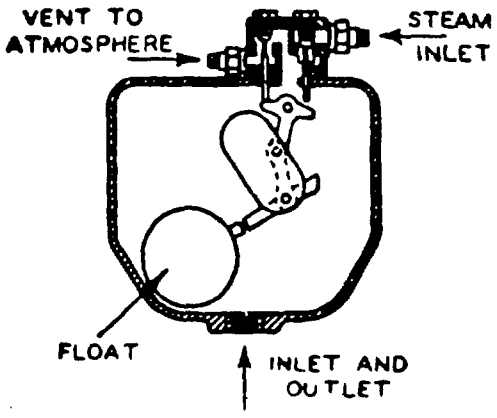


FIGURE 13-12

Boiler Return Trap

When the pressure difference between the boiler and the end of the dry return is too great to permit proper drainage of condensate back to the boiler, a boiler return trap is used as shown in Figure 13-12. The trap should be located close to the boiler and about 18 inches above the boiler water line.

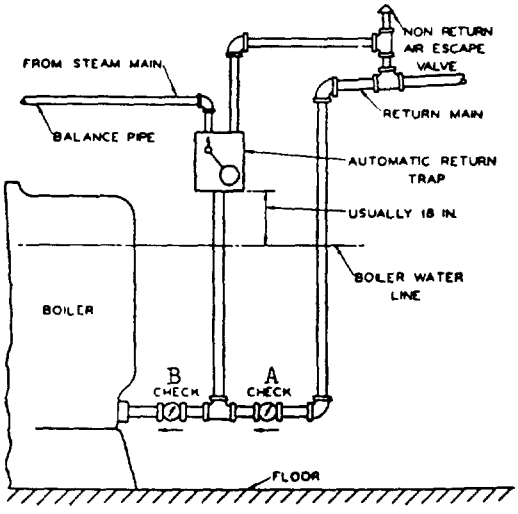


FIGURE 13-13

During the period the trap is collecting condensate, the float has dropped. See Figure 13-13. Condensate flows from the end of the dry return, down the drop leg through through check valve A and up into the trap. Check valve B prevents water from leaving the boiler. Also during this process air in the trap may be vented to atmosphere or back to the dry return where it is discharged to atmosphere through a vent valve.

Eventually the float rises, closes the air vent port and opens the steam inlet port, allowing boiler pressure to enter the trap through the balance pipe from the steam main.

Both trap water and boiler water are now under the same steam pressure. The additional head, because trap water level is above boiler water level, closes a check valve A to condensate return and opens a check valve B to the boiler return and condensate flows from trap to boiler.

Impulse Traps

In steam mains, unit heaters, laundry equipment, sterilizers, and other equipment impulse traps are used for draining widely varying amounts of condensate. Pressure at the trap outlet must be 25% or less than the pressure at the trap inlet.

Condensate Pump

In some steam systems the end of the main is too low to permit the condensate return to the boiler by gravity. A condensate pump must be used.

A condensate pump consists of a motor-driven centrifugal pump plus a receiving tank with an automatic float level control.

Condensate flows by gravity into the return tank, the top of which must, of course, be below the lowest return.

A vent, run to at least 10 inches above the boiler water level, avoids pressure in the tank, allowing it to fill properly.

The float level control in the tank, when it reaches a specified height, actuates the pump motor. The water in the tank is then pumped into the boiler through a swing check valve located as close to the pump on the boiler side.

Make-up water, regulated by another float valve in the receiving tank is delivered to the tank.

Vacuum Pump

The vacuum pump draws air and condensate from the return side of the system...separates them...and discharges air to the atmosphere and pumps the condensate back into the boiler.

Low vacuum pumps can maintain approximately 10" of vacuum...high vacuum pumps, 20" or more.

Where a return from the system is below the inlet to the receiver, it is necessary to install an accumulator tank below the lowest return along with a pump. The accumulator pump is also controlled by a float lever.

Vacuum Lift

The vacuum lift demonstrates one of the advantages of a vacuum steam system. It can raise condensate from a radiator or return to a higher level in the return system.

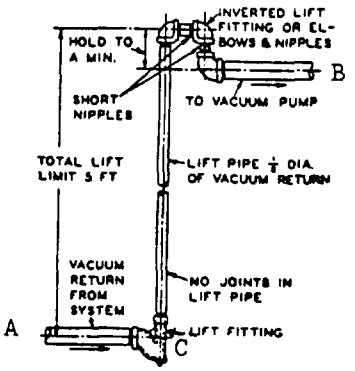
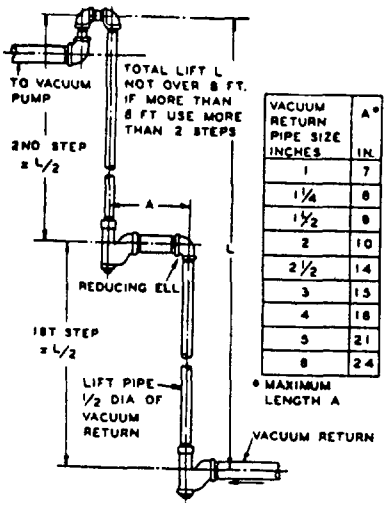


FIGURE 13-14

Assume a return main (A) below return main (B). Condensate will accumulate at bend (C) until it fills up the bend sealing off A from B.

Because the vacuum pump reduces the pressure on the outlet side B, the slug of water in the elbow will be drawn up the vertical pipe into return main B.



VACUUM RETURN PIPE SIZE INCHES	A* IN.
1	7
1¼	8
1½	9
2	10
2½	14
3	15
4	18
5	21
6	24

FIGURE 13-15.

Single lifts should be limited to 5 feet. When lifts are greater than 5 feet, a step lift is employed. See Figure 13-15.

Lifts should be avoided whenever possible.

Generally, steam mains are not sized smaller than two inches -- fewer hangers required to prevent sagging. On large systems the end of the main is kept at least half the size of the main at the boiler. Mains should be pitched at least 1/4" every 10 feet, and radiation connections should be pitched 1/2" per foot. Where radiator runouts longer than 8 feet cannot be pitched 1/2" per foot, use the next largest pipe size.

Tables 15-4 and 15-4A show nominal pipe sizes for returns -- both mains and risers -- for varying pressure drops from 1/32 psi to 1/2 psi. Again the pressure drop is assumed to occur in 100 feet, from an initial pressure of 3-1/2 psi, and the figures in the body of the table represent capacities in EDR. Three columns for each pressure drop are shown--wet, dry and vacuum returns. This table can be used in sizing the piping on two-pipe systems. The wet return column is also used for one-pipe systems.

Table 15-4

Return Main and Riser Capacities for Low-Pressure Systems---EDR

Pipe Size Inches	1/32 PSI or 1/2 Oz.			1/24 PSI or 2/3 Oz.			1/16 PSI or 1 Oz.		
	WET	DRY	VAC.	WET	DRY	VAC.	WET	DRY	VAC.
MAINS									
3/4	-	-	-	-	-	168	-	-	100
1	500	248	-	580	284	572	700	320	700
1-1/4	852	520	-	992	596	976	1,200	672	1,200
1-1/2	1,352	824	-	1,572	944	1,522	1,900	1,060	1,900
2	2,800	1,880	-	3,240	2,140	3,260	4,000	2,300	4,000
2-1/2	4,720	3,040	-	6,320	3,472	5,440	6,720	3,800	6,720
3	7,520	5,840	-	8,520	6,240	8,720	10,720	7,000	10,720
3-1/2	11,000	7,880	-	13,200	8,800	13,000	16,000	10,000	16,000
4	15,520	11,720	-	18,320	13,400	18,000	22,000	15,000	22,000
5	-	-	-	-	-	31,520	-	-	38,720
6	-	-	-	-	-	50,400	-	-	62,000
RISERS									
3/4	-	192	-	-	192	572	-	192	700
1	-	452	-	-	452	976	-	452	1,200
1-1/4	-	992	-	-	992	1,552	-	992	1,900
1-1/2	-	1,500	-	-	1,500	3,260	-	1,500	4,000
2	-	3,000	-	-	3,000	5,440	-	3,000	6,720
2-1/2	-	-	-	-	-	8,720	-	-	10,720
3	-	-	-	-	-	13,000	-	-	16,000
3-1/2	-	-	-	-	-	17,920	-	-	22,000
4	-	-	-	-	-	31,520	-	-	38,720
5	-	-	-	-	-	50,400	-	-	62,000

Table 15-4A

Return Main and Riser Capacities for Low Pressure Systems---EDR

Pipe Size Inches	1/8 PSI or 2 Oz.			1/4 PSI or 4 Oz.			1/2 PSI or 8 Oz.		
	WET	DRY	VAC.	WET	DRY	VAC.	WET	DRY	VAC.
MAINS									
3/4	-	-	568	-	-	800	-	-	1,132
1	1,000	412	996	1,400	460	1,400	-	-	1,976
1-1/4	1,700	868	1,704	2,400	964	2,400	-	-	3,392
1-1/2	2,700	1,360	2,696	3,800	1,512	3,800	-	-	5,360
2	5,600	2,960	5,680	8,000	3,300	8,000	-	-	11,320
2-1/2	9,400	4,920	9,520	13,400	5,440	13,400	-	-	18,920
3	15,000	9,000	15,200	21,400	10,000	21,400	-	-	30,240
3-1/2	22,000	12,920	22,720	32,000	14,320	32,000	-	-	45,200
4	31,000	19,320	31,240	44,000	21,520	44,000	-	-	62,000
5	-	-	54,800	-	-	77,600	-	-	109,200
6	-	-	88,000	-	-	124,000	-	-	175,200
RISERS									
3/4	-	192	996	-	192	1,400	-	-	1,976
1	-	452	1,704	-	452	2,400	-	-	3,392
1-1/4	-	992	2,696	-	992	3,800	-	-	5,360
1-1/2	-	1,500	5,680	-	1,500	8,000	-	-	11,320
2	-	3,000	9,520	-	3,000	13,400	-	-	18,920
2-1/2	-	-	15,200	-	-	21,400	-	-	30,240
3	-	-	22,720	-	-	32,000	-	-	45,200
3-1/2	-	-	31,240	-	-	44,000	-	-	62,000
4	-	-	54,800	-	-	77,600	-	-	109,200
5	-	-	88,100	-	-	124,000	-	-	175,200

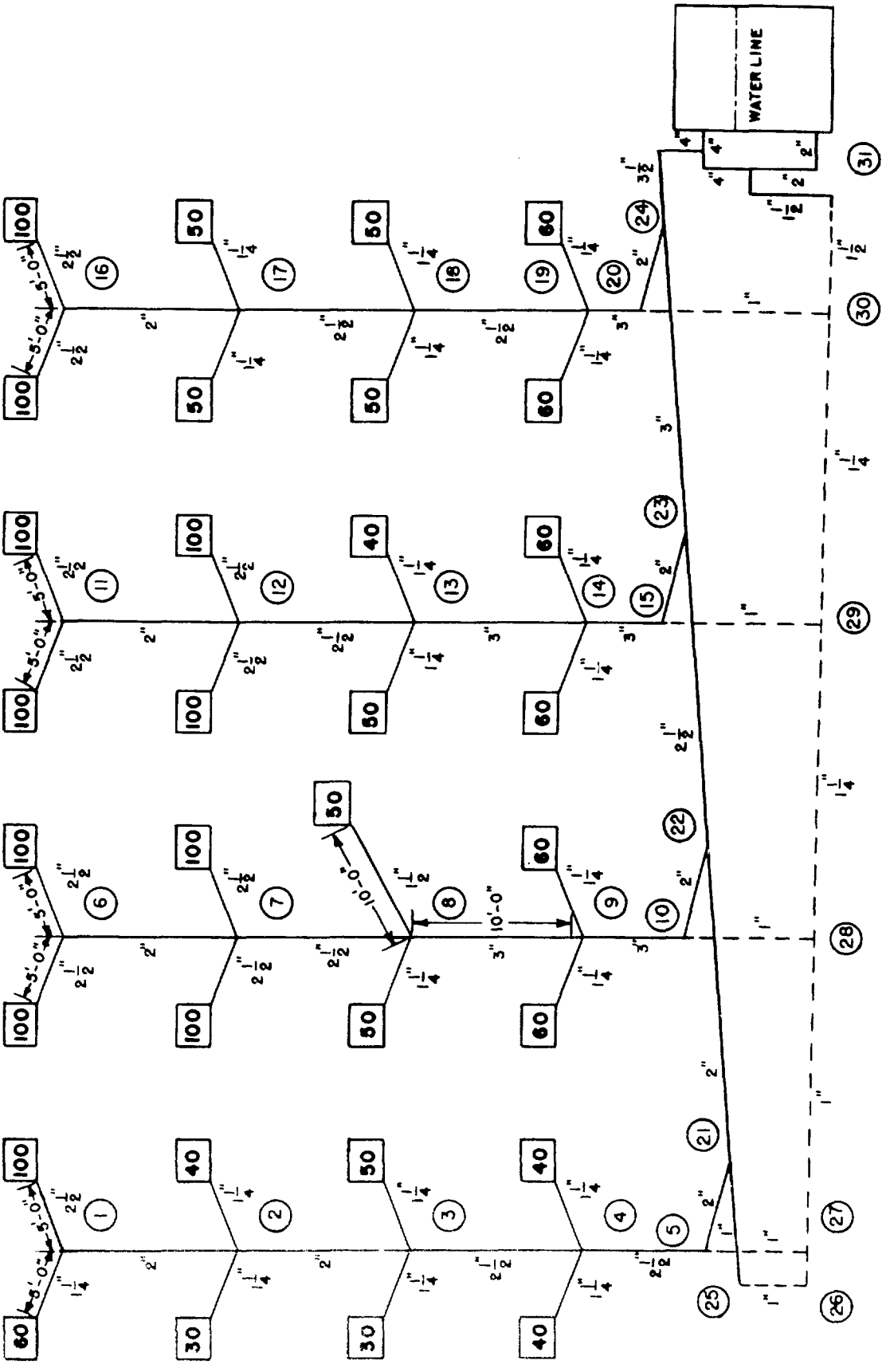


FIGURE 15-1

The schematic drawing (Figure 15-1) shows a typical one-pipe low pressure system. Since it is most frequently used, we assume a designed pressure drop of 1/8 psi.

All pipe sizes are shown. However, using our tables, let's see how they were calculated.

Pipe sizing is always started at the top of the steam riser farthest from the boiler -- 1 to 5 in this example.

Since this is a one-pipe system in which the condensate flows against the steam we use Table 15-3 to size all steam risers and radiator runouts.

All runouts are 5 feet in length and can be installed with the proper pitch of 1/2" per foot.

Since the smallest radiator on this riser has an EDR capacity of 30 square feet (which is larger than 28--See Table 15-3), the radiator runouts must be at least 1-1/4" pipe, which will accommodate all pieces of radiation up to 64 EDR. The radiator larger than this, upper left, with an EDR of 100 square feet, needs a 2-1/2" pipe.

To size riser #1, which must supply two radiators with a total of 160 square feet of EDR, we find under the heading "Supply Risers Upfeed" that a 1-1/2" pipe will not do it, but that a 2" pipe will handle up to 288 square feet of radiation which will not only take care of the 160 in the two top radiators, but will also serve the additional 70 square feet in the two radiators on the next floor down. So, we can use two-inch pipe in both Risers #1 and #2.

However, the addition of 80 more square feet of EDR on the next lower floor raises the accumulated total to 310. This, the table tells us, needs a 2-1/2" pipe for Riser #3.

Adding 80 more square feet of EDR for the first floor radiation...a total, now, of 390, is still well below the maximum of 464 for 2-1/2" pipe so that we can use 2-1/2" pipe for Riser #4, too.

All other risers and radiator runouts can be calculated from the table in the same way, except the 50 square feet of EDR radiator to the right of Riser #8 on the second floor. The amount of radiation calls for a 1-1/4" pipe, but because the runout is 10 feet long and we can't get 1/2" pitch per foot, we use the next largest pipe size, 1-1/2".

The next step is to size the riser runouts from steam main to risers: 5 to 21...10 to 22...15 to 23...20 to 24. Since these runouts are dripped and do not carry condensate flowing against the steam, they are sized according to Table 15-2, under the column 1/8 psi.

Adding up the total load on each riser we get for riser runouts...

	Square feet of EDR
5 to 21...	390
10 to 22...	620
15 to 23...	610
20 to 24...	520

Since all loads are greater than 336 and less than 648 square feet of EDR, all riser runouts will be 2" pipe

To complete the supply side of the system we calculate sizes for the four steam main segments. Again, because the risers are dripped we use the 1/8 psi column in Table 15-2. The load is increased by the load of each riser as the boiler connection is approached.

Steam Main	Cumulative Riser Load	Pipe Size
21 to 22	390 sq. ft.	2"
22 to 23	1,010 " "	2-1/2"
23 to 24	1,620 " "	3"
24 to boiler	2,140 " "	3-1/2"

To figure pipe sizes in the returns use Table 15-4A and the same pressure drop: 1/8 psi.

The "dry" return 21 to 25 has a load of 390 square feet, so a 1" pipe will handle it.

The riser drips (5 to 27...10 to 28...15 to 29...and 20 to 30) are all "wet" and carry loads of less than 1,000 square feet which is indicated in the "wet" column as needing a 1-1/4" pipe...all riser drips in our example can safely use a one-inch pipe.

We complete sizing the system by sizing the wet return by segments: 25 to 26...26 to 27...27 to 28...28 to 29...29 to 30...30 to 31...31 to boiler connection.

As in sizing the steam main segments we increase pipe size as load increases closer to the boiler connection. Again using the "wet" column we get:

Return Segment	Cumulative Load Sq. Ft.	Pipe Size
25 to 26	390	1"
26 to 27	390	1"
27 to 28	390	1"
28 to 29	1,010	1-1/4"
29 to 30	1,620	1-1/4"
30 to 31	2,140	1-1/2"
31 to boiler	2,140	1-1/2"

The following table shows a step-by-step procedure for sizing the piping in this example. It is also a form to follow in estimating jobs. Take the time to study it carefully...then set up a duplicate on a blank paper -- leaving out only the last column. Then using the tables, calculate all the pipe sizes required. Like most old proverbs, it is true that practice makes perfect.

Table 15-5

Procedure Step	Part of System	Section of Piping	E.D.R. Sq. Ft.	Theoretical Pipe Size
1	Radiator runout	1	60	1-1/4
	Radiator runout	1	100	2-1/2
	"	2	30	1-1/4
	"	2	40	1-1/4
	"	3	30	1-1/4
	"	3	50	1-1/4
	"	4	40	1-1/4
	"	4	40	1-1/4
2	Riser	1 to 2	160	2
3	"	2 to 3	230	2
4	"	3 to 4	310	2-1/2
5	"	4 to 5	390	2-1/2
6	Radiator runout	6	100	2-1/2
	"	6	100	2-1/2
	"	7	100	2-1/2
	"	7	100	2-1/2
	"	8	50	1-1/4
	"	8	50	1-1/2
	"	9	60	1-1/4
	"	9	60	1-1/4
7	Riser	6 to 7	200	2
	"	7 to 8	400	2-1/2
	"	8 to 9	500	3
	"	9 to 10	620	3

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Table 15-5A

Procedure Step	Part of System	Section of Piping	E.D.R. Sq. Ft.	Theoretical Pipe Size
8	Radiator runout	11	100	2-1/2
	"	11	100	2-1/2
	"	12	100	2-1/2
	"	12	100	2-1/2
	"	13	50	1-1/4
	"	13	40	1-1/4
	"	14	60	1-1/4
	"	14	60	1-1/4
	Riser	11 to 12	200	2
	"	12 to 13	400	2-1/2
	"	13 to 14	490	3
	"	14 to 15	610	3
9	Radiator runout	16	100	2-1/2
	"	16	100	2-1/2
	"	17	50	1-1/4
	"	17	50	1-1/4
	"	18	50	1-1/4
	"	18	50	1-1/4
	"	19	60	1-1/4
	"	19	60	1-1/4
	Riser	16 to 17	200	2
	"	17 to 18	300	2-1/2
	"	18 to 19	400	2-1/2
	"	19 to 20	520	3
10	Riser runout	5 to 21	390	2
	"	10 to 22	620	2

Table 15-5B

Procedure Step	Part of System	Section of Piping	E.D.R. Sq.Ft.	Theoretical Pipe Size
10 (cont'd.)	Riser runout	15 to 23	610	2
	"	20 to 24	520	2
11	Steam main	21 to 22	390	2
	"	22to 23	1010	2-1/2
	"	23 to 24	1620	3
	"	24 to boiler	2140	3-1/2
12	Dry return main	21 to 25	390	1
13	Riser drip	5 to 27	390	1
	"	10 to 28	620	1
	"	15 to 29	610	1
	"	20 to 30	520	1
14	Wet return main	25 to 26	390	1
	"	26 to 27	390	1
	"	27 to 28	390	1
	"	28 to 29	1010	1-1/4
	"	29 to 30	1620	1-1/4
	"	30 to 31	2140	1-1/2
	"	31 to boiler	2140	1-1/2

For two pipe systems (low pressure, vapor and vacuum) pipes are sized in the same way. A pressure drop of 1/8 psi is recommended and the same pressure drop should be used for steam mains, risers and condensate return piping.

Tables 15-2, 15-3 and 15-4 should be used.

And again, mains should be pitched 1/4" in 10 feet, horizontal runouts to risers and radiators should be pitched 1/2" per foot and the next larger pipe size should be specified for runouts of over 8 feet where a 1/2" pitch per foot is not possible.

CHAPTER 16

PRESSURE STEAM SYSTEMS

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CHAPTER 16

PRESSURE STEAM SYSTEMS

Steam heating systems operating under pressures higher than atmosphere (gage pressures) are classified as either low pressure (up to 15 psig) or high pressure (above 15 psig...usually 30 to 150 psig).

This chapter will be mainly devoted to the low pressure system beginning with the review of some of the basic material covered in Chapter 12.

A one-pipe low pressure system uses the same pipe to carry steam to the radiation and condensate back to the steam main.

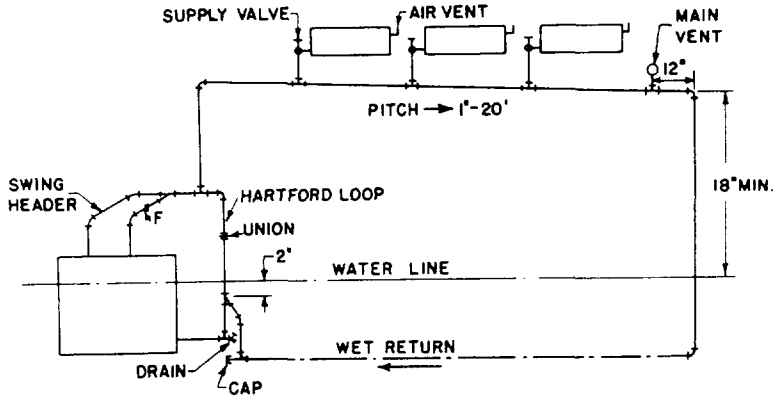


FIGURE 16-1

This is a typical one-pipe up-feed system with gravity return usually found in residences. This arrangement has limits on its size and capacity.

As the steam is generated in the boiler, it pushes the air in the system ahead of it and out the vents on each radiator and the main vent at the end of the supply main.

When steam hits any vent, it closes. The supply main is pitched away from the boiler to carry condensate in the same direction as the steam is flowing.

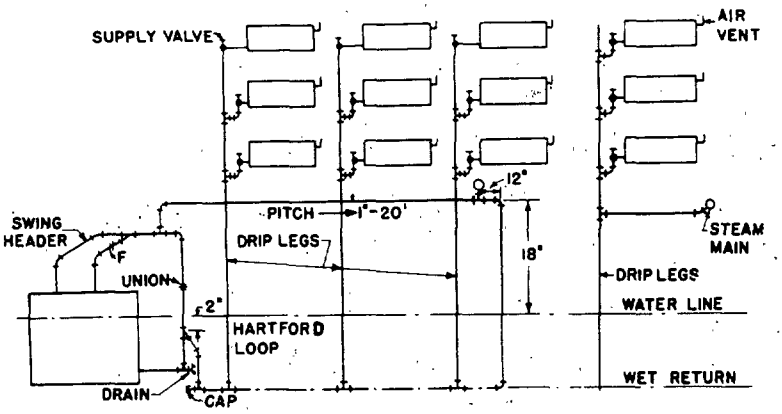


FIGURE 16-2

This version of a one-pipe up-feed system shows the risers being dripped into the wet return.

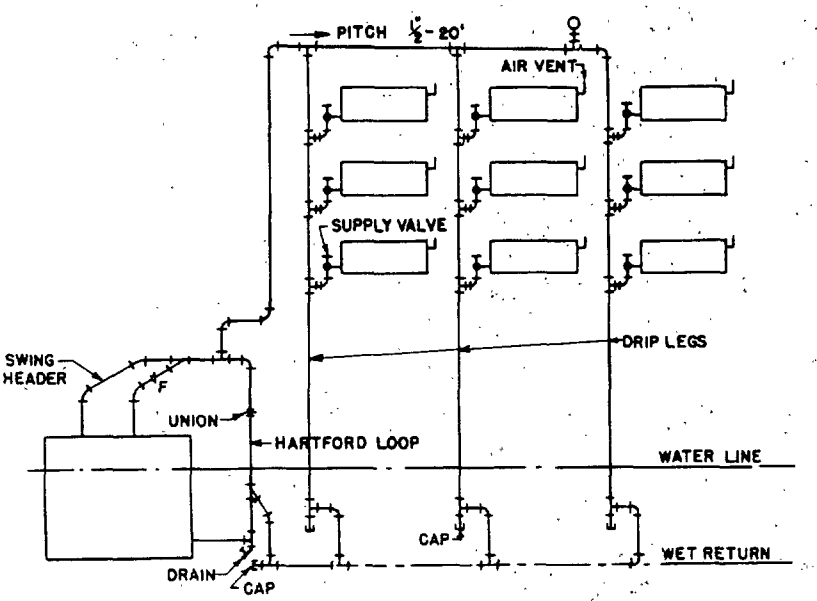


FIGURE 16-3

Another piping arrangement is the overhead main or downfeed system. It is often used in multi-story apartment houses. (See Figure 16-3).

The downfeed risers are dripped into the wet return. This arrangement, of course, also allows for carrying the design load quietly.

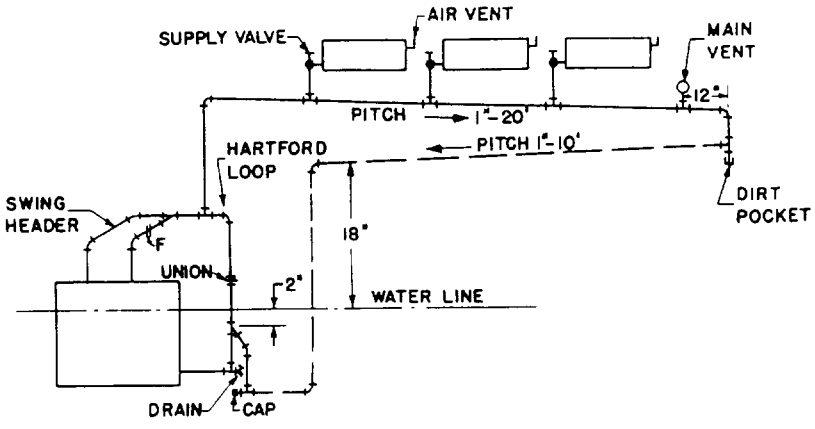


FIGURE 16-4

This variation...with a dry return...is used where a wet return is either impossible or impractical...in most cases where there is only a partial basement.

Note that we need the same 18" clearance between the end of the dry return and the water line as at the end of supply mains in earlier examples.

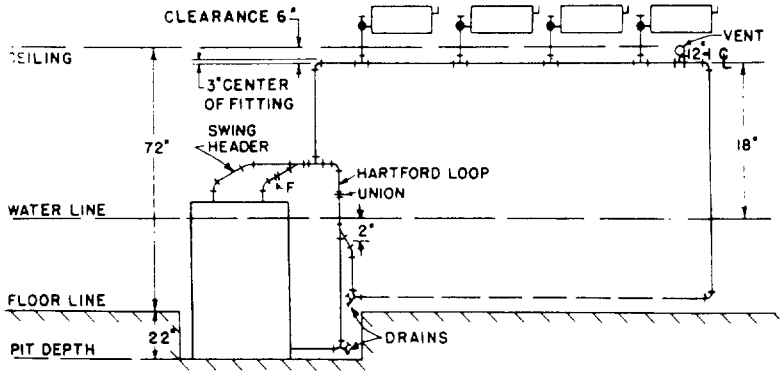


FIGURE 16-5

Occasionally a limited ceiling height in the basement will not allow for this minimum 18" clearance between boiler water line and the end on the steam main.

The solution is to dig a pit and set the boiler enough below the floor level to provide the clearance.

In all one-pipe systems the steam main is vented as well as each piece of radiation to allow for quick elimination of air.

One-pipe systems should use vent valves with adjustable venting rates. This kind of valve provides uniform distribution of steam throughout the system, particularly on automatically fired jobs. By adjusting the venting rates at each radiator, a more uniform heating of each room is obtained. Of course, the boiler must produce enough steam to fill all pieces of radiation.

Another type of vent valve used on radiators is the vacuum vent which allows air to escape but prevents its re-entry into the radiator. These valves are sometimes used on radiators in one-pipe steam systems to convert the system to vapor type operation. These systems so converted should not be confused with the conventional two-pipe vapor system which is discussed in the next chapter.

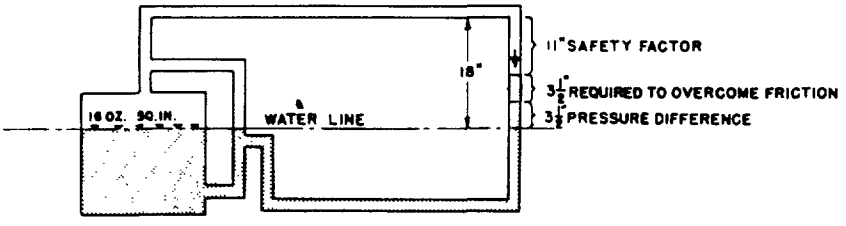


FIGURE 16-6

To fully explain the necessity for an 18" differential between boiler water level and the end of the steam main or dry return...

Our steam system is, in effect, a giant, closed "U" tube with a boiler built into it. The water in the tube seeks its own level, and with the boiler shut down, will stand as high in the drop leg as in the boiler.

With the boiler firing, pressure will build up from the surface of the boiler water and flow in all directions, pressing down on the surface of the water in the boiler and also on the surface of the water in the drop leg.

If both pressures were equal, the water in the leg would continue to stand at the same level as the water in the boiler.

However, some pressure is lost in overcoming the friction in the steam main.

Assuming 16 ounces of pressure at the boiler end and 2 ounces of resistance to flow in the steam main, then pressure on the water in the leg would be only 14 ounces...a pressure drop of 2 ounces or 1/8 psig.

Since one pound of pressure is equal to 28" of water (see Chapter 2), a drop of 1/8 psig means a rise in the water level in the leg of 3-1/2".

Resistance to water flow in the wet return is estimated at 3-1/2 and must also be added.

The normal total rise of water in the drop leg becomes 7 inches.

Allowing 11 inches of additional height for the greater pressure drop occurring during heating up of the system and as protection against surging makes the total height between normal boiler level and end of the steam main 18 inches.

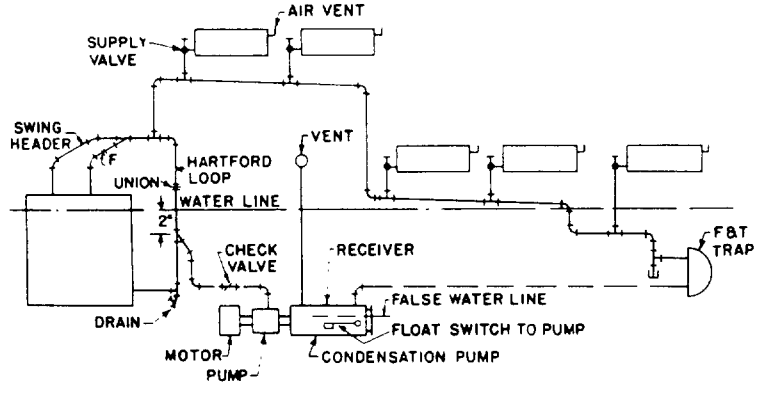


FIGURE 16-7

When the available head room will not allow the recommended 18" clearance, the boiler can be pitted, as shown in Figure 16-5, or a false water line can be established by the use of a condensate pump and float and thermostatic (F & T) trap. (See Chapter 13).

The water line of the condensate pump is the new theoretical water line of the boiler, eliminating the need for the minimum 18" clearance between water line and the end of the main.

The use of condensate pumps allows for a number of variations in one-pipe systems.

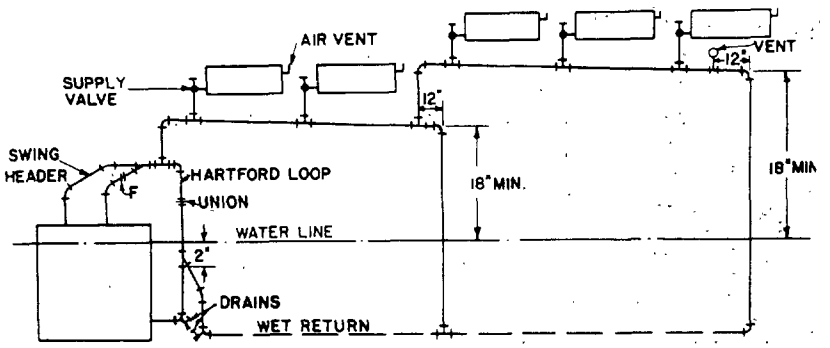


FIGURE 16-8

Another problem sometimes encountered with one-pipe systems is the necessity for an exceptionally long main in a low-ceilinged basement.

The answer is to drip the main when it reaches 18" above the water line and then use a riser so you can continue the proper pitch from a higher level.

This method of handling head room problems must still consider pressure drops to the end of the last main section and pipes must be sized accordingly.

This is just one example of the fact that, when the basic fundamentals of steam flow and pressure drops are observed, steam piping can, literally, be bent like a pretzel.

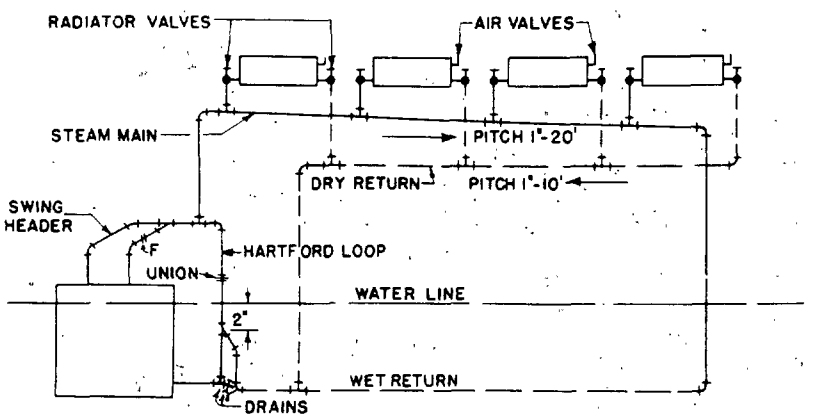


FIGURE 16-9

The original two-pipe, low pressure system had a number of built-in weaknesses.

Here each piece of radiation drips into a dry return (the second pipe).

The most obvious fault was steam entering the return leg of a radiator and closing the air valve before the radiator was fully vented, resulting in a partially filled radiator.

Another problem was caused by closing the supply valve to any radiator. With the return valve open, the system itself was open to the atmosphere, creating an unbalanced pressure that could back water out of the boiler.

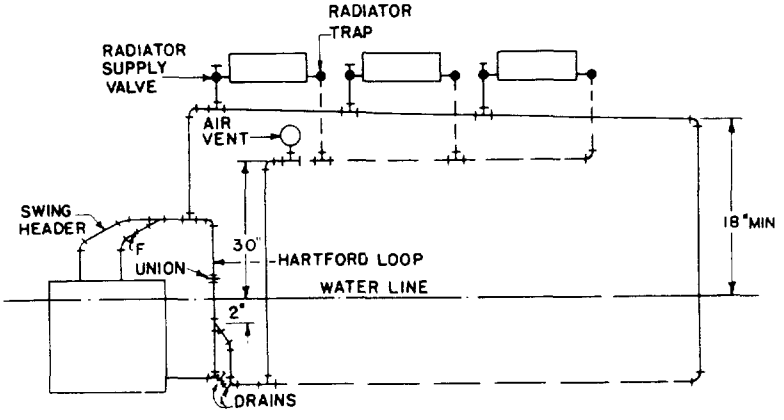


FIGURE 16-10

With the advent of the radiator steam trap, the two-pipe system took a little different design.

The angle valve on the return end of the radiator is replaced by a radiator steam trap.

The trap passes both air and condensate into the return. All air is vented from the system at the one main vent.

Because the traps stops its flow, steam cannot pass through a small radiator into the return and feed back up the return side of another radiator, closing the air valve and sealing air into it.

However, this system still has the fault of being open to the atmosphere through the single air vent.

With the traps closed, the pressure on the water in the boiler can become considerably higher than the pressure in the dry return, causing the water to back out of the boiler and into the dry return, sealing off the air vent and air-binding the system.

The drawing calls for a differential of 30 inches between boiler water level and vent. This means that if we confine the system total pressure drop to 8 ounces (14 inches of water) or less, the vent will continue above water and air released properly. In systems where air binding becomes a problem, a boiler return trap can be installed.

Some years back American-Standard sponsored an "adjustable orifice system" which was of this same general design. Orifices in the supply valves could be adjusted to the size of the radiator. Theoretically, no traps were necessary to stop steam flow into the return, because steam was supplied to the radiator in direct relationship to its ability to condense it. It is still necessary to maintain 30" between the end of the dry return and the water line.

A number of these systems are still in use.

These are the principal variations in low-pressure steam systems.

There are, too, high pressure systems for very large buildings and for process work where cast iron boilers have little or no application.

CHAPTER 17

STEAM SYSTEMS— PRESSURE BELOW ATMOSPHERE

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CHAPTER 17

STEAM SYSTEMS -- PRESSURE BELOW ATMOSPHERE

There are three different kinds of two-pipe steam heating systems that operate at pressures below atmospheric pressure or under vacuum.

1. The vapor system
2. The vacuum system
3. The subatmospheric system

The Vapor System

Vapor systems are intermittently fired usually under the control of a thermostat.

When starting the system, the burner operates until above-atmospheric pressures are generated and the air has been expelled from the system.

During the off-period of the burner, the condensation of the steam creates a vacuum which is held by the vacuum vent valve as described in Chapter 13. This valve, while permitting air to escape from the system, prevents the air from re-entering.

As the vacuum is increased and the absolute pressure in the system becomes less, the water continues to boil. The source of heat for the boiling is the heat stored in the water itself and the metal of the boiler.

For example, at 1 psig the boiling temperature of water is 215°F, while at 10 inches of mercury vacuum the boiling temperature is only 192°F. At 20 inches of vacuum, the boiling temperature drops to 162°F.

Once the space is heated and the air expelled, the system stays full of steam but at progressively lower temperatures until the burner is recycled. At that time there is a uniform increase of pressure throughout the system, a correspondingly uniform increase in steam temperature in all the radiation and even distribution of heat to the space. Usually on the second and subsequent burner cycles, the thermostat is satisfied before the steam pressure gets above atmosphere.

However, the vacuum vent valves are not absolutely tight. During a twenty-four hour period air leaks into the system and the vacuum is gradually destroyed.

On morning pick-up of the space temperature the burner is on a sufficient time to once again develop stream pressures above atmosphere and cause the expulsion of air from the system.

Vapor systems may be identified from the following:

1. Packless steam valves installed at the top of the radiation.
2. Thermal traps on the radiation.
3. Vacuum vent valves on the dry returns.
4. Boiler return trap usually used. See Chapter 13.

The Vacuum System

In the vacuum system the boiler and supply mains operate under pressures above atmosphere. The return mains--and sometimes the radiation--operate under a vacuum created by a pump which removes the air from the system--and discharges it to atmosphere. The same pump also acts as a condensate pump returning the condensate to the boiler.

Pressure on the supply side of the system vary from 1 to 10 psig and vacuums on the return side are held at approximately 10 inches of Hg.

Packless valves are employed on the radiation. These valves installed at the top of the radiation may be adjusted to vary the steam delivery to the radiation and the heat delivery to the space.

The Subatmospheric System

The subatmospheric system, which is a vacuum system, is designed to provide modulated heat input to the space.

Heat output from the radiation is controlled by varying pressure, temperature and specific volume of the steam. Unlike the vacuum system with the vacuum primarily on the return side, the subatmospheric system has a controllable vacuum maintained on both the supply and return by a pump.

Pressures in this system may be in excess of atmospheric in severe winter weather and vacuums of as much as 25 inches may be maintained in the system during mild weather.

When heating of domestic hot water by the boiler is required, the boiler operates at sufficient pressure to satisfy the domestic hot water demand. Steam to the system is supplied in modulated quantity by thermostatically controlled floating type steam valve on the main.

If the boiler is not required for domestic hot water, steam generation may be thermostatically modulated by controlling the burner.

Radiator supply valves may have adjustable orifices in them. In some systems orifices are omitted.

The thermal traps are designed to operate from 15 psig to 26 inches Hg.

Some of these systems are proprietary.

STEAM BOILER MANIFOLD SIZING

CHAPTER 18

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CHAPTER 18

STEAM BOILER MANIFOLD SIZING

Steam boiler manifolding is important. Manifold boiler connections must be large enough to restrict steam velocities to 60 feet per second or less. At the same time, oversized manifolds and boiler connections add unnecessary costs to the installation.

For any boiler being fired at its specified rate, the velocity of the steam leaving the boiler depends on the size of the connecting piping. Obviously, to carry the same amount of steam, a four inch pipe will require higher velocities than a six inch.

Steam velocities higher than 60 feet per second are to be avoided because:

1. Noisy steam flow
2. Steam carries entrained water particles out of boiler -- is wetter
3. Increases possibility of priming or slugging
4. Greater chance of water hammer with unnecessary water in the mains.

In the larger sizes of sectional cast iron boilers, multiple outlets are desirable to maintain a uniform water level.

For example, if steam were taken from one end of a long boiler, the steam pressure at the end of the steam outlet would be lower than at the opposite end. The water in the boiler would behave as in a U-tube with the level sloping upward toward the steam outlet end. The possibility of priming, slugging, or other erratic operation is prevalent depending on the location of the low water cut-off and boiler feed control.

All of these factors have been considered in the design of the recommended piping arrangements shown in the Technical Catalogue Bulletins issued on American-Standard boilers.

The pound rate-of-flow of dry or nearly dry steam through a pipe is determined by the size of the pipe and the specific volume (cubic feet per pound) of the steam.

The basic flow equation is:

$$\text{Volume} = \text{Area} \times \text{velocity}$$

To determine the proper pipe size the above equation is re-written:

$$\text{Area} = \frac{\text{Volume}}{\text{Velocity}}$$

Where:

Area is expressed in square inches
Volume is expressed in cubic inches of steam per second
Velocity is expressed in feet per second

We know that about 1,000 Btu per hour or 1 Mbh of gross boiler capacity is equivalent to the heat required to produce one pound of steam at 1 psig and a standard barometer. A pound of steam under these conditions occupies approximately 25 cubic feet.

To substitute properly in the basic equation:

Volume in cubic inches per second

$$= \frac{\text{Mbh} \times 25 \times 1728}{3600}$$

$$= \frac{\text{Mbh} \times 43{,}200}{3600}$$

Velocity in inches per second

Velocity in ft./sec. x 12

Substituting the values for volume and velocity in the equation:

$$\text{Area} = \frac{\text{Volume}}{\text{Velocity}}$$

Area in square inches

$$= \frac{\text{Mbh} \times 43{,}200}{\text{Velocity (ft./sec.)} \times 12 \times 3600}$$

$$= \frac{\text{Mbh} \times 43{,}200}{\text{Velocity (ft./sec.)} \times 43{,}200}$$

$$= \frac{\text{Mbh}}{\text{Velocity (ft./sec.)}}$$

Example 1

Determine the size of the outlet piping from a steam boiler with a gross capacity of 1,000,000 Btu per hour (1,000 Mbh) at a steam velocity of 55 feet per second.

The total cross sectional area of the outlet piping in square inches equals

$$\frac{1000}{55} = 18.18 \text{ square inches}$$

Referring to Table 18-1 it is found that
one 5" outlet has 20.00 square inches of cross sectional area
or
two $3\frac{1}{2}$" outlets have a total of 19.78 square inches of cross sectional area.
Either one 5" pipe or two $3\frac{1}{2}$" pipes is adequate.

Example 2

What size piping is necessary to handle the steam flow at 60 feet per second velocity from a G-6013 boiler with an output of 1,392 Mbh?

The equation is:

$$\text{Area} = \frac{1392}{60} = 23.2 \text{ square inches}$$

From Table 18-1 it is found that

a 5" pipe has 20.00 quare inches of cross sectional area

and

a 6" pipe has 28.90 square inches of cross sectional area

Therefore, the 6" pipe is used.

Example 3

If the six inch outlet is installed what will the actual steam velocity be?

Transposing the equation

$$\text{Area} = \frac{\text{Mbh}}{\text{Velocity}} \quad \text{to Velocity} = \frac{\text{Mbh}}{\text{Area}}$$

to

$$\text{Velocity} = \frac{1392}{28.83} = 48.57 \text{ feet per sec.}$$

Table 18-1 may be used directly to determine the size of the steam outlet on piping.

Example 4

An 710-S0 boiler has a gross I=B=R rating of 1,760,000 But per hour. It also has four 4-inch flow tappings. If all four flow tappings are connected to a header, what will be the approximate steam velocity through each.

Dividing

$$\frac{1,760,000 \text{ Btu/hr.}}{4 \text{ flow pipes}} = 440,000 \text{ btu/hr. per pipe}$$

Referring to Table 18-1 on the 4-inch pipe size line, it is found that at a velocity of 45 feet per second a 4-inch line can handle 572,900 Btu per hour. Obviously, four flow connections are very safe. They are also more costly to make.

Let us try three four-inch supply connections to the boiler. Each four-inch line will carry 1,760,000 3 or approximately 586,700 Btu/hr. The velocity (Table 18-1) is slightly greater than 45 feet per second.

Three $3\frac{1}{2}$-inch supply connections could be used with a load of approximately 586,700 Btu per hour and a velocity under 60 feet per second.

Several sizes of manifold can be selected depending on what steam velocity is desired. (Table 18-1) The total load is 1,760,000 Btu per hour.

6-inch manifold---- velocity under 60 feet per second
7-inch manifold --- velocity under 45 feet per second

The most economical combination becomes:

Three $3\frac{1}{2}$-inch flow connections to the boiler
One 6-inch manifold.

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Table 18-1

Gross* Boiler Load

Pipe Size	Pipe Area	Btu/hr --- Square Feet of Radiation — Steam Velocity 45 ft./sec. Btu/hr.	45 ft./sec. sq. ft.	50 ft./sec. Btu/hr.	50 ft./sec. sq. ft.	55 ft./sec. Btu/hr.	55 ft./sec. sq. ft.	60 ft./sec. Btu/hr.	60 ft./sec. sq. ft.
2	3.36	151,000	630	167,800	700	184,600	770	201,400	840
2½	4.79	215,500	896	239,400	1,000	263,400	1,095	287,300	1,195
3	7.39	332,700	1,385	369,700	1,540	406,600	1,695	443,600	1,845
3½	9.89	444,800	1,850	494,300	2,060	543,700	2,260	593,100	2,470
4	12.70	572,900	2,385	636,500	2,650	700,200	2,910	763,800	3,180
5	20.00	900,000	3,760	1,000,300	4,160	1,100,300	4,600	1,200,400	4,000
6	28,90	1,297,400	5,400	1,441,500	10,650	2,813,900	11,700	3,069,700	12,800
8	51.2	2,302,300	9,600	2,558,100	10,650	2,813,900	11,700	3,069,700	12,800
10	81.60	3,671,000	15,400	4,079,300	17,000	4,487,200	18,650	4,895,100	20,400
12	114.80	5,166,000	21,500	5,740,000	23,900	6,314,000	26,300	6,888,000	28,700

*Gross: boiler output at boiler outlets

These figures are true for 1 psig of pressure. At greater pressures each pipe size can deliver more btu/hr.

Use the following factors to calculate btu/hr at greater pressures than 1 psig

3 psig	Add 9%	7 psig	Add 31%
5 psig	Add 19%	10 psig	Add 47%

HOT WATER—FORCED CIRCULATION

CHAPTER 19

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CHAPTER 19

HOT WATER -- FORCED CIRCULATION

There are gravity feed hot water heating systems, but they are found only in residences in older systems.

Forced circulation is used in all large hot water heating systems. They are rated in two different categories, depending on water temperature. High temperature hot water systems operate above 250°...usually 300° or higher and require high pressure boilers.

We will discuss, in this chapter, low temperature hot water (LTHW) systems operating below 250°. These systems also fall into two basic classifications according to whether they use one or two pipes as mains.

However, sometimes both basic piping arrangements are used in the same system...two-pipe systems can be either direct or reverse return...and sometimes a "series loop" is incorporated in the system.

So, there are four terms describing piping arrangements for forced circulation hot water systems:

1. One-pipe
2. Two-pipe (reverse or direct return)
3. Series Loop
4. Combinations of items 1 through 3

One-pipe

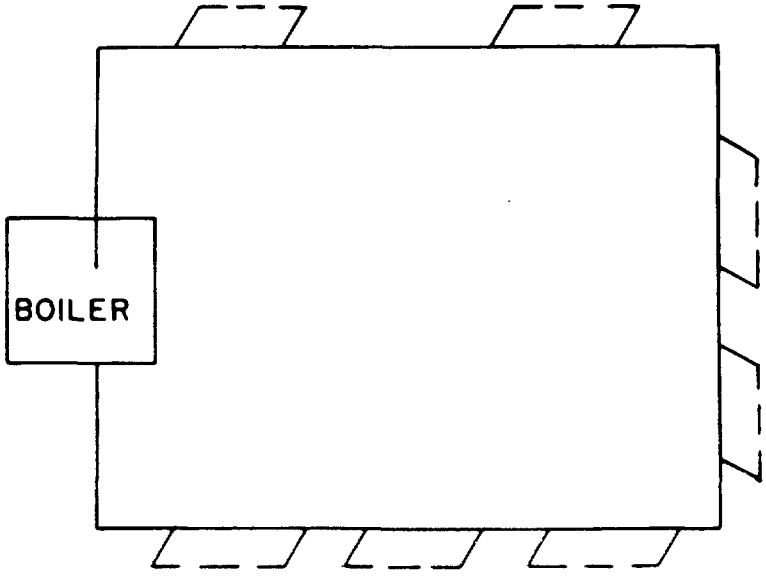


FIGURE 19-1

The one-pipe system is often the least expensive piping arrangement for large hot water systems. Its major problem is the drop in water temperature around the circuit. On carefully designed systems the water temperature drop is readily estimated and radiation sized for its average water temperature.

Two-Pipe

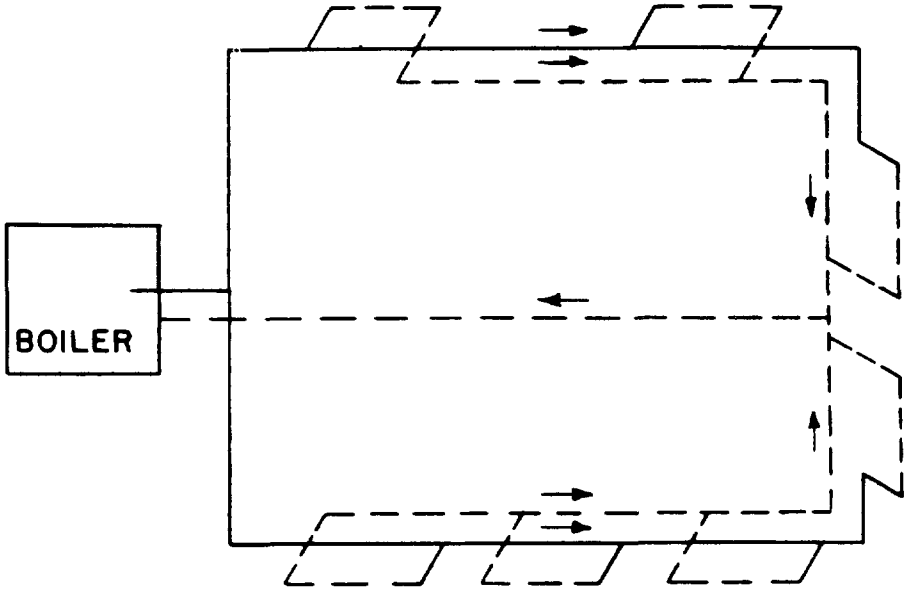


FIGURE 19-2

The two-pipe reverse return system with the water flowing in the same direction in both the supply and return mains provides an inherent balance. In general, the distance the water must flow from the boiler to any one radiator and return to the boiler is the same as the distance to and from another radiator.

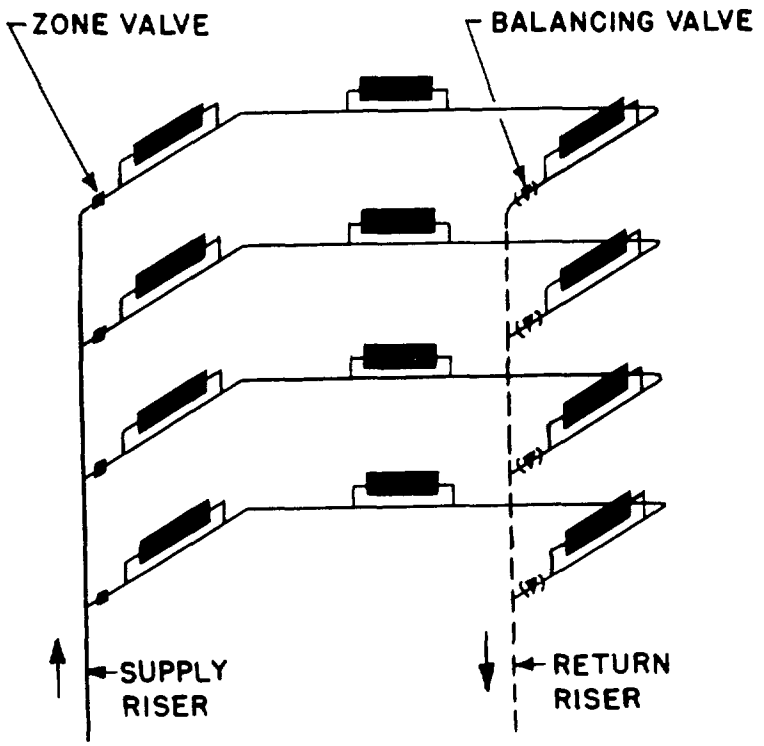


FIGURE 19-3

In some large systems the circuits connected to the supply and return risers perform as though they were on a two-pipe direct return system. For best operation the circuits should be installed with balancing valves to prevent possible short circuiting and insure the delivery of proper water quantities to each.

Zone valves may be installed on each circuit in addition to the balancing valves, when thermostatic control is desired. This circuiting is adaptable to apartments.

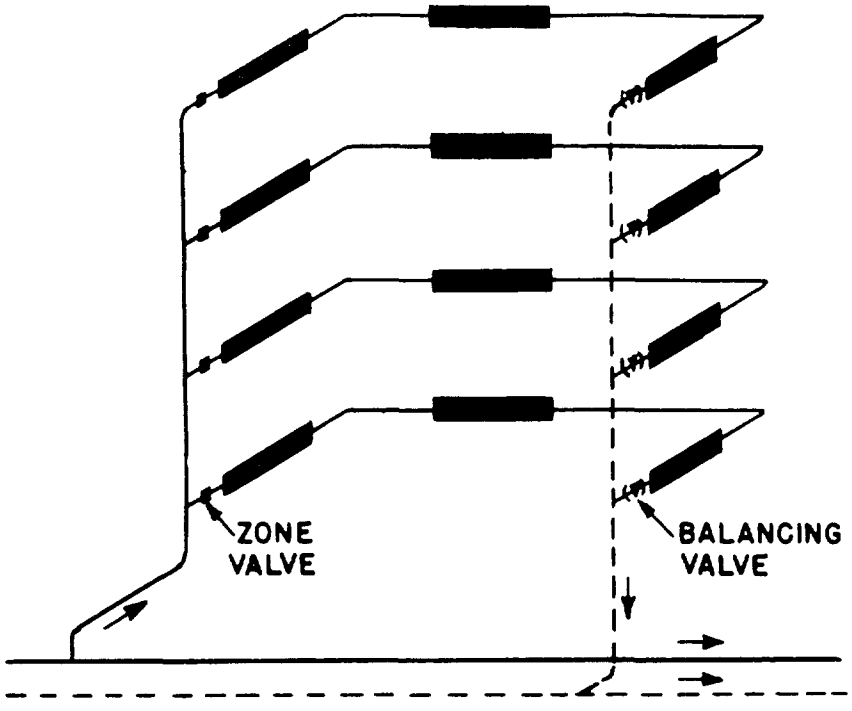


FIGURE 19-4

Figure 19-4 shows a combination of a two-pipe reverse return system in the basement, with a two-pipe direct return riser arrangement and series loop circuits for baseboard between the risers. This circuiting is adaptable for apartments.

Balancing valves are needed on each circuit to prevent short circuiting. When thermostatic control is required, zone valves on each circuit are used with the balancing valve.

HOT WATER—SIZING AND LOCATION OF PUMP

CHAPTER 20

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CHAPTER 20

HOT WATER -- SIZING AND LOCATION OF PUMP

Proper pipe-sizing in large hot water systems will help to guarantee these two important items in any heating system: economical installation and reliable, trouble-free operation.

In this chapter we will discuss only LTHW...low temperature hot water... defined by the American Society of Heating, Refrigerating and Air Conditioning Engineers as "any temperature up to 250°F."

Hydrostatic pressure...the pressure created at the base of a column of water by the weight of the water itself...is a consideration in hot water system design.

From Chapter 2 "Fundamentals of Pressure":

1 ft. of water	=	.433 lbs./sq. in
1 lb./sq. in.	=	2.31 ft. of water
1 lb./sq. in	=	27.72 in. of water

Example 1

An American-Standard boiler is to be installed in the basement of a six-story building. The highest piece or radiation is 65 feet above and 100 feet to the west of the boiler.

The lateral distance is not a consideration. Only the height of the water affects hydrostatic pressure.

Therefore, since 1 foot of water equals .433 psi, 65 feet of water equals:

65 x .433 psi or 27.1 psig

Example 2

In the same system, the fill line to the system is connected at the boiler. To what pressure must the reducing valve on the fill line be adjusted to provide a pressure of 5 psi in the topmost piece of radiation?

The pressure at the point of fill must equal the hydrostatic pressure plus the pressure at the topmost piece of radiation...or 27.1 psig + 5 psig... or 32.1 pounds per square inch gage. The reducing valve must, therefore, be adjusted to supply against a pressure of at least 32 psig.

Example 3

What theoretical height of system measured from the boiler to the highest piece of radiation will an American-Standard boiler fit when it is designed for 50 psig pressure when 5 psig pressure is required in the topmost radiator?

Deducting the radiation pressure from the total pressure, the boiler can withstand 45 psi in hydrostatic pressure.

Since 1 psi will support a column of water 2.31 feet high, 45 psi will support:

45 x 2.31 or 103.95 feet of water

Pumps

All large hot water systems are mechanically circulated by a pump. In heating systems the head -- or pressure -- of the pump is expressed in feet or the height, to which it will pump water in an open system.

Another way of saying it is that the feet-of-head of a pump is equal to the height of a column of water that creates a hydrostatic pressure equal to the difference between the water pressure at pump suction and pump discharge.

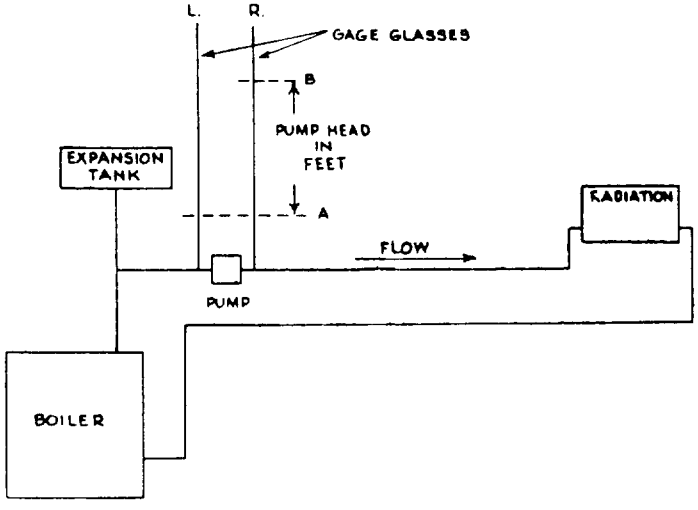


FIGURE 20-1

The difference in height between A and B, measured in feet, is the head of the pump.

Standard design practice today is to drive hot water heating system pumps by direct drive from the motor...sometimes with a flexible coupling between motor and pump shafts.

This means that pump speed is equal to motor speed. For any given motor the pump has one characteristic performance curve which plots the pump's ability to create a pressure in feet of head related to the number of gallons of water per minute circulated.

For any one pump, the pressure (feet of head) at which it pumps will decrease as the amount of water it pumps increases. This characteristic is common to all centrifugal pumping devices.

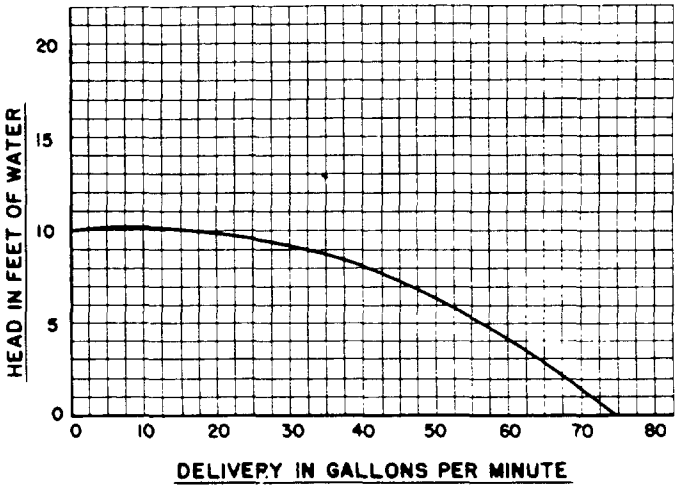


FIGURE 20-2

This is a typical pump characteristic curve. Assume that a pump with this performance curve is being considered for a job designed for a 20°F temperature drop with a 300,000 Btu/hr heat loss.

The minimum number of gallons per minute of water required is established by the equation:

$$\text{Btu/hr} = (\text{gpm} \times 60 \times 8\text{-}1/3) \times (1 \times 20)$$
$$= \text{gpm} \times 10{,}000$$

where:

Btu/hr	= heat load
gpm	= minimum gallons per minute required
60	= 60 minutes per hour
8-1/3	= 8-1/3 lbs. of water per gallon
1	= specific heat of water (Btu per pound per degree)
20	= 20°F temperature drop in system

therefore:

300,000	= gpm x 10,000
gpm	= 300,000 ÷ 10,000...or 30

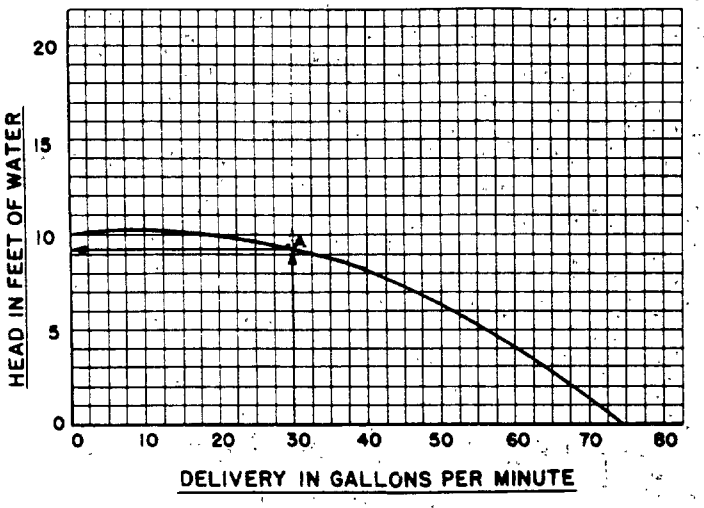


FIGURE 20-3

Now...entering the base line at 30 gallons per minute, a vertical line is erected until it meets the curve at A. A horizontal line from A, on the vertical scale, will show the feet of head available from this pump.

Small System Pipe Sizing

In small systems for residences and small commercial buildings, it is good practice to select the pump, determine the head it will deliver when circulating the minimum quantity of water, then size the piping against this head.

Since pipe-sizes are no larger than 1-1/2" and resulting water velocities are less than the critical 48" per second, this procedure works well.

Large System Pipe Sizing

On large systems, where pipe-sizes exceed 1-1/2 inches, piping is sized on the basis of pressure drop per foot of equivalent length at water velocities that will assure quiet flow.

A safe and practical rule of thumb is to assume that feet of equivalent length equals 1-1/2 times the feet of linear length.

Resistance in each pipe-size-segment of the system is calculated separately by multiplying the equivalent length in feet by the resistance per foot. They are all added together to establish the total resistance of the system.

The pump is then selected which will have a head equal to or greater than the resistance of the circuit while circulating the minimum water requirements of the system.

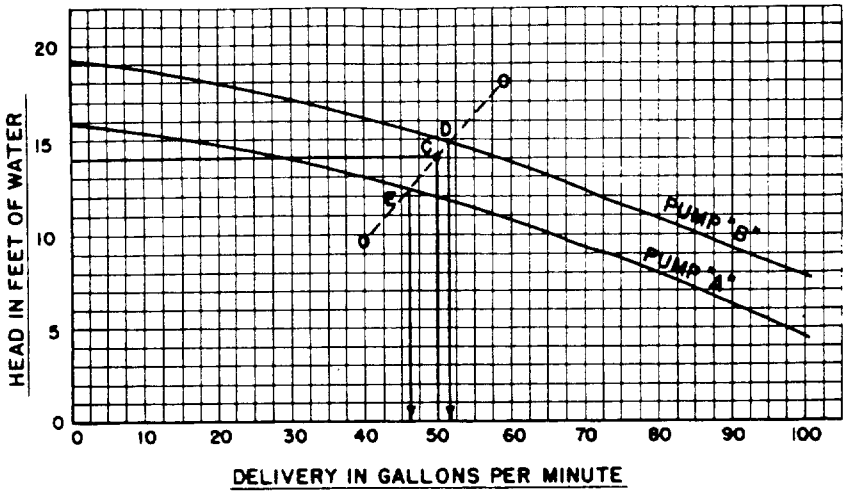


FIGURE 20-4

As a demonstration, assume a system requiring the circulation of 50 gpm against a friction of 14 feet of head. This point is plotted at point "C".

Friction head for other circulation requirements can then be determined by the formula: friction varies as the square of the velocity...or, as the square of gpm.

Dropping a vertical from "E", where the friction curve intersects the characteristic curve of pump A, indicates a circulation rate of only 46 gpm.

To be safe, we must specify pump B with a surplus capacity (52 gpm).

Actually, of course, in practical usage, it is only necessary to establish point "C". Always specify the next larger pump rather than an undersize.

The total resistance of the circuit includes friction loss through piping, fittings and the boiler.

Recent studies by pump manufacturers have demonstrated that placement of both pump and expansion tank on the supply side of the boiler is more desirable.

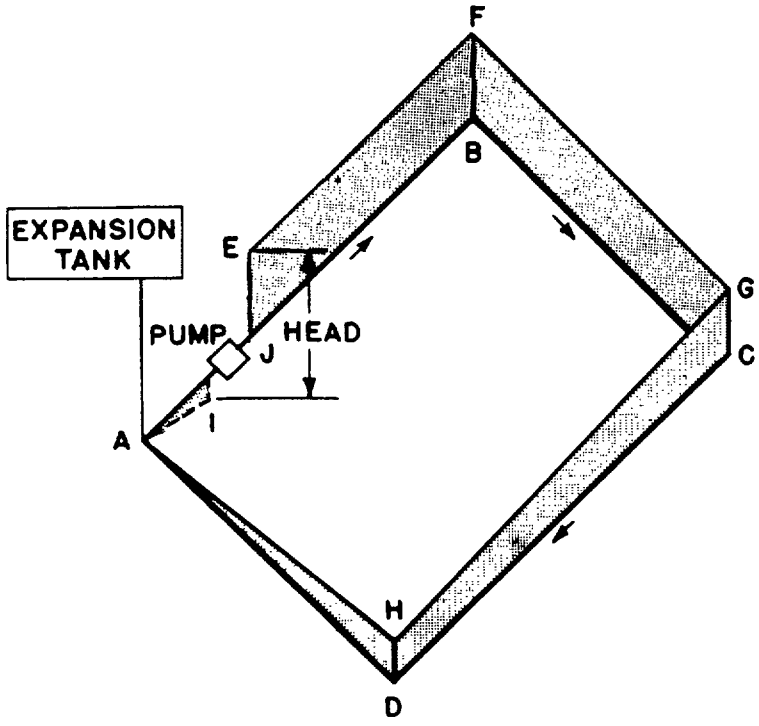


Figure 20-5

Figure 20-5 shows a schematic drawing of the pressures in a hot water system with both pump and expansion tank on the supply side of the boiler.

The solid line ABCD indicates the constant pressure level throughout the system with the pump idle.

When the pump starts up, there is a slight pressure drop at I, the pump intake, and a rise to E at pump discharge. The difference between E and I is the head of the pump.

The line JE shows the pressure increase in the system when the pump is running. From E to F the pressure decreases due to friction in the piping. This loss continues around the circuit through G, H and finally to A, where the pressure is equal to system pressure whether the pump is running or not.

With the pump in this location, the lowest pressure is at I -- the pump suction. Assuming a pressure of 15 psig with the pump not running, then pressure at I, because of friction, will be only slightly below 15 psig -- maybe 14.9 or 14.8 psig. Still well above the atmospheric pressure.

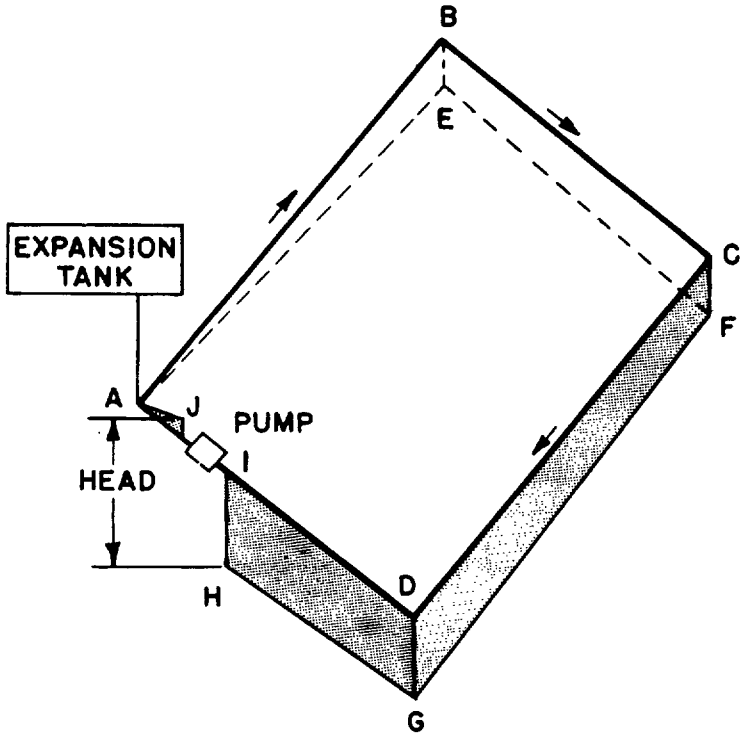


Figure 20-6

But! With the same pump on the return side, sucking water around the circuit instead of pushing it, the pressures change.

The solid line ABCD again represents system pressure with an idle pump.

A, at the expansion tank, is the constant pressure point in the system.

Again, the lowest pressure is at the suction end of the pump (H). Going backwards around the circuit, the pressure is higher at each succeeding point G, F and E because the pump is pulling rather than pushing and lowering pressure constantly around the circuit.

Note that pressures all around the circuit, now, are lower than the pressure in the system with the pump not working.

With a low head pump in the same 15 psig system, pressure H might be as low as 11.8 psig instead of the 14.9 or 14.8 psig with the pump on the supply side.

While 11.8 psig is still substantially above atmosphere and would operate the system satisfactorily, trouble would arise if:

1. The system static pressure fell too low
2. The pump had a high head

For either or both of these reasons, pressure at pump suction point H might fall too low. This situation could set up two problems:

1. If pressure fell below atmosphere, the pump might suck air into the system around the shaft seal which is designed to hold water but will let air in.

2. The pressure might fall low enough to cause the hot water to flash into steam...creating noise as well as damaging the pump.

The relation between pump and expansion tank location is not important normally in residential systems, because pump heads are low.

In large hot water systems with high pump heads, safer, more dependable and quieter operation demand that both pump and expansion tank both be located on the supply side of the boiler.

Except in noisy factories, piping is always designed for quiet operation, which means air-free water traveling through the system at less than 4 feet per second.

Table 20-1, showing gallons per minute, indicates design limits in pipe size and friction for quiet operation.

Gallons per minute can be converted to Btu/hr. by multiplying by 10,000 for a system designed for a 20° drop, 5,000 for a 10° drop.

Friction is expressed in milinches which are equivalent to a loss of pressure equal to a column of water 1/1,000ths of an inch high.

One foot of head, then, will overcome 12,000 milinches of friction.

TABLE 20-1

CARRYING CAPACITY OF PIPES IN GPM*
FOR BLACK IRON OR COPPER PIPE
GPM

Nominal Pipe Size Ins.	MILINCHES PER FOOT OF EQUIVALENT LENGTH																
	600	550	500	475	450	425	400	375	350	325	300	275	250	225	200	175	150
1/2	2.0	1.9	1.8	1.8	1.7	1.7	1.6	1.6	1.5	1.4	1.4	1.3	1.3	1.2	1.1	1.0	1.0
3/4	4.3	4.1	3.9	3.8	3.7	3.5	3.4	3.3	3.2	3.1	2.9	2.8	2.7	2.5	2.4	2.2	2.0
1	8.1	7.7	7.3	7.1	6.9	6.7	6.4	6.2	6.0	5.8	5.5	5.3	5.0	4.7	4.4	4.1	3.8
1-1/4	17.2	16.4	15.4	15.1	14.7	14.2	13.7	13.2	12.8	12.2	11.7	11.2	10.7	10.0	9.4	8.7	8.0
1-1/2	25.9	24.7	23.4	22.7	22.1	21.5	20.7	20.1	19.4	18.6	17.7	16.9	16.1	15.2	14.2	13.2	12.2
2	51.2	49.0	46.5	45.3	44.0	42.6	41.0	39.7	38.3	36.7	35.3	33.5	32.0	30.1	28.2	26.2	24.2
2-1/2	83.3	79.4	75.4	73.1	71.0	68.8	66.1	64.2	62.0	59.4	56.7	54.1	51.2	48.3	45.3	42.2	39.0
3	150	144	137	133	129	126	121	117	113	108	103	98.0	93.0	87.6	82.0	76.2	70.5
3-1/2	226	215	203	198	193	187	180	175	168	161	153	147	139	131	123	114	105
4	320	303	289	280	271	262	253	245	236	226	216	206	195	184	173	160	147
5	595	567	537	521	505	490	472	456	440	421	402	384	363	343	322	298	274
6	992	945	898	872	845	820	790	764	735	705	672	642	610	575	539	499	459

* Select above heavy lines for water velocities less than 48 inches per second. Selection below heavy lines will provide noisy water flow. For values for 5 and 6 inch pipe in quiet zone, refer to ASHRAE GUIDE.

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Consider a 2" pipe. The table indicates that it can handle 41 gpm at a milinch friction of 400 per foot of length. If it is asked to carry 42.6 gpm, milinches of friction rise to 425 per foot of length and water velocity exceeds the permissible limit of 48 ft/sec.

Similarly, the safe, quiet loading of a 2-1/2" pipe is 59.4 gpm; for a 3" pipe, 93 gpm.

Note that as pipe size increases, allowable friction for quiet operation diminishes.

This table can be used to specify pump size. For instance, the system will use 265 feet of pipe. By rule of thumb we multiply this by 1-1/2 to get 397 -- or 400 feet of equivalent length.

Requirements are for 40 gallons per minute. The smallest pipe that will carry 40 gpm is 2" and it will set up 400 milinches of friction per foot.

400 milinches X 400 feet - 160,000 milinches of friction. 160,000 ÷ 12,000 - 13.3 feet. A pump with at least 13.3 feet of head is required, for a circulation of 40 gpm.

Two or more pumps in series in the same circuit are not frequently used. Since their heads are additive, the effect on the system is that of a single large pump having a head equal to the sum of the heads of the two smaller ones.

When the quantity of water to be circulated is small and the head requirements are high, two small pumps in series may be more economical to install than one large one.

A small but growing practice uses two pumps in series but in separate zones or circuits of the same system. This arrangement is called primary and secondary pumping.

Primary and secondary pumping for zone control is relatively new.

It has certain advantages.

1. A better degree of control is achieved than with motorized valves.
2. Pump and pipe sizes can be reduced by allowing for greater temperature drops in the primary circuit.
3. Hydraulic and heat balance is achieved with economical operation.
4. Total pumping horsepower is usually less.
5. Continuous water circulation with modulated temperatures and heat delivery to the zone is possible.
6. Installation costs are frequently less than other control systems.

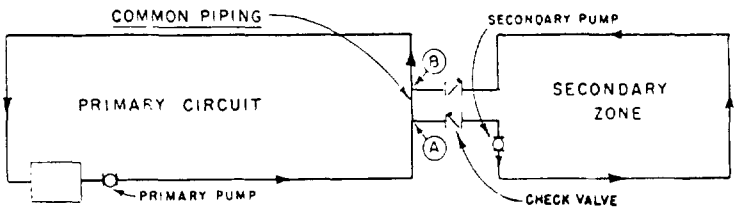


FIGURE 20-7

Figure 20-7 illustrates a basic piping arrangement for a system using primary and secondary pumping.

Its principal of operation is simple. When a secondary circuit is connected to a primary circuit, such as in a one-pipe system, the resistance created in the primary circuit between the supply and return tees will cause a water flow in the secondary circuit. However, when two ordinary supply and return tees are placed close together on the primary circuit, the resistance becomes so small between them as to cause almost zero circulation in the secondary piping.

With this piping arrangement, pumps may be selected to overcome the resistance of each of the two circuits.

The primary pump operates continuously. The secondary pump is put under thermostatic control to supply heat to the zone needed.

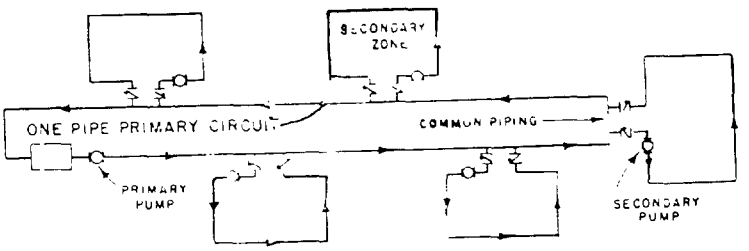


FIGURE 20-8

Figure 20-8 illustrates the development of multiple zones from the basic diagram in Figure 20-7.

The primary circuit performs similarly to a one-pipe system and is normally designed for a 20°F temperature drop. When temperature drops of more than 20°F are used, the radiation must be selected on the basis of the average water temperature available at each zone.

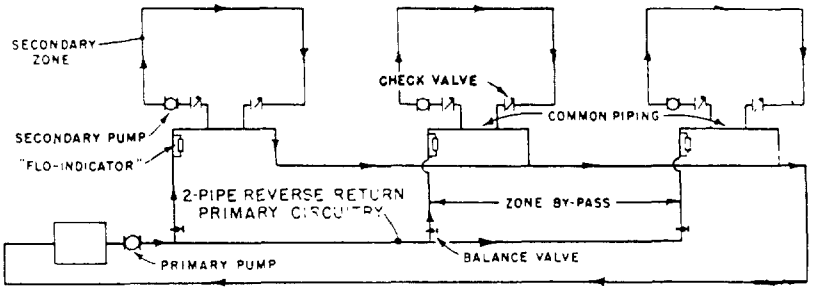


FIGURE 20-9

Primary and secondary pumping may be used with either a two-pipe direct return or a two pipe reverse return primary circuit. The reverse return arrangement is preferred as shown in Figure 20-9.

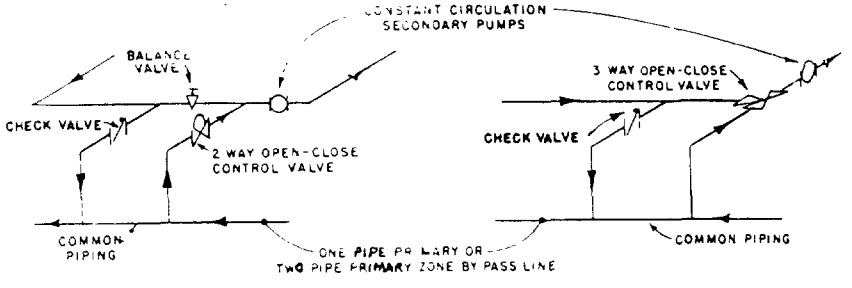


FIGURE 20-10

FIGURE 20-10A

In all of the previously described arrangements the secondary pump is operated intermittently.

Figure 20-10 illustrates an arrangement for continuous pump operation in the zone with a two-way (open - close) valve, thermostatically controlled, that admits hot water from the primary circuit. The zone still gets heat in shots.

An identical effect can be obtained with a three-way (open - close) valve. See Figure 20-10A.

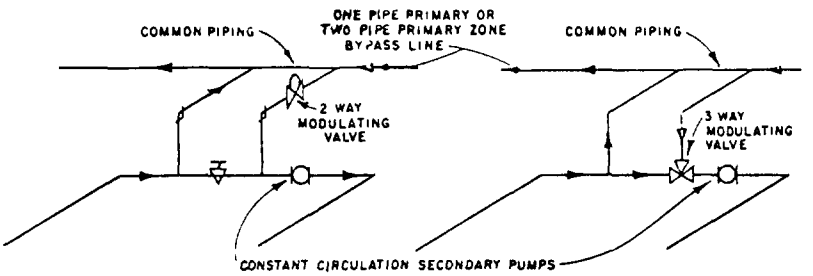


FIGURE 20-11

FIGURE 20-11A

When modulated water temperatures are desired for continuous heat delivery to the zone, the piping arrangement illustrated in Figures 20-11 and 20-11A are required. Modulating valves, either two-way or three-way, are under thermostatic control.

HOT WATER—
SIZING THE EXPANSION TANK

CHAPTER 21

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CHAPTER 21

HOT WATER -- SIZING THE EXPANSION TANK

Tanks are necessary in hot water heating systems to allow for water volume expansion as it is heated. Economical sizing of this component as well as others is important to the sale.

Originally, open expansion tanks were used above the highest piece of radiation. They allowed for water expansion but had no effect on system pressure.

Modern tanks are closed...containing about 60% water and 40% air. As system water expands it moves into the tank, compressing the air cushion and raising the system pressure.

Water logged tanks...insufficient air...cause too great a rise in system pressure...spitting by the relief valve and the necessity for replacing the lost water.

Factors governing the sizing of expansion tanks are the same for all systems. They are:

1. The system volume of water
2. Maximum average water temperature
3. The fill pressure of the system
4. Tank location in the system

However, small system tank sizing is commonly accomplished by allowing one gallon of tank capacity for each 5,000 Btu/hr of heat loss.

This "rule of thumb" does not work for large systems because of the greater importance of the four factors listed above.

In the guide published by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, tank volume is determined from the ASME equation:

$$V_t = \frac{(.00041t - .0466)}{\frac{Pa}{Pt} - \frac{Pa}{Po}} \times V_s$$

This equation assumes expansion tank temperature of no greater than 70°F.

where:

- V_t = Minimum gallonage of the tank
- V_s = System volume in gallons
- t = Maximum average water temperature in F°
- P_a = Absolute pressure in tank when water first enters... usually atmospheric (14.7 psi at sea level)
- P_t = Absolute pressure in tank created by initial fill... psig fill pressure plus atmospheric
- P_o = Maximum absolute operating pressure...maximum psig pressure at tank plus atmospheric

To develop a similar working formula we will substitute "C" for the first factor in the right hand side of the equation...for sea level conditions.

$$V_t = C \times V_s$$

$$\text{where } C = \frac{.00041t - .0466}{\frac{P_a}{P_t} - \frac{P_a}{P_o}}$$

Let's solve for C:

$t = 230^{\circ}F$

$P_a = 14.7$ psi absolute...atmospheric pressure

$P_t = 26.7$ psi absolute...12 psig initial fill pressure

$P_o = 59.7$ psi absolute...45 psig maximum operating pressure

then $$C = \frac{.00041t \times 230 - .0466}{\frac{14.7}{26.7} - \frac{14.7}{59.7}} = .157$$

then...if:

$$V_t = C \times V_s$$

and...the specified system has a volume of 960 gallons

$V_t = .157 \times 960$ or 150.7 gallons

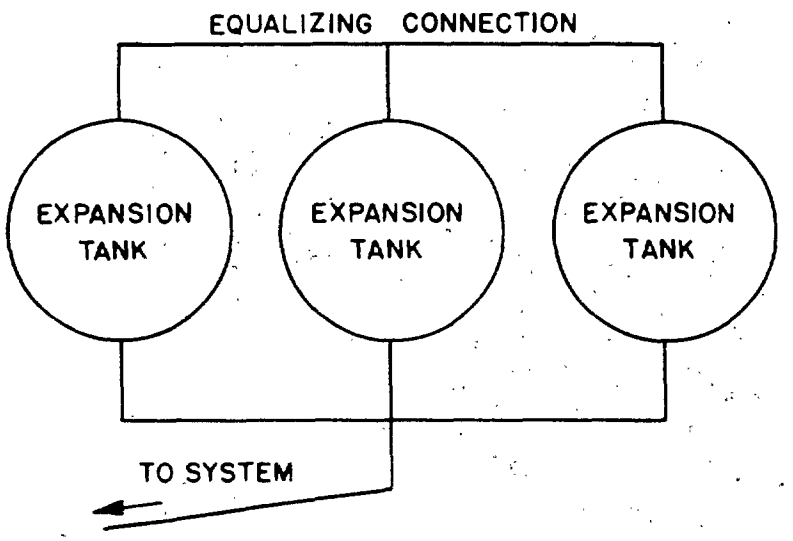


FIGURE 21-1

To supply an expansion tank volume of 150.7 gallons, it would be common to pipe three tanks in parallel.

The tables in this chapter will supply the information necessary to easily and readily determine expansion tank size.

Tables 21-2 through 21-6 specify gallonage in different segments of the system. Simple addition will establish V_s...total gallonage in the system.

Table 21-7 provides factors for C under varying pressures and temperatures. Simple multiplication of the established system volume by this factor will determine the minimum size of the expansion tank.

When expansion tanks are located in an area where the temperature may become considerably more than 70°F, the ASME equation must be corrected for the following reasons:

1. The air cushion becomes hotter. Since the air has no place to expand, its pressure increases.

2. The hotter water in the expansion tank causes additional vapor or steam to mix with the air cushion. This action increases the pressure on the water in the tank and, hence, the system pressure.

The corrected factor C becomes:

$$C = \frac{.00041t - .0466}{\frac{P_a}{P_t - P_{vf}} - \frac{T_t}{T_f} \times \frac{P_a}{P_o - P_{vt}}}$$

where:

t = Maximum average water temperature in degrees F.

P_a = Absolute pressure in tank when water first enters...usually atmospheric

P_t = Absolute pressure in tank created by initial fill.

P_o = Maximum absolute operating pressure.

P_{vf} = Absolute vapor pressure in psi at the fill temperature.

P_{vt} = Absolute vapor pressure in psi at the tank operating temperature.

T_t = Absolute temperature of the tank at operating conditions... 460° plus degrees F.

T_f = Absolute temperature of the tank at fill conditions...460° plus degrees F.

To illustrate the effect of a heated tank, it is assumed the system was filled at 40°F. With the system in operation the tank temperature gets to 125°F. All other conditions remain the same as in the example immediately preceding.

Solving for C

t = 230°F

P_a = 14.7 psi absolute--atmospheric pressure

P_t = 26.7 psi absolute--12 psig initial fill pressure

P_o = 59.7 psi absolute--45 psig maximum operating pressure

P_{vf} = .122 psi absolute at 40°F tank temperature. See Table 21-1

P_{vt} = 1.942 psi absolute at 125°F tank temperature. See Table 21-1

T_t = 585° Absolute--460° + 125° = 585°

T_f = 500° Absolute--460° + 40° = 500°

$$C = \frac{.00041 \times 230 - .0466}{\frac{14.7}{(26.7 - .122)} - \frac{585}{580} \times \frac{14.7}{59.7 - 1.942}} = .186$$

Therefore under these conditions

$V_t = .186 \times V_s$

for a specified system volume of 960 gallons

$V_t = .186 \times .960 = 179$ gallons

Because the tank operates at a temperature of 125°F -- all other conditions in the two examples remaining the same -- its volume must be increased 19%.

Warning

This percentage increase applies only to the conditions in the examples and should not be used for other operating or fill pressures.

To illustrate the use of the tables a three-story garden type apartment with a boiler in the basement requires 390,000 Btu/hr. The linear feet of N85-L Heatrim installed is 550.

Linear feet of piping in the installation is:

2" pipe --- 50 ft.
1-1/2" pipe ---400 ft.
1" pipe ---200 ft.
3/4" tubing---200 ft. (not including radiation)

Boiler used is a G-4010

Volume of water in system

Boiler	-	77.4	gal.	(See table	21-2)
50 ft. of 2" pipe	-	8.7	"	" "	21-6
400 ft. of 1-1/2" pipe	-	42.4	"	" "	21-6
200 ft. of 1" pipe	-	9.0	"	" "	21-6
200 ft. of 3/4" tubing	-	5.0	"	" "	21-6
550 ft. of 3/4" Heatrim	-	14.8	"	" "	21-4
TOTAL		157.3			

Assuming expansion tank located in a cool space in the basement and allowing 10 feet between each floor of the apartment building, the radiation on the third floor will be approximately 22 feet above the expansion tank. Using the line showing a minimum height of the system above the tank of 27.7 feet (see Table 21-7) with the corresponsing fill pressure of 16 psig and an average system temperature of 210°F, a constant "C" is found to be .16. The expansion tank volume is 16% of the system volume. Therefore the tank volume is:

V_t = .16 x 157.3 gallons

= 25 gallons

Generally, in large systems, expansion tanks are located in a reasonably cool area making Table 21-7 valid.

Should the tank or tanks be located over the boiler, where the heat rising from the boiler will warm the tank, a correction of tank volume for a temperature of 125°F should be adequate.

Table 21-8 has been prepared to provide correction factors. These factors are applicable only to Table 21-7. They are used as multipliers.

Example

The value of C used in the problem solved on page 21-5 was .160 assuming the tank was located in a space of 70°F or less. If the tank had to be located immediately over the boiler, a new constant "C" must be obtained. At 16 psig fill pressure the correction multiplier from Table 21-8 is found to be 1.20.

Therefore the new value of C equals .16 x 1.20 or .192 and the tank volume for the system in the example becomes

157.3 x .192 or 30 gallons

Table 21-1

Tank Temperature Degrees F	Vapor Pressure psi absolute
40	.122
45	.148
50	.178
55	.214
60	.256
65	.306
70	.363
75	.430
80	.507
85	.596
90	.698
95	.815
100	.949
105	1.102
110	1.275
115	1.471
120	1.692
125	1.942
130	2.222
135	2.537

Table 21-2

VOLUME OF WATER IN GALLONS - AMERICAN-STANDARD HOT WATER BOILERS

Sections	G-40	G-60	A-5	A-5
3	26.6			
4	33.8		57.5	
5	41.1		64.0	151
6	48.4	91.5	75.5	167
7	55.5	105.5	84.0	189
8	62.9	119.0	93.0	211
9	70.1	133.0	102.0	234
10	77.4	146.5	111.0	256
11		160.5	118.5	278
13		225.0		302
15		253.0		
17		280.0		
19		308.0		
21		336.0		
22		387.0		
25		427.0		
28		470.0		
31		510.0		
33		575.0		
37		630.0		
41		685.0		

Table 21-3

VOLUME OF WATER
IN
CAST IRON RADIATORS AND BASEBOARD

Thin Tube (Arco Radiators) --- 8.4 gallons per 100 sq. ft. (EDR)

Large Tube Radiators --------- 15.25 " " " " " "

Sunrad -- 5 x 20 ------------ 9.00* " " " " " "

-- 7-1/2 x 23 --------- 10.00* " " " " " "

Radiantrim

New Model 8 ------------ 18.0 Gallons per 100 linear feet

Model 10 ------------ 24.0 " " " " "

* Applies to radiators of 10 sections or larger.
For radiators less than 10 sections, multiply indicated capacities by 1.5.

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Table 21-4

VOLUME OF WATER
IN
FINNED TUBE RADIATION

Heatrim

N50	Element	1.32	gallons per 100 linear feet
N85	"	2.69	" " " " "
N110	"	4.42	" " " " "

Temtrim or Similar
Finned-tube Radiation

2"	Steel tube element	17.4	gallons per 100 linear feet
1½"	" " "	7.8	" " " " "
1¼"	" " "	6.8	" " " " "

Table 21- 5

VOLUME OF WATER
IN
CONVECTORS

Multifin

#4	Element	.24	gallons per 100 inches*
#6	"	.43	gallons per 100 inches
#8	"	.67	" " " "
#10	"	1/12	" " " "

Arco

#4	Element	1.87	gallons per 100 inches*
#6	"	2.89	gallons per 100 inches
#8	"	3.84	" " " "
#10	"	4.85	" " " "

* Inches of length taken as length of cabinet

Table 21-6

VOLUME OF WATER
IN
STANDARD PIPE AND TUBE

Pipe Size	Gallons per 100 linear feet	
	Std. Steel Pipe	Copper tube Type L
1/2	1.57	1.21
3/4	2.77	2.51
1	4.49	4.29
1-1/4	7.79	6.53
1-1/2	10.6	9.24
2	17.4	16.1
2-1/2	24.9	24.8
3	38.4	35.4
3-1/2	51.4	47.9
4	66.1	62.2
5	104.0	97.0
6	150.0	13.9
8	266.0	24.3
10	419.0	37.8

Table 21-7

Fill Press. (psig)	Max. ht. of system above Tank (ft)	Value of Constant C (A) Max. temp. 230°F	Value of Constant C (B) Max. temp. 210°F
12	18.4	.148	.123
16	27.7	.193	.160
20	37.9	.242	.201
24	46.2	.312	.259
28	55.3	.408	.338
32	64.7	.542	.450

NOTE:

1. For this table to apply, tank temperatures should be no more than 70°F.
2. This table is based upon a relief valve setting of 50 psig. Smaller size tanks result.
3. Maximum temperature equals maximum system average temperature.
4. For multiple story structures of three or more floors use column B (Max. temperature of 210°F).
5. Fill pressure determined by maximum height of system above fill points - 4 psig allowed in highest piece of radiation.

Table 21-8

Multiplication Applicable only to Table 21-7

Fill Pressure psig	Correction Factor
12	1.55
16	1.20
20	1.30
24	1.42
28	1.65
32	2.14

CHAPTER 22

HOT WATER BOILER MANIFOLD SIZING

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CHAPTER 22

HOT WATER BOILER MANIFOLD SIZING

As in sizing steam boiler manifolds (Chapter 18) it is necessary to restrict water velocity to insure quiet flow.

Unlike steam manifolds, there is no fixed velocity limit in manifolds. For pipes two inches and smaller, water flow should be limited to 4 feet per second, while velocities up to 10 feet per second are frequently used in 6-inch or larger pipes. It should be kept in mind that in large systems the mains with the high water velocities are a considerable distance from the radiation where water velocity noise would be objectionable.

To arrive at the proper manifold size with hot water boilers, the basic equation is:

$$\text{Volume} = \text{Area} \times \text{Velocity}$$

or

$$\text{Area} = \frac{\text{Volume}}{\text{Velocity}}$$

where

Area is expressed in square inches
Volume is expressed in cubic inches per second
Velocity is expressed in inches per second

In low temperature hot water heating, the volume is based upon a 20°F temperature drop of the system.

The weight of one cubic foot of water is taken as 62.5 pounds. When cooled 20°F it gives up 62.5 X 1 X 20 or 1250 Btu.

Volume in cubic feet per hour

$$= \frac{\text{Btu per hour}}{1250} \quad \text{or} \quad \frac{\text{Mbh}}{1.25}$$

Volume in cubic inches per hour

$$= \frac{\text{Mbh} \times \text{cubic inches per cubic foot}}{1.25}$$

$$= \frac{\text{Mbh} \times 1728}{1.25}$$

Volume in cubic inches per second

$$= \frac{\text{Mbh} \times 1728}{1.25} \div \text{seconds per hour}$$

$$= \frac{\text{Mbh} \times 1728}{1.25 \times 3600} = \frac{\text{Mbh}}{2.6}$$

Velocity in inches per second equals velocity in feet per second X 12.

Therefore, the equation for cross sectional area in square inches becomes:

$$\text{Area} = \frac{\text{Mbh}}{2.6} \div 12 \times \text{Velocity (ft/sec)}$$

$$= \frac{\text{Mbh}}{2.6 \times 12 \times \text{Velocity}}$$

$$= \frac{\text{Mbh}}{32 \times \text{Velocity}}$$

or

$$\text{Velocity (ft/sec)} = \frac{\text{Mbh}}{32 \times \text{Area (sq. in.)}}$$

Example:

Calculate the velocity necessary to supply 336,000 Btu through a 2-inch pipe, which has an area of 3.356 square inches.

$$\text{Velocity} = \frac{336}{3.356 \times 32} = \frac{336}{107.4} = 3.13 \text{ ft/second}$$

Table 22-1 has been prepared to aid in selecting manifold pipe sizes on hot water boilers.

It may be used as follows.

Example No. 1

The G-6022 and G-6028 boilers have respective outputs of 2,436,000 Btu/hr and 3,132,000 Btu/hr. Minimum piping recommendations for these two boilers, as shown in the Technical Catalog Bulletin, specify two 4 inch supply risers and a 6 inch header. Under conditions of maximum output, what water velocities will be obtained?

The G-6022

Load per riser = $\frac{2,436,000}{2}$ = 1,218,000 Btu/hr.

Referring to Table 22-1 on the 4 inch pipe size line, it is found that a load of 1,218,000 Btu/hr requires a velocity of 3 feet per second.

The 6 inch header, after it has picked up the entire boiler capacity of 2,436,000 Btu/hr is working at velocity of less than 3 feet per second.

The G-6028

Load per riser = $\frac{3,132,000}{2}$ = 1,566,000 btu/hr.

Referring to Table 22-1 on the 4 inch pipe size line it is found that a 4 inch pipe with 3.5 feet per second velocity is good for 1,425,000 btu/hr. and at 4 feet per second the capacity is 1,630,000 btu/hr. Obviously, the water velocity is less than 4 feet per second and more than 3.5 feet per second. By interpolation the actual velocity is shown to be approximately 3.8 feet per second.

The velocity in the 6 inch header after it has picked up the entire boiler capacity of 3,132,000 btu/hr is slightly less than 3.5 feet per second.

Velocities in the return header and risers are identical.

Example No. 2

The A-710-WO boiler has a gross I=B=R output of 1,760,000 btu/hr. It is supplied with four 4" flow tappings and two 6 inch return tappings. To carry the full capacity of the boiler, how many supply risers and return risers will be required and what manifold size is needed?

Referring to Table 22-1 it is found that a 4 inch riser with a velocity of 4 feet per second is capable of handling only 1,630,000 btu/hr. Obviously, it is desirable to use two supply risers instead of one.

The load per riser = $\frac{1,760,000}{2}$ = 880,000 btu/hr. From the table it is found that a 3 inch supply riser is good for the 880,000 btu/hr. at a velocity about halfway between 3.5 and 4.0 feet per second, which is entirely satisfactory.

The header carrying the full load of 1,760,000 btu/hr. must be 5 inches since 4 inch pipe is too small. The velocity in the header will be less than 3 feet per second, again a satisfactory condition.

Return risers and manifold will be sized identically.

Tappings selected for both supply and return risers must be chosen to provide good distribution of water to the boiler.

It is suggested that the F and first CX sections be used for the flow risers. The returns should be divided between the two 6 inch return tappings (bushed to 3 inches) in the F section.

TABLE 22-1

Btu/Hr. Capacity of Steel Pipe
for
20°F System Temperature Drop

Pipe Size Inches	I. D. Cross-Section Pipe Area	Flow Rate -- Ft / Sec		
		3.0	3.5	4.0
1/2	.304	29,200	34,000	39,900
3/4	.533	52,000	59,600	58,100
1	.8413	80,700	94,000	107,800
1-1/4	1.496	143,500	167,200	191,000
1-1/2	2.036	195,500	228,000	260,000
2	3.356	322,000	376,000	429,000
2-1/2	4.788	460,000	535,000	612,000
3	7.393	710,000	827,000	945,000
3-1/2	9.885	950,000	1,108,000	1,265,000
4	12.730	1,220,000	1,425,000	1,630,000
5	20.006	1,925,000	2,240,000	2,560,000
6	28.830	2,770,000	3,230,000	3,820,000
8	51.162	4,910,000	5,730,000	6,540,000
10	81.585	7,840,000	9,130,000	10,430,000

CHAPTER 23

CONVERTORS

CHAPTER 23

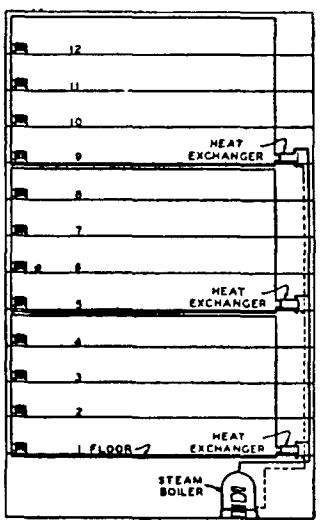
CONVERTORS

A convertor is a heat exchanger that is used specifically to transfer heat from steam to water. These units are employed when:

1. A hot water heating system is connected to a steam system with the steam boilers being the initial source of the heat.

2. Steam is purchased from a district steam system but hot water heating is desired.

3. Hot water heating is desired in tall buildings, where the hydrostatic pressure created by the water would become too severe on the piping and radiation at the lower levels.



In Figure 23-1 the method for supplying hot water heating in tall buildings is illustrated. Here the solution is to use one steam boiler in the basement to generate the steam necessary to operate a convertor for each group of four floors. Each convertor acts as a hot water boiler supplying heat to a zoned four-story segment of the building. Hydrostatic pressures are brought within practical limits, and only one low-pressure steam boiler is necessary even though it operates three separate and distinct heating systems.

FIGURE 23-1

The small individual systems thus created use relatively smaller piping and less costly pumps and accessories.

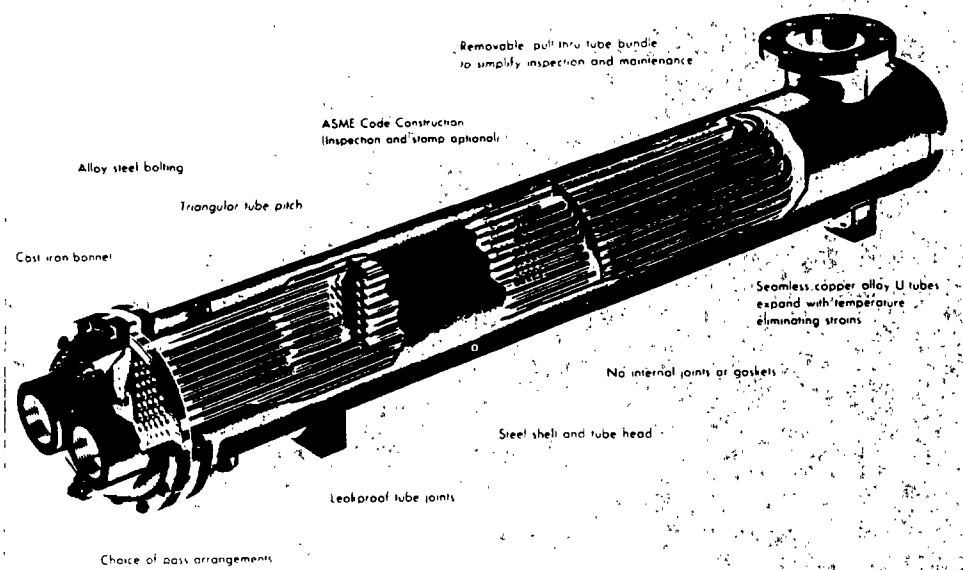


FIGURE 23-2

Figure 23-2 shows a convertor as manufactured by the Industrial Division of American-Standard. The steam enters the top of the shell at the right and fills the space around the water tubes. The water flowing through the tubes becomes heated and the condensate formed from the condensed steam drains off from the bottom of shell at the left hand end. The water enters one side of the cast iron bonnet flowing through the U-tube bundle and returns to the other side. This flow arrangement is called a two-pass. By rearranging the spacing of the tubes and the partitions cast into the bonnet, a four-pass flow can be obtained. The two-pass flow is advantageous to use because the pressure drop of the water through the heater is less than for the four-pass arrangement. The U-tube design with the tubes fastened to a tube sheet at one end only of the unit permits a free expansion of the tube bundle. With a steam to water heat exchanger, baffling is not as much a problem as with the water to water type. The round plates seen in the illustration serve to support and hold the tubes apart so the steam can completely surround them.

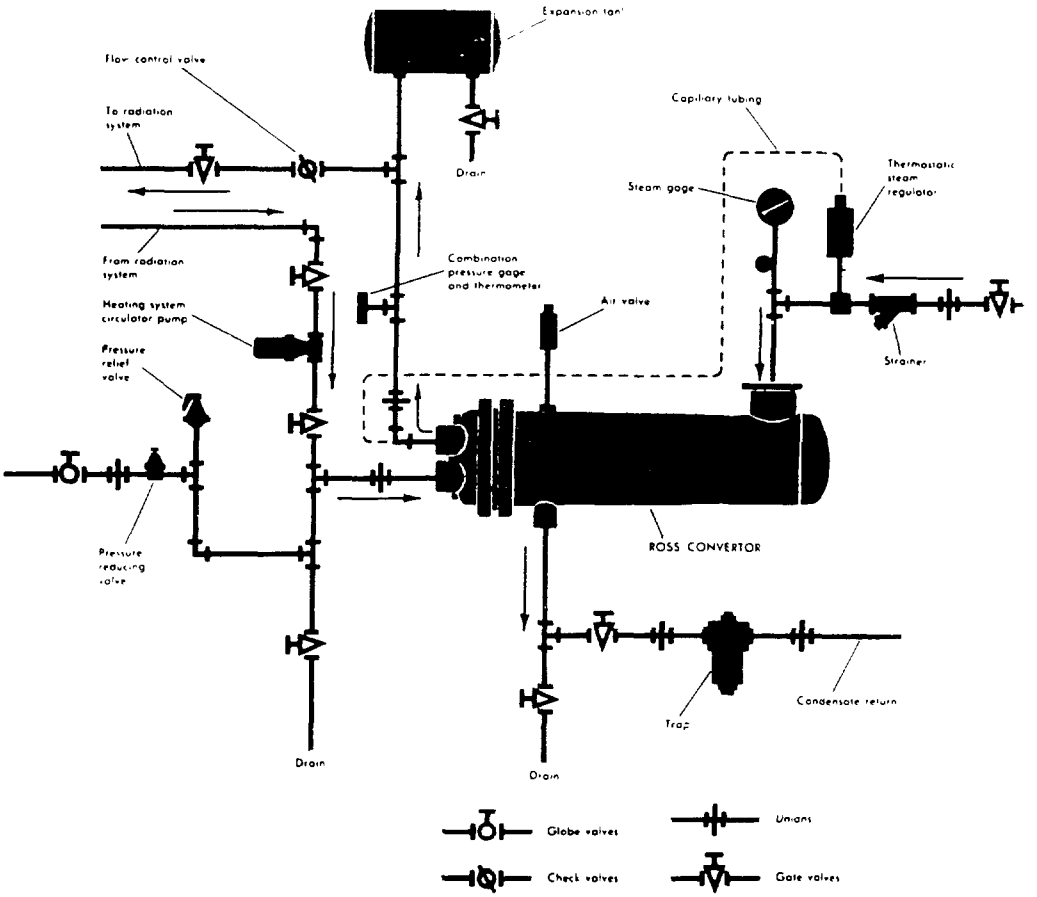


FIGURE 23-3

Figure 23-3 illustrates a convertor piping arrangement which is self-explanatory. In this illustration the thermostatically controlled steam regulator holds the water leaving the convertor at some fixed temperature, in the same manner as a boiler operating on a fixed water temperature. To control the space temperature, the pump could be cycled by the space thermostat or a three-way valve, thermostatic control by either indoor or outdoor thermostat can be provided to modulate the water temperature to the space. In this last instance, the pump would operate continuously.

Sizing the piping for the system would follow conventional engineered design. The heat loss of the space served by the steam is determined and a decision made to design for either a 20°F or 10°F temperature drop. The gallons of water to be circulated is then determined. If the g.p.m. are large, pipe sizes will be selected and the resistance of the piping and fittings determined. To this valve will be added the resistance of the water flowing through the convertor. A pump is then selected that has a head equal to, or greater than, the total resistance of the system.

CHAPTER 24

SIZING THE BOILER

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CHAPTER 24

SIZING THE BOILER

To size a boiler accurately for either steam or hot water, its capacity must be equal to, or greater than, the load. Either a net installed radiation load or a gross load may be estimated.

These requirements can be any or all of the following defined loads:

1. Radiation Load -- the estimated heat emission in square feet, E. D. R. or Btu per hour, of the connected radiation.

2. Piping Load -- the estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler. A close approximation for this can be considered at 15% of the net load when 1" magnesia pipe insulation is used.

3. Pick-up Load -- the estimated load in Btu per hour required to bring the space up to daytime temperature following a nighttime setback of the thermostat. This load can also be considered as 15% of the net radiation load.

4. Miscellaneous Load -- any load other than those listed above such as domestic hot water, snow removal systems, industrial processes, etc.

In residential application it is common practice to size the boiler on the net installed radiation load, since allowances have been built into the net rating of the boiler to allow for pipe loss, pick-up and the heating of a nominal amount of domestic hot water.

Where abnormal conditions exist, such as a heavy domestic hot water demand or excessive piping losses, the piping, pick-up, miscellaneous and radiation loads should be added to arrive at the total gross load requirement. The gross capacity or output of the boiler selected must be equal to or greater than the gross load.

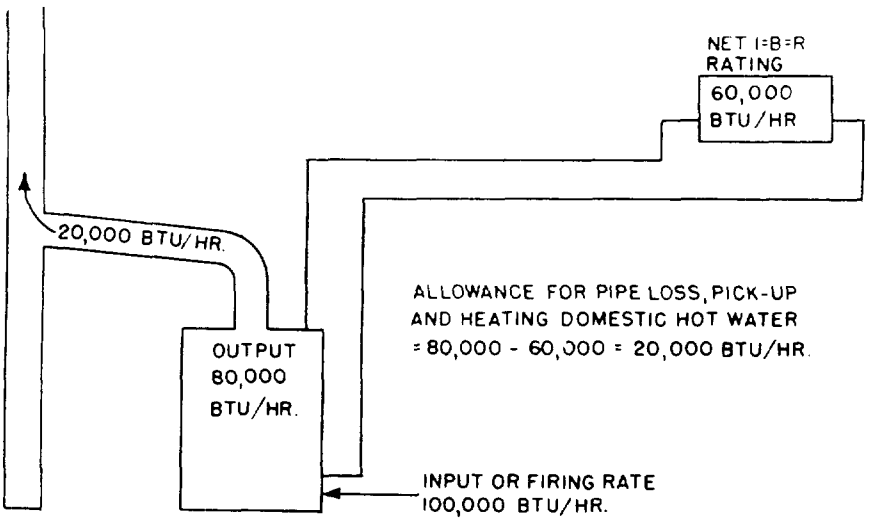


FIGURE 24-1

Referring to Figure 24-1, it is seen that only a fraction of the heat input to a boiler gets into the boiler water and becomes the boiler's gross capacity or output. In gas boilers that are rated by the American Gas Association this fraction is 0.8 -- this figure also says that the boiler efficiency is 80%. In oil boilers the fraction varies. It may be 0.7 to 0.75 depending on the boiler and the burner adjustment.

The I=B=R net installed radiation capacity of the boiler is about 75% of its output or gross capacity.

It is noted that while some of the btu/hr. difference between input and output is lost from the boiler jacket, the major portion of this heat quantity goes into the chimney.

From Figure 24-1 it is evident that the 20,000 Btu/hr. allowed for pick-up, pipe loss and the heating of domestic hot water is about 33-1/3% of the net I=B=R rating.

This allowance is divided approximately as follows:

Pipe Loss----------	15%
Pick-up----------	15%
Domestic Hot Water---	3-1/3%
TOTAL	33-1/3%

Example No. 1

For examples of these various conditions, let's assume a hypothetical modernization job involving a boiler replacement. Our problem will be to size a steam boiler in an old four story home of an unknown heat loss. The boiler must be replaced and the owner is desirous of changing from stoker-firing to oil. When the existing boiler is examined, it is found to be an American-Standard Ideal Water Tube S-36-T-7 with a net steam rating of 2,070 square feet.

It was common practice years ago to oversize the boiler to insure adequate grate area and sufficient capacity for quick pick-up after long shut-down periods. A further examination disclosed that, by actual count of radiators, only 1,470 square feet of radiation or 352,000 Btu per hour was connected to the boiler. Piping and pick-up loss was considered normal because all piping was insulated with 1" magnesia insulation.

Based on both loads indicated above, a decision was required to select a size between an A-509S boiler which is equal to or greater than the old boiler size, or an A-507S which was equal to or greater than the net load requirements of the system.

Since the owners indicated that the existing radiation maintained good comfort conditions during the 25 years of operation, satisfying the radiation load of 1,470 square feet with an A-507SB boiler is all that is required. The maximum requirements for domestic hot water are estimated to be 34 gallons per hour.

To determine the adequacy of the boiler selected, an analysis may be made as follows:

Gross Capacity of the A-507SB	=		508,000 Btu/hr.
Net Load (1,470 sq. ft.)	=	352,800 Btu/hr.	
Pick-up Load - 15% of 352,800	=	52,920 Btu/hr.	
Pipe-loss - 15% of 352,800	=	52,920 Btu/hr.	
* Domestic Hot Water (34 gph x 833)	=	28,320 Btu/hr.	
Total Gross Load	=		486,960 Btu/hr.
Surplus Boiler Capacity	=		21,040 Btu/hr.

*NOTE: 1 gallon of water heated 100°F requires 8-1/3 x 1 x 100 or 833 Btu.

Example No. 2

If this same hypothetical house could be made into a multiple apartment structure by merely inserting partitions and adding bathrooms, without necessitating any major change in the heating system outside of an isolated relocation of a radiator, then a parallel can be made to the original example, and an illustration of how increased water heater requirements can change the boiler size.

Let's assume the house can be converted into a total of 16 apartments, one bedroom with single bath and kitchenette for each one. The total domestic hot water demand is estimated at 153 gallons per hour.

To size the boiler for this example a similar analysis to the one made for Example No. 1 is in order.

Installed Radiation (1,470 sq. ft.)	=	352,800 Btu/hr.
Pipe Loss (15% of 352,800)	=	52,920 Btu/hr.
Pick-up (15% of 352,800)	=	52,920 Btu/hr.
Domestic Hot Water (153 x 833)	=	127,450 Btu/hr.
Gross Load	=	568,090 Btu/hr.

The gross capacity of an A-508SB boiler is 588,000 Btu/hr. Therefore the eight section boiler would be adequate.

SIZING—CHIMNEY AND BREECHING

CHAPTER 25

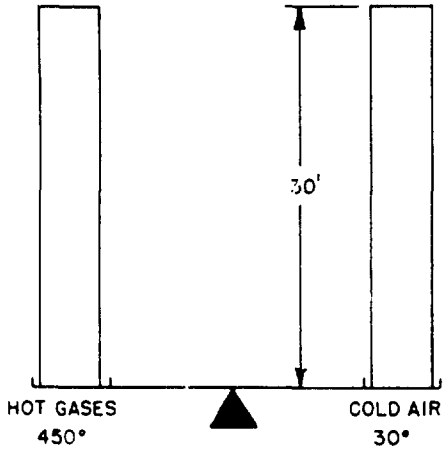
CHAPTER 25

SIZING -- CHIMNEY AND BREECHING

Proper draft in the combustion space of a boiler is of utmost importance to the satisfactory operation of the burner. Without adequate draft, sooting will occur, efficiency of combustion is lost and noisy burner operation can result.

Excessive draft conditions can be as troublesome as insufficient draft. Usually this condition is readily corrected by flue dampers, draft regulators, divertors or these devices in combination.

What is draft? Draft is really a difference in pressure exerted by a column of hot gases in a chimney and a column of cold air of the same height outside the chimney.



To illustrate, assume two columns, each 30 feet high, and each 1 square foot in cross sectional area, supported by the pans of a giant balance. See Figure 25-1.

Further assume that one column is made up of flue gases at 450°F, the other of air at 30°F.

Each column, then, will contain 30 cubic feet. Each has a different weight.

FIGURE 25-1

	Hot Gases	Cold Air
Weight per cubic foot	.044	.081
Number of cubic feet	30.	30.
Total Weight (pounds)	1.32	2.43
Pressure (lbs/sq. ft.)	1.32	2.43

The greater weight of air would tip the scale, raising the column of hot gases. The degree to which it tips depends on the pressure difference -- in this case 1.11 pounds per square foot.

1.11 pounds per square foot divided by 144 gives .0077 pounds per square inch.

Because draft is measured in inches of water -- and 1 pound per square inch equals 27.72 inches of water -- we multiply .0077 x 27.72 to get .21 inches of water draft. This is a very tiny force. Because it is so tiny the boiler room must be open to plenty of air for ready combustion.

This .21 inches of water is the theoretical draft available from a 30-foot high chimney when the stack gases weigh .044 pounds per cubic foot and the outside air weighs .081 pounds per cubic feet.

The draft would increase if the outside air cooled, or if the gases got hotter or more humid. It would decrease if the opposite occurred.

Other factors affecting draft are:

1. The tightness of the chimney. If it leaks, the cold entering air lowers the temperature of the gases and adds to the volume of gases to be vented.

2. The roughness of the chimney lining.

3. Off-sets in the chimney and other obstructions in the path of the gases.

4. Outside air turbulence, either natural on a gusty day or caused by adjacent trees or buildings, may create down drafts.

The draft calculated above is theoretical. As the gases start to flow, the friction in the chimney causes a loss so that there is less draft at the chimney thimble where the smoke pipe enters.

Friction in the smoke pipe and the boiler flues reduces the draft still further so that the draft available in the combustion chamber is usually a very small force.

Draft may be classified as:

1. Natural
2. Forced
3. Induced
4. Combinations of two or all three.

Natural draft occurs when a chimney is employed to overcome all resistance to the air flow into the combustion chamber and the friction of the flue gases flowing through the boiler, the smoke pipe and the chimney itself.

Forced draft is the case where a fan is used to force the air into the combustion chamber. With gun type oil burners and mechanical stokers forced draft is employed.

Induced draft uses a fan to suck the flue gases through the boiler and deliver them to the chimney, The use of draft booster fans is an example of this type of draft.

The use of either the forced draft or the induced draft principle reduces the natural draft requirements on a job -- which reduces the required chimney height.

Within limits, chimneys may be increased in cross sectional area to reduce the gas velocity and the flow friction through it. If the cross sectional area is made too great for the quantity of flue gases flowing, these gases will not fill the chimney. Air will flow down inside the lining mixing with the flue gases and retarding their flow. The draft will be materially reduced to unsatisfactory levels.

When chimneys are too large in area, but sufficiently high, a smaller size liner may be inserted from the top. The space between the new liner and the old must be capped off at the chimney top to prevent cold outside air from circulating down between them.

The chimney height is determined by the draft requirements and the area by the Btu per hour input to the boiler.

The height and cross sectional size of chimneys required by American-Standard boilers are specified in the Technical Catalog Bulletins.

Chimneys serving more than one appliance should have separate flues for each.

Where it is necessary to hook up two fuel-burning appliances to the same flue, they must each have their own smoke-pipe with the smaller appliance connected to the chimney at a higher level.

If they are not connected separately, the pipe not in operation may damp off the operating pipe. The smaller appliance should be piped into the chimney at a higher level where its effect will be minimized.

Gas-fired boilers must be equipped with draft hoods to let air into the vent and to divert down-drafts that might blow out pilot flames.

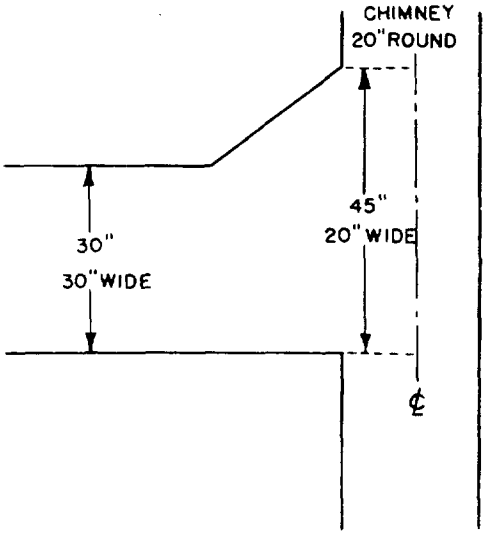
The same result is achieved for oil-fired or stoker-fired coal boilers by installing a draft stablizer in the smoke pipe.

Adjustment is made by moving a weight on the damper door.

Boilers should be installed with not more than one 90° elbow in the smoke pipe between the boiler and the chimney. If additional elbows or fittings are used, which add resistance to the flow of combustion products within the venting system, the equivalent length of such fittings should be added to the measured length of the venting system. Never use connectors smaller than the boiler flue outlets.

A breeching is essentially a manifold. Its prime purpose is to collect the gases from a large boiler with multiple smoke pipes, or the gases from a battery of boilers and deliver these products of combustion to the chimney. A constant size breeching is recommended. It must be large enough to handle the combined flue gases from all connections to it.

The breeching size is usually greater than the chimney, a situation requiring an adaptor, to make the connection of the breeching to the chimney.



For example: The problem is to tie a 30" square piece of breeching into a circular chimney with a 20" diameter.

Obviously, the breeching must be brought down to a width of 20 inches.

This means that in order to maintain the same 900 square inches of cross-sectional area, the 20-inch width will need a 45 inch height.

The solution is to add the extra 15 inches in height at the top of the breeching to introduce the flue gas into the chimney in a rising pattern.

FIGURE 25-2

When batteries of boilers are step-operated -- one on at a time and one off at a time -- it is advantageous that the boiler nearest the chimney be operating first on the "on" cycle and last on the "off" cycle.

This arrangement with properly designed breeching will eliminate the problems of "spillage" of flue gases from one boiler to another.

Breeching for more than one boiler should, at a minimum, have the total cross-sectional area of all the smoke-piping for each individual boiler.

If the requirements of space or dollars dictate a smaller breeching area, it may be reduced to a maximum of 10%, providing stack height and cross-sectional area are adequate, and a straight breeching can be designed.

Breeching sizing is based on one square inch of area for each 9,000 Btuh of heat release in the fuel.

Table 25-1

Column A Diameter inches	Column B Area Square Inches	Column C Equivalent in Rectangular size
12	113	12 x 10
13	133	12 x 12
14	154	14 x 12
15	177	14 x 14
16	201	15 x 14
17	227	15 x 16
18	254	17 x 16
19	283	17 x 18
20	314	19 x 18
21	346	21 x 18
22	380	20 x 20
23	415	22 x 20
24	452	22 x 22
25	491	24 x 22
26	531	26 x 22
27	572	26 x 24
28	615	28 x 24
29	660	30 x 24
30	706	32 x 24
31	754	34 x 24
32	804	36 x 24
33	855	39 x 24
34	907	41 x 24
35	962	36 x 30
36	1017	38 x 30
37	1075	40 x 30
38	1134	42 x 30

Example No. 1

Design the flue headers, manifolds and breeching for two G-6022 boilers in a battery.

Heat input = 3,045,000 btuh per boiler

Header & manifold area = $\frac{3,045,000}{9,000}$ = 339 square inches

Diameter of Header

$$D^2 = \sqrt{\frac{339 \times 4}{\pi}} = 432$$

$$D = \sqrt{432} \text{ or } 21"$$

Example No. 2

Breeching area for two boilers equals twice the area for each header or 2 x 339 or 678 square inches.

If the breeching is circular then:

$$D^2 = \sqrt{\frac{678 \times 4}{\pi}} \text{ or } \sqrt{854}$$

D = 29.3" or 30"

If the breeching is rectangular, it could be: 32" x 24"

To save time, use Table 25-1. Having determined that the breeching area required is 339 square inches (Example No. 1) enter Table 25-1 in Column B, going down the column to the area in square inches that is equal to or greater than 339. In this instance the number is 346. Reading to the left a round manifold of 21 inches is indicated. Reading to the right a rectangular manifold of 21 x 18 is shown as being satisfactory.

See Figure 25-2 for breeching arrangement.

To summarize, helpful hints are listed below.

HELPFUL HINTS

1. Always use individual vents in preference to combined vents, when possible and practical.
2. Never use connectors smaller than the boiler flue products outlet(s).
3. Always make the rise above the boiler flue products outlet(s) as great as possible. This is especially important when connecting more than one boiler to a single chimney.

4. Keep the length of connectors as short as possible.

5. Use an inside chimney in preference to an outside chimney, when possible and practical.

6. Use insulated vents, when long connections to chimney cannot be avoided. Insulation on the rise from the boiler flue products outlet(s) is particularly important.

7. Eliminate all possible flue turns in the connection to the chimney. Remember that when more than one elbow is used between the boiler flue products outlet(s) and the chimney, allowance for the added resistance must be made by adding equivalent length of such fittings to the vent system measured length before applying draft loss curves.

8. Never let the connector protrude beyond the inside wall of the chimney nor permit projections into the flue gas passage.

9. Avoid caps on chimney that will restrict the free area of the chimney.

10. "Streamline" the vent system as much as possible -- changes in direction, shape and flow area should be accomplished separately and as gradually as possible.

11. Round connectors are to be preferred over square or rectangular. Square or rectangular flue passages are apt to promote unpredictable turbulence that will interfere with the normal flow of combustion products. Also, flat sides of square or rectangular connectors are more likely to develop noise due to resonance.

12. Lateral connectors should be run horizontal, or with minimum slope permitted by local code, to keep the rise above the boiler flue products outlet(s) as great as possible.

13. Breechings that connect two or more boilers to a common chimney must be sized to handle the combined load. A single size manifold of sufficient size for the load should be used.

14. Never connect 2 or more chimneys together to increase flue area available for a boiler installation. The ASHRAE Guide says "When two chimneys are connected at the bottom, hot gases in the U-tube thus formed would be in unstable equilibrium. Cold air from the top would descend through one such chimney and drive the hot gases out of the other, thus annulling the draft." If 2 or more chimneys are used, each chimney should be used separately for part of the boilers. Never use a continuous manifold between the chimneys.

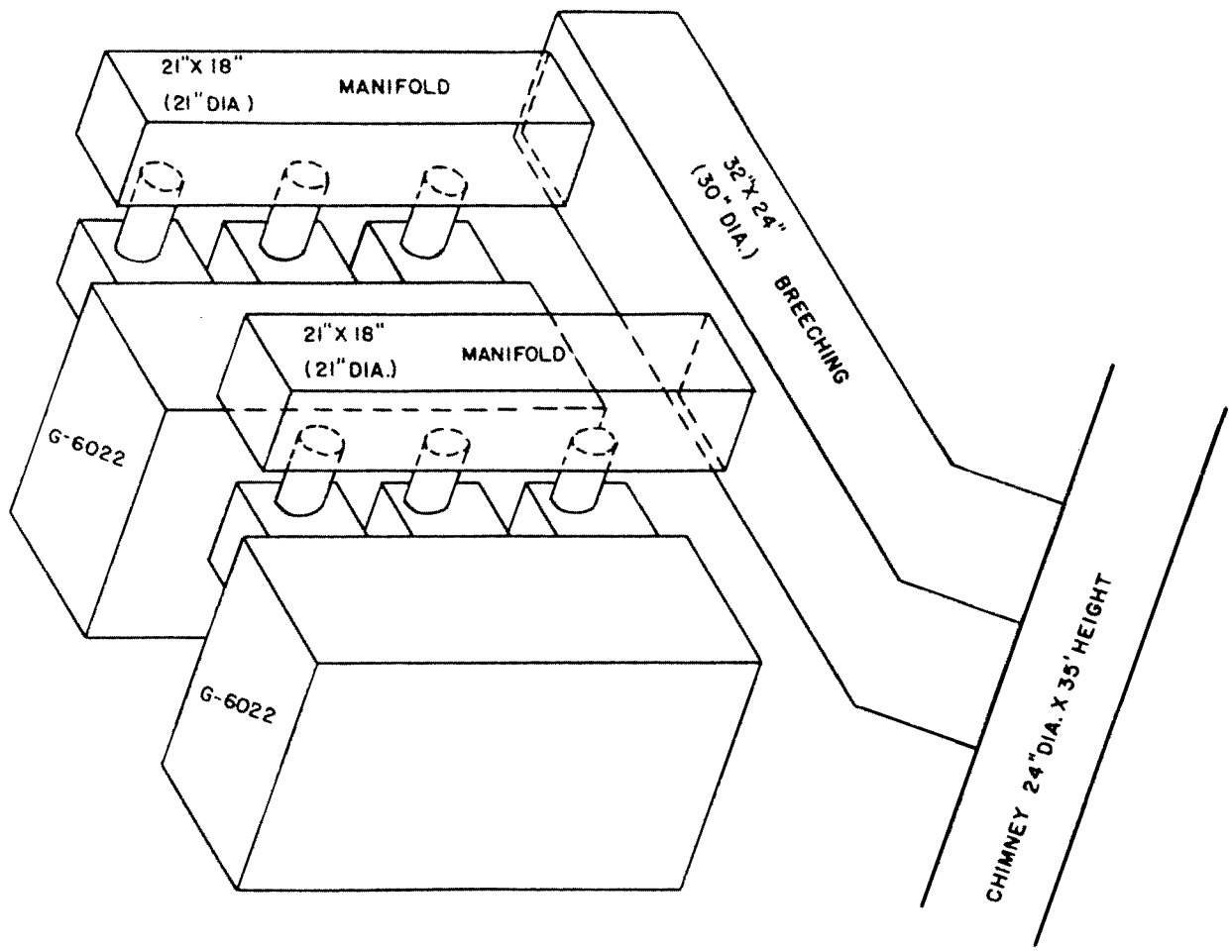

21"X 18"
(21" DIA)
MANIFOLD
21"X 18"
(21" DIA.)
MANIFOLD
G-6022
G-6022
32"X 24"
(30" DIA.)
BREECHING
CHIMNEY 24" DIA. X 35' HEIGHT

FIGURE 25-2

LOCATING THE BOILER

CHAPTER 26

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CHAPTER 26

LOCATING THE BOILER

There are three possible places to locate a boiler:

1. Below the system...in a basement.
2. Above the system...in a penthouse, on the roof of the building.
3. Alongside the system...in a separate room close to, but detached from, the space to be heated...industrial.

Below the System

Locating the boiler below the system creates...for any building...the best possible draft condition. Chimney height is at its maximum.

For steam systems, the basement is the ideal location because gravity helps to bring condensate back to the boiler, even though, for reasons of fuel economy, a vacuum system using condensate pumps is often specified.

Even in tall buildings, low pressure steam boilers operate successfully out of a basement location because steam does not build up high hydrostatic pressures.

For any single hot water system, hydrostatic pressures in tall buildings present insurmountable problems. The solution (see Chapter 25) is to use a steam boiler and a series of convertors.

Our present line of large boilers, improved from 30 psig operation to 50 psig, can be used in buildings 80% taller than for former maximum capability.

Because of hydrostatic pressures, expansion tanks must be larger as building height increases.

Above the System

Roof or penthouse systems are rarely, if ever, acceptable for either coal or oil-burning boilers because of chimney height.

The use of steam boilers on building roofs is possible but impractical because head requirements on condensate pumps become too great.

Because of the help from gravity, low pressure hot water systems can be used in roof installations. In addition, expansion tank sizes are economical.

Alongside the System

Because of the low chimney height, gas is the ideal fuel for industrial heating systems, though oil and stoker-fired boilers with somewhat higher

chimney requirements are often used. Hand-fired coal boilers usually present architecturally unacceptable chimney height specifications.

For steam systems, the smaller installations can usually be set up to return condensate by gravity. Usually, however, condensate pumps are used.

For hot water systems, industrial applications eliminate the major problems of high hydrostatic pressures and large expansion tanks.

Locating the Second Boiler in a Steam System

When more than one boiler is used in a steam system, it is imperative that all boiler water lines be at the same level.

While the water line problem makes it desirable to use boilers of the same design in new installations, it is not always possible.

When, for instance, an estimated heating load indicates the use of a G-60 boiler with a 44" water line plus a G-40 with a 35-1/2" water line... it would be necessary to install the G-40 on an 8-1/2" concrete pad.

At times it may be necessary to pit a steam boiler to effect the equalizing of water lines.

BLAST COILS AND UNIT HEATERS

CHAPTER 27

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CHAPTER 27

BLAST COILS AND UNIT HEATERS

Two types of concentrated heat loads on a boiler are blast coils and unit heaters.

In both cases a coil, either steam or hot water, is used to heat forced hot air.

The difference is that blast coils are used in duct systems while unit heaters hang free in the space being heated. Propeller type fans are mounted on the unit heaters, either manually or thermostatically controlled, and the air leaving the heater is directed by louver as it leaves the heater.

The forced circulation of air condenses steam so rapidly, particularly at start-up of the daily heating cycle, as to create a vacuum in the heater which can suck water from the wet return up into the heater itself.

To overcome this vacuum and to insure adequate removal of condensate, steam systems using unit heaters normally carry a condensate pump for each heater, or for a battery of heaters.

With the heaters installed at a high level, a direct, vertical, drop-leg to the condensate pump allows for fast and positive drainage of the condensate.

Unit heaters, for best results, should be operated at a minimum of 2 psig steam pressure at the heater inlet. Where entering air temperature, however, is below freezing, pressure should be at least 5 psig.

Pages 27-2 and 27-3 illustrate recommended piping arrangements for unit heaters.

Blast coils serve two functions:

1. To temper ventilation air.

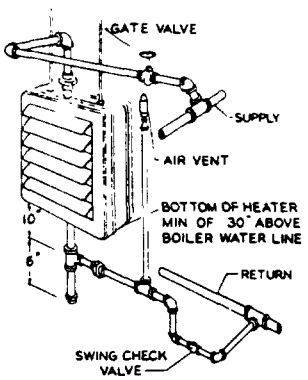
2. To supply heated air.

In blast coil systems, air is circulated through the ducts by centrifugal blowers.

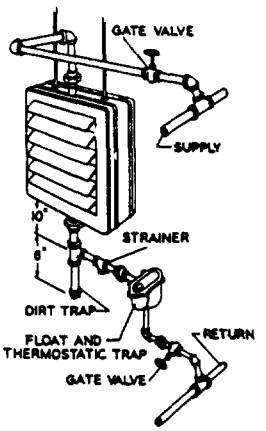
The following pages show piping diagrams for individual blast coils in various systems plus a list of "do's and don'ts".

PIPING DIAGRAMS

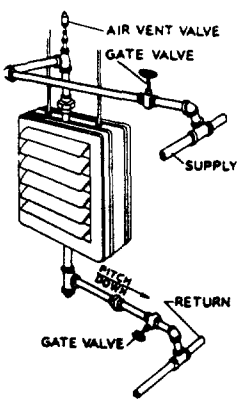
HORIZONTAL UNIT HEATERS



LOW PRESSURE GRAVITY SYSTEM



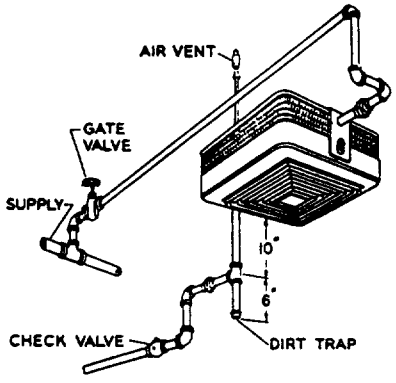
VAPOR OR VACUUM SYSTEM



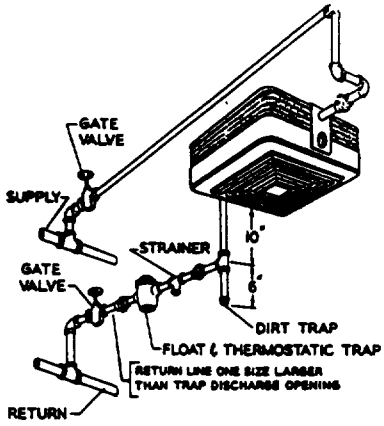
HOT WATER SYSTEM

PIPING DIAGRAMS

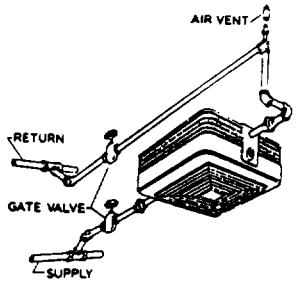
VERTICAL UNIT HEATERS



LOW PRESSURE GRAVITY SYSTEM



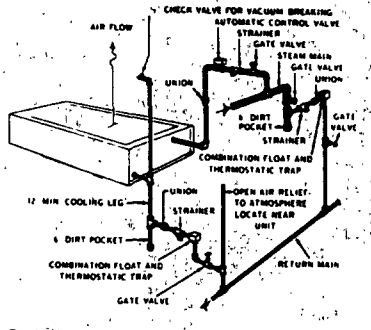
VAPOR OR VACUUM SYSTEM



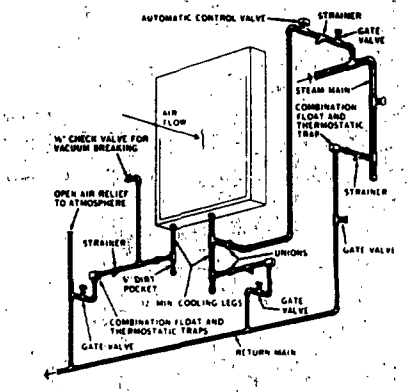
HOT WATER SYSTEM

PIPING DIAGRAMS

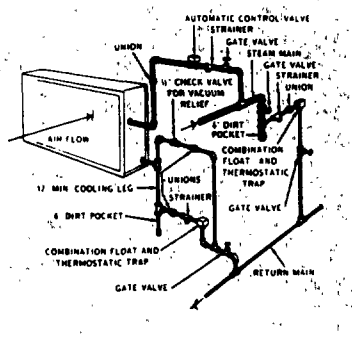
BLAST COILS



Typical piping diagram—open gravity system, low pressure steam—coil tubes horizontal



Typical piping diagram—open gravity system, low pressure steam—coil tubes vertical



Typical piping diagram—vacuum system, low pressure steam—coil tubes horizontal

Coil Installation
Coils must be accurately

Install casings of all heavy duty coils level for either horizontal or vertical air flow. Install coils accurately with a spirit level, not by measurement from building members.

Coil Piping
Multiple Coil Banks

The diagram on preceding pages show the piping arrangements for single coils. The same arrangements should be used on multiple coil banks, the most satisfactory arrangement being to build up the proper system by duplicating the diagrams on each coil in the bank. However, coils in parallel (side by side) or in series (one behind the other) in the air flow may, if desired, be controlled by a common valve. Each coil in a coil bank, whether individually controlled or controlled by a common valve, must have its own trap and air venting system.

Provide All Coils with Air Vents

On low pressure systems (with steam pressure 15 psig or less) where non-venting traps are used for condensate removal to a vacuum return, a 1/2" thermostatic air trap should be located in an air line bypassing the condensate trap. The air line to the trap should be of 1" pipe and the trap located at least 12" above the coil return connection. On low pressure systems where non-venting traps are used for condensate removal to a gravity return, 3/4" individual return line automatic air vents are recommended. These vents should be located at least 12" above the coil return connection and vent to the room. The use of thermostatic traps for air removal is permissible (but not recommended) with a gravity return, provided the return line is vented adequately to the outside atmosphere.

Provide Vacuum Relief

A vacuum breaker must always be provided when automatic steam control is used. Under many conditions of operation with modulating control, or when steam is shut off by a two-position on-off control, a vacuum will occur in the coil unless proper relief is provided. The vacuum holds condensate in the coil tubes causing water hammer and, when sub-freezing air is being handled, allows the tubes to freeze. It is recommended that vacuum breakers also be used with manual steam control to avoid the possibility of condensate hold-up when the valve is shut off.

On gravity systems, install a 1/2" check valve in a riser at the return connection from each coil. On vacuum systems, install a check valve in a bypass line around the trap to prevent the coil vacuum from exceeding that in the return.

Allow for Removal of Heating Element

Make steam and return connections to the coils through an elbow to a union as shown in the piping diagrams. This leaves the coil connection end free of piping allowing removal of the heating element when the unions are diconnected. This procedure is not necessary on Type HA Coils since the heating element is removed from the end opposite the coil connections.

Avoid Lifting Condensate and Pressure Returns

Lifting of condensate above the coil return into an overhead main should always be avoided if possible and must never be done with automatic control. Temperature control by modulating or by on-and-off automatic steam valves must not be used with systems in which a pressure is maintained in the return lines.

Protect Automatic Control Valves, Coils and Traps by Strainers

Automatic valves operate much of the time at small openings and are easily plugged by scale and dirt. Each valve should be preceded by a line strainer to assure continuous operation. Each coil should be preceded by a line strainer to protect the supply tube orifices against plugging. If traps are not equipped with strainers, they should be preceded by line strainers to protect their small orifices against plugging.

Avoid Rigid Piping Connections

All steam and return piping must be provided with swing joints at or near the heating coils so that piping and expansion strains are not communicated to the coils. All piping should be supported and anchored independently of the heaters.

Drip All Steam Lines

Steam line condensate should never be passed through heating coils. Steam mains should be dripped from the pressure side of the valve to the return line through a drip trap. Supply branches should be pitched back to the dripped steam main. Failure to observe this precaution will result in damage to coils through hammer and corrosion.

Do Not Bush Return Connections

Return piping must be made the full size of the coil return connection from the coil to the manifold or trap. Coil outlets must not be bushed. Piping from the coil return connection should have a vertical cooling leg at least 12" long and a dirt pocket at least 6" long -- all full size of the return connection.

Traps Should be of Ample Size

Blast heating coils should have traps of ample size to handle the high condensation rate of these coils during the starting up period. In general, traps should be selected for 2-1/2 times the calculated coil load.

Isolate Coils, Valves and Traps

Aside from any question of control, each coil in a multiple coil bank should be provided with a gate valve in its steam supply line and in the condensate line beyond the trap to permit servicing of coils and traps without shutting down the entire system. For the same reason, each automatic control valve should be preceded by a gate valve.

Turn Steam On Before Starting Fans

Steam should be turned on to heat up coils before air flow is started, particularly when sub-freezing air is being handled. A 10-minute warm-up period before starting fans is desirable. This will help to reduce the flood of condensate which occurs when steam is suddenly admitted to a cold coil. The possibility of freezing and water hammer due to the coil filling with condensate will thus be reduced.

DOMESTIC HOT WATER

CHAPTER 28

CHAPTER 28

DOMESTIC HOT WATER

The estimating of domestic hot water requirements for apartments, motels, hotels, clubs, office buildings, schools and the like is open to considerable question.

A number of different methods are employed. For example, the manufacturers of commercial hot water heaters usually use a figure of 15 gallons per hour per person. In a motel of 30 twin bed room units, allowing 1-1/2 people per unit, the total hot water requirement at peak load would be estimated at 30 x 1-1/2 x 15 or 675 gallons per hour. This approach for large apartments, hotels, housing projects and hospitals results in astronomical amounts of hot water, which are not realistic.

There are reliable data available, on hot water demand of fixtures, which may be used in estimating connected hot water loads. It is suggested that these demands tabulated in Table 28-1 be used, particularly in cases of large requirements.

To illustrate its use assume a 60 suite garden apartment each with a single bath, with shower and kitchen with dishwashing. Using Table 28-1, the estimated connected load is

60 lavatories	x	2	=	120 gallons per hour
60 showers	x	30	=	1,800 gallons per hour
60 kitchen sinks	x	10	=	600 gallons per hour
60 dish washers	x	15	=	900 gallons per hour
Total connected demand			=	3,420 gallons per hour

Not all fixtures will be in use simultaneously. For apartments the use factor is estimated at .3 making the actual hourly demand .3 x 3,420 = 1,026 gallons per hour.

Warning

Obviously, 1,026 gallons per hour averages out to 17.1 gallons per minute. But, do not assume that a tankless heater good for 17.1 gallons per minute would be satisfactory. During the hour instantaneous demands for hot water might be at a rate of several hundred gallons per minute. Such an amount would be inpractical to supply with a tankless heater unless there was unlimited boiler capacity available.

The preferred method of handling large hot water demands is with a separate boiler (the IS method) or with tank-type heaters installed in the space heating boiler. In each case a storage tank is used.

It is also possible to use the tankless heaters supplied with the A-5 boiler and the two largest sizes of those furnished with the G-40 as tank type heaters. When this is done a pump must be used to circulate the water through the heater and the tank since the small size of tubing used in these heaters has too much friction to permit adquate wqter circulation by gravity. In

Table 28-1

Hot Water Demand per Fixtures for Various Types of Buildings

Gallons of water per hour per fixture, calculated at a final temperature of 140 F

	Apartment House	Club	Gymnasium	Hospital	Hotel	Industrial Plant	Office Building	Private Residence	School	Y.M.C.A.
1. Basins, private lavatory	2	2	2	2	2	2	2	2	2	2
2. Basins, public lavatory	4	6	8	6	8	12	6	....	15	8
3. Bathtubs	20	20	30	20	20	....	....	20	....	30
4. Dishwashers[a]	15	50-150	....	50-150	50-200	20-100	....	15	20-100	20-100
5. Foot basins	3	3	12	3	3	12	....	3	3	12
6. Kitchen sink	10	20	....	20	30	20	20	10	20	20
7. Laundry, stationary tubs	20	28	....	28	28	....	....	20	....	28
8. Pantry sink	5	10	....	10	10	....	10	5	10	10
9. Showers	30	150	225	75	75	225	30	30	225	225
10. Slop sink	20	20	....	20	30	20	20	15	20	20
11. Hydro-therapeutic showers				400						
12. Hubbard baths				600						
13. Leg baths				100						
14. Arm baths				35						
15. Sitz baths				30						
16. Continuous-flow baths				165						
17. Circular wash sinks				20	20	30	20		30	
18. Semi-circular wash sinks				10	10	15	10		15	
19. Demand factor	0.30	0.30	0.40	0.25	0.25	0.40	0.30	0.30	0.40	0.40
20. Storage capacity factor[b]	1.25	0.90	1.00	0.60	0.80	1.00	2.00	0.70	1.00	1.00

[a] Dishwasher requirements should be taken from Table 14 or from manufacturers' data for the model to be used, if this is known.
[b] Ratio of storage tank capacity to probable maximum demand per hour. Storage capacity may be reduced where an unlimited supply of steam is available from a central street steam system or large boiler plant.

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making such application the Hydronic Product Group should be consulted.

The tankless heater capacities, when used as tank-type heaters, are as follows:

Heater No.	Btu/hr.
530	160,000
232	249,000
445	299,000

There are tank-type heaters on the market, produced by heater manufacturers, that will fit American-Standard boilers.

IS Domestic Hot Water

The IS in the title stands for "indirect steam". In this kind of system a steam boiler is used to heat domestic hot water in much the same way steam is used with a convertor to supply forced hot water to a heating system.

The IS system has a number of advantages over the DW or direct water heating.

1. The hot water has no discoloration. In DW systems the water becomes slightly tinted from its contact with cast iron of the boiler.

2. Scale forming substances which would normally be deposited on the inside boiler surface with DW systems are now deposited on the outside of the steam coil in the external heater. Since these heaters are designed for coil accessibility or removal, they can be readily cleaned when liming becomes excessive.

3. The boiler is working on a closed system with the condensate being returned to it, thereby reducing internal liming to an inconsequential minimum.

G-2 boiler prices are quoted only for IS use. However, any steam boiler in the American-Standard line can be used in an IS system. Choice depends on the load required and the fuel desired.

Boiler requirements for an IS system are:

1. Standard steam equipment including steam gauge.

2. ASME safety valve.

3. Pressure limit control.

4. Gauge glass with trimmings.

5. Resetting low water cut-off.

In addition, the system should be supplied with a CA-666 immersion operating control, installed in the water storage tank at or near the cold water inlet.

Heat is transferred to the water through either a coil in the storage tank or an external indirect heater. In both cases, the equipment is supplied by other manufacturers who should specify the sizing of the heat exchange equipment.

The principle of sizing the boiler in these installations is simple because the domestic hot water demand is the only load...and, in addition, the storage tank is located close by the boiler.

The boiler's ability to heat domestic hot water is based on its I=B=R gross capacity or its A. G. A. output.

Computing the gallons of hot water any boiler can supply is a simple problem of multiplying the gross capacity of the boiler in MBH by a factor dependent on the temperature rise desired.

Factors are	for ..... Various Water Temperature Rises
1.2	100°F
1.3	90°F
1.5	80°F
1.7	70°F
2.0	60°F

The equation:

Demand (GPH) = MBH x Factor

Example:

How many gallons of water per hour, at a 100°F temperature rise can be supplied by an American-Standard G-25 boiler, which has a gross capacity of 96,000 Btu/hr.

GPH = 96 x 1.2.... or 115.2 gallons per hour

Similarly ...

GPH (at 90°F rise) = 96 x 1.3 ... or 124.8 GPH
GPH (at 80°F rise) = 96 x 1.5 ... or 144.0 GPH
GPH (at 70°F rise) = 96 x 1.7 ... or 163.2 GPH
GPH (at 60°F rise) = 96 x 2.0 ... or 192.0 GPH

For economy reasons, particularly with severe peak demands, it is usual to employ a storage tank in an IS system to permit the heating and storing of water during off-peak periods.

The size of the storage tank and boiler will, of course, depend on the length of peak demand.

The equation, with "N" being number of hours of peak demand is:

N x Demand (GPH) = N x Boiler Capacity (GPH) + .7 x Tank Volume (GPH)

OR:

$$\text{Tank Volume (Gals)} = \frac{N\ (\text{Demand} - \text{Boiler Capacity})}{.7}$$

The following tables show various combination of boiler capacity and tank volume to meet various demands of both gallonage and temperature rise.

Table 28-2

MINIMUM STORAGE TANK CAPACITIES 1 HOUR PEAK DEMAND

100°F Temperature Rise

Hot Water Demand g.p.h.	G-2 Boiler-Sizes									G-40 Boiler-Sizes								G-60 Boiler-Sizes			
	23	24	25	26	27	28	29	210	211	403	404	405	406	407	408	409	4010	606	607	608	609
100	60																				
150	131	92	50																		
200		163	121	80						80											
250			207	151	110	70				166	49										
300				223	181	141	100			223	120										
350					253	213	171	130		295	192	89									
400						284	243	201	160	366	263	160	58								
450							314	273	231		335	232	129								
500								415	303		406	303	200	98							
550									374			375	272	169	66						
600									446				343	240	137						
650														312	209	106					
700	Tank vol.(gals.) = N(demand[gph]-boiler cap.[gph]) ÷ 0.7													383	280	177	75				
750	N = Number of hours of peak demand													455	352	249	145	77			
800															423	320	217	149			
850																392	289	220			
900																463	360	291	93		
950																	432	362	164		
1000																		434	236		
1050																		506	307	109	
1100																			379	180	
1150																			450	251	
1200																			521	323	123
1250																				394	194
1300																				466	265

Assume gas as the fuel and a demand for 200 GPH. Opposite the 200 GPH demand, under G-2 boiler sizes there are 3 possible combinations to meet these requirements.

1. G-24 boiler and 163 gallon tank (min)
2. G-25 boiler and 121 gallon tank (min)
3. G-26 boiler and 80 gallon tank (min)

Naturally, the proposal will suggest the most economical combination.

If the peak demand time were doubled to two hours, tank size would have to be doubled. If peak demand went to three hours tank size would have to be tripled. Tank sizes cannot be increased indefinitely. At some point, both economy and necessity will dictate the next larger size boiler.

Table 28-3

MINIMUM STORAGE TANK CAPACITIES 1 HOUR PEAK DEMAND

80°F Temperature Rise

Hot Water Demand g.p.h.	G-2 Boiler-Sizes									G-40 Boiler-Sizes								G-60 Boiler-Sizes				
	23	24	25	26	27	28	29	210	211	403	404	405	406	407	408	409	4010	606	607	608	609	610
100	41																					
150	103	62																				
200	184	133	82	30						30												
250	256	204	153	101	50					100												
300		276	224	173	121	70				172	43											
350			296	244	193	141	90			243	114											
400				316	264	213	161	110	59	314	186	57										
450					336	284	233	182	130	386	257	129										
500						356	304	253	201		329	200	71									
550							375	324	273		400	272	143									
600								396	344			343	214	86								
650									415			415	286	157								
700	Tank vol.(gals.)= N(demand[gph]-boiler cap.[gph]) ÷ 0.7												357	229	100							
750	N = Number of hours of peak demand												429	300	172							
800														372	243	114						
850														444	314	186	57					
900															386	257	129					
950															457	329	200	114				
1000																400	272	186				
1050																471	343	257				
1100																	415	329	80			
1150																	486	400	152			
1200																		471	223			
1250																			294			
1300																			366	117		

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Here a demand for 700 gallons an hour with an 80°F temperature rise dictates either:

1. G-406 boiler with 357 gallon tank
2. G-407 boiler with 229 gallon tank
3. G-408 boiler with 100 gallon tank

Again, one will cost less...and, again tank sizes will have to be doubled, if peak demand stretches out to two hours.

The following table covers the same three boilers with a 60°F temperature rise, and still smaller tank requirements.

Table 28-4

MINIMUM STORAGE TANK CAPACITIES 1 HOUR PEAK DEMAND

60°F Temperature Rise

Hot Water Demand g.p.h.	G-2 Boiler-Sizes									G-40 Boiler-Sizes								G-60 Boiler-Sizes
	23	24	25	26	27	28	29	210	211	403	404	405	406	407	408	409	410	606
100																		
150	77																	
200	149	80																
250	220	152	83															
300		223	155	86						86								
350		295	226	157	87					157								
400		366	297	229	160	91				229	57							
450			369	300	232	163	94			300	129							
500			440	372	303	234	166	98		372	200							
550			512	443	375	306	237	169	100	443	272	100						
600					446	377	309	240	172	515	343	171						
650						449	380	312	243		415	243	72					
700							452	383	315		586	314	143					
750								455	386			386	214					
800									457			457	286	115				
850												529	357	186				
900													429	257	86			
950													500	329	157			
1000														400	229	57		
1050														472	300	129		
1100														544	372	200		
1150															443	272	100	
1200															515	343	171	57
1250																415	243	129
1300																586	314	200

Tank vol.(gals.)=N(demand[gph]-boiler cap.[gph])÷ 0.7

N = Number of hours of peak demand

The following three tables provide the same kind of information on Arcoliner, A-5 and A-7 boilers.

Table 28-5

MINIMUM STORAGE TANK CAPACITIES 1 HOUR PEAK DEMAND

100°F Temperature Rise

Hot Water Demand g.p.h.	Arcoliner-Sizes				A-5 Boiler-Sizes								A-7 Boiler-Sizes	
	153	154	155	156	504	505	506	507	508	509	510	511	705	706
100														
150	49													
200	120	60												
250	191	131	76											
300	263	203	147	106										
350	334	274	219	177	53									
400	406	346	290	247	124									
450		417	362	320	196	50								
500			433	392	268	121								
550				463	339	193	56							
600					410	264	127							
650					482	336	199	57						
700						407	270	129						
750						479	342	200	63					
800							413	272	134					
850							485	343	206	69				
900								415	278	140			Tank vol.(gals.)=N(demand[gph]-boiler cap.[gph]) ÷ 0.7	
950								486	349	211	74		N = Number of hours of peak demand	
1000									420	283	146			
1050									492	354	217	77		
1100										426	289	149	87	
1150										497	360	220	158	
1200											432	292	230	
1250											503	363	302	
1300												435	373	123

Table 28-6

MINIMUM STORAGE TANK CAPACITIES 1 HOUR PEAK DEMAND

60°F Temperature Rise

Hot Water Demand g.p.h.	Arcoliner-Sizes				A-5 Boiler -Sizes								
	153	154	155	156	504	505	506	507	508	509	510	511	
100													
150													
200													
250	80												
300	166	51											
350	223	123	31										
400	295	194	103										
450	366	266	174	106									
500	437	337	246	177									
550		409	318	249									
600		480	389	320	111								
650			460	392	183								
700				463	255	83							
750					326	155							
800					397	226							
850					469	297	69						
900						369	140						Tank vol.(gals.)=N(demand[gph]-boiler cap.[gph]) ÷0.7
950						440	211						N = Number of hours of peak demand
1000						512	283						
1050							354	49					
1100							426	120					
1150							497	191					
1200								263					
1250								334	106				
1300								406	177				

Table 28-7

MINIMUM STORAGE TANK CAPACITIES 1 HOUR PEAK DEMAND

80°F Temperature Rise

Hot Water Demand g.p.h.	Arcoliner Sizes				A-5 Boiler Sizes								
	153	154	155	156	504	505	506	507	508	509	510	511	
100													
150													
200	79												
250	150	74											
300	222	146	77										
350	294	217	149	97									
400	365	289	220	169									
450	436	360	292	240	83								
500	507	432	363	312	155								
550		503	435	383	226	44							
600			506	455	297	116							
650				526	369	187							
700					440	259	87						
750					512	330	160						
800						401	232	54					
850						473	303	126					
900						545	375	197					
950							446	269	97				
1000							516	340	169				
1050								412	240	69			
1100								483	312	140			
1150								555	383	211			
1200									455	283	111		
1250									526	354	183		
1300										426	254	79	

Tank vol.(gals.)= N(demand[gph]-boiler cap.[gph])÷0.7

N = Number of hours of peak demand

CHAPTER 29

GREENHOUSE HEATING

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CHAPTER 29

GREENHOUSE HEATING

Greenhouse heating is, of course, different from space heating for human comfort.

Temperature is frequently maintained at less than 70°F, which increases the emission rate of radiation.

Studies of greenhouse heat loss indicate that the glass area is a good base for calculating the load. The necessary allowances for wall loss and small amounts of infiltration have been made in the tables that make up the body of this chapter.

These Tables, 29-1 through 29-7, have been prepared for 1,000 square feet of glass...they cover an outdoor temperature range of 20°F down to -30°F and an indoor range from 40° to 70°F. They are based on data compiled by the United States Department of Agriculture and published by the "Heating and Ventilating Magazine".

Knowing the number of thousands of square feet of glass in a greenhouse, the following information can be obtained:

1. The total Btu per hour heat loss for sizing the boiler or boilers.

2. The square feet of radiation required for
   a. Steam at 220°F.
   b. Gravity circulated hot water at 160°F.
   c. Forced circulated hot water at 210°F.

3. Linear feet of pipe radiation required for
   a. Steam at 220°F
   b. Gravity circulated hot water at 160°F
   c. Forced circulated hot water at 210°F

As the tables themselves indicate, only two additional pieces of information are necessary:

1. The indoor design temperature

2. The outdoor design temperature

As a general rule the greenhouse owner will know what maximum inside temperature he requires for his crops. If there is any question, Table 29-9 provides a list of vegetable and flower crops and their customary temperature requirements.

The outside temperature is the outdoor design temperature for heating that is in common use in the area.

Let's see how they work.

The problem: A greenhouse is being built for general purpose use. Its glass area is 9,500 square feet...the outdoor design temperature is -10°F, the indoor design temperature 55°F.

The builder plans a forced hot water system and you have sold him on using cast iron radiation.

Inside design temperature of 55°F is covered in Table 29-4.

Then under the -10°F outside temperature heading, you will find on the last line "Forced hot water 210°F" that 326 square feet of EDR is needed for each 1,000 square feet of glass area.

9,500 square feet of glass, then, will require 9.5 x 326 or 3,097 square feet of EDR.

On the top line, you will note the heat loss of 85,300 Btu/hr. per 1,000 square feet of glass.

This greenhouse will need a boiler with a capacity of 9.5 x 85,300 or 810,350 Btu/hr. of net installed radiation.

To size supply and return mains for steam systems, use the standard pipe sizing tables given in Chapter 15. The figures should be used in conjunction with the Btu/hr. load equated to EDR by dividing by 240.

To size supply and return mains for forced hot water heating, use the standard milinch method discussed in Chapter 20.

For supply and return main sizing on hot water gravity systems, use Table 29-8 on Page 29-10.

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Inside Temperature
40°F

Table 29-1

REQUIREMENTS FOR EACH 1,000 SQUARE FEET OF GLASS

	Outside Temperature					
	20	10	0	-10	-20	-30
Heat Loss Btu/hr.	28,600	41,000	53,800	66,800	79,200	92,200
Linear Feet of 1-1/4" Pipe Steam	132	190	250	305	366	425
Linear Feet of 2" Pipe Gravity Hot Water	189	271	358	443	525	610
Linear Feet of 3-1/2" Pipe Gravity Hot Water	116	167	220	272	323	374
Linear Feet of 2" Pipe Forced Hot Water	120	173	226	280	333	388
Cast Iron Radiation Square Feet						
Steam	88	126	165	204	242	283
Hot Water Gravity	154	220	289	360	426	495
Hot Water Forced	97	139	162	225	268	312

Inside Temperature
45°F

Table 29-2

REQUIREMENTS FOR EACH 1,000 SQUARE FEET OF GLASS

	Outside Temperature					
	20	10	0	-10	-20	-30
Heat Loss Btu/hr.	34,800	47,500	60,100	72,600	85,300	98,000
Linear Feet of 1¼" Pipe Steam - 220°F	168	230	292	352	414	475
Linear Feet of 2" Pipe Gravity Hot Water - 160°	242	330	419	505	591	680
Linear Feet of 3½" Pipe Gravity Hot Water - 160°F	150	205	258	313	366	422
Linear Feet of 2" Pipe Forced Hot Water - 210°F	153	209	264	320	374	429
Cast Iron Radiation Square Feet						
Steam - 220°F	111	152	192	232	272	313
Gravity Hot Water - 160°F	198	270	342	413	485	555
Forced Hot Water - 210°F	122	166	211	254	299	343

Inside Temperature
50°F

Table 29-3

REQUIREMENTS FOR EACH 1,000 SQUARE FEET OF GLASS

	Outside Temperature					
	20	10	0	-10	-20	-30
Heat Loss Btu/hr.	41,200	53,800	66,450	79,000	91,700	104,500
Linear Feet of 1¼" Pipe Steam - 220°F	207	270	333	397	460	524
Linear Feet of 2" Pipe Gravity Hot Water - 160°F	304	397	490	582	676	770
Linear Feet of 3½" Pipe Gravity Hot Water - 160°F	188	245	302	360	419	475
Linear Feet of 2" Pipe Forced Hot Water - 210°F	189	247	305	362	420	478
Cast Iron Radiation Square Feet						
Steam - 220°F	140	182	224	267	310	353
Gravity Hot Water - 160°F	248	324	398	474	550	626
Forced Hot Water - 210°F	150	196	242	286	234	380

Inside Temperature
55°F

Table 29-4

REQUIREMENTS FOR EACH 1,000 SQUARE FEET OF GLASS

	Outside Temperature					
	20	10	0	-10	-20	-30
Heat Loss Btu/hr.	47,500	60,500	72,800	85,300	98,500	111,000
Linear Feet of 1¼" Pipe Steam - 220°F	245	315	377	440	508	571
Linear Feet of 2" Pipe Gravity Hot Water - 160°F	368	472	568	663	763	862
Linear Feet of 3½" Pipe Gravity Hot Water - 160°F	231	296	356	415	481	541
Linear Feet of 2" Pipe Forced Hot Water - 210°F	228	292	350	409	472	532
Cast Iron Radiation Square Feet						
Steam - 220°F	166	212	255	299	345	398
Gravity Hot Water - 160°F	300	382	460	540	623	702
Forced Hot Water - 210°F	182	221	278	326	376	424

Inside Temperature
60°F

Table 29-5

REQUIREMENTS FOR EACH 1,000 SQUARE FEET OF GLASS

	Outside Temperature					
	20	10	0	-10	-20	-30
Heat Loss Btu/hr.	53,700	66,600	79,100	91,700	104,500	117,300
Linear Feet of 1¼" Pipe Steam - 220°F	289	358	426	493	562	629
Linear Feet of 2" Pipe Gravity Hot Water - 160°F	446	549	654	768	862	971
Linear Feet of 3½" Pipe Gravity Hot Water - 160°F	278	342	408	472	541	606
Linear Feet of 2" Pipe Forced Hot Water - 210°F	270	336	397	461	524	588
Cast Iron Radiation Square Feet						
Steam - 220°F	196	243	289	334	381	427
Gravity Hot Water - 160°F	360	447	531	615	700	788
Forced Hot Water - 210°F	215	267	315	367	418	469

Inside Temperature
65°F

Table 29-6

REQUIREMENTS FOR EACH 1,000 SQUARE FEET OF GLASS

	Outside Temperature					
	20	10	0	-10	-20	-30
Heat Loss Btu/hr.	59,700	72,600	86,400	98,300	111,000	123,000
Linear Feet of 1¼" Pipe Steam - 220°F	336	403	479	550	621	690
Linear Feet of 2" Pipe Gravity Hot Water - 160°F	529	641	752	870	980	1,087
Linear Feet of 3½" Pipe Gravity Hot Water - 160°F	333	403	476	549	617	685
Linear Feet of 2" Pipe Forced Hot Water - 210°F	313	379	446	513	578	645
Cast Iron Radiation Square Feet						
Steam - 220°F	228	277	326	375	424	470
Gravity Hot Water - 160°F	427	519	610	702	792	877
Forced Hot Water - 210°F	250	304	357	411	464	515

Inside Temperature
70°F

Table 29-7

REQUIREMENTS FOR EACH 1,000 SQUARE FEET OF GLASS

	Outside Temperature					
	20	10	0	-10	-20	-30
Heat Loss Btu/hr.	66,250	79,200	91,700	104,800	117,500	129,500
Linear Feet of $1\frac{1}{4}$" Pipe Steam - 220°F	394	470	546	621	700	770
Linear Feet of 2" Pipe Gravity Hot Water - 160°F	613	735	855	971	1,099	1,205
Linear Feet of $3\frac{1}{2}$" Pipe Gravity Hot Water - 160°F	391	467	543	617	694	763
Linear Feet of 2" Pipe Forced Hot Water - 210°F	361	431	503	571	645	709
Cast Iron Radiation Square Feet						
Steam - 220°F	265	317	367	419	470	518
Gravity Hot Water - 160°F	505	605	700	798	896	988
Forced Hot Water - 210°F	289	346	400	457	513	565

Table 29-8

Size of Mains for Gravity Hot Water Pipe Coils
(Based on 160°F water, and a total length of supply and return mains of not over 200 feet)

Size of Supply and Return, In	Diameter of Pipe in Coil, Inches	
	3-1/2	2
	Linear Feet of Pipe in Coil	
1-1/4	50	80
1-1/2	80	140
2	190	320
2-1/2	380	650
3	660	1,100
3-1/2	1,000	1,700
4	1,500	2,600
5	3,000	5,000
6	5,200	8,700

Table 29-9

Customary Temperatures for Greenhouses

Type of Plant	Temperature Range, Degrees F
Carnation	45 - 55
Conservatory (general collection, winter, garden, etc.)	60 - 65
Cool	45 - 50
Cucumber	65 - 70
Fern	60 - 65
Forcing	60 - 65
General Purpose	55 - 60
Lettuce	40 - 45
Orchid, warm	65 - 70
Orchid, cool	50 - 55
Palm, warm	60 - 65
Palm, cool	50 - 55
Propagating	55 - 60
Rose	55 - 60
Sweet Pea	45 - 50
Tomato	65 - 70
Tropical	65 - 70
Violet	40 - 45

CHAPTER 30

HEATING SWIMMING POOLS

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CHAPTER 30

HEATING SWIMMING POOLS

There are two common methods for establishing the heat loss of the water in a swimming pool. They are based on:

1. The volume of water
2. The surface area of the water

The volume method considers only the heat necessary to raise the water volume a certain number of degrees fahrenheit in an hour. Arbitrarily, the industry has a set standard for sizing boilers to heat swimming pools on the basis of raising the water temperature by 20° in a 40 hour period in the South and with indoor pools...in a 30 hour period across the middle part of the country and in 20 hours in the North.

These factors can be converted to degrees of water temperature rise per hour:

1. $1/2^{\circ}F$ per hour in the South and in indoor pools
2. $3/4^{\circ}F$ per hour across the middle part of the country
3. $1^{\circ}F$ per hour in the northern part of the country

These are arbitrary and there may be minor differences in any one locality, depending on altitude and humidity.

However, since the quantity of heat estimated at these levels is greater than the heat loss, once the pool is up to temperature this procedure has been found to meet the requirements.

The equation is:

$$\text{Btu/hr Input} = L \times W \times D \times \frac{62}{\text{Boiler Efficiency}} \times T$$

Where L = Length of the pool -- feet

W = Width -- feet

D = Average Depth -- feet

62 = Weight in pounds of 1 cu. ft. of water

T = Degrees of Temperature rise per hour

Actually, of course, any similar equation involving any substance other than water would also include the factor of the specific heat of the substance. Since the specific heat of water is "1", and anything multiplied by "1" is unchanged, it is not necessary to use the factor in this kind of equation when water is the substance.

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Example:

In Connecticut, what size gas-fired boiler, operating at 80% efficiency, is needed to heat a pool 30 feet long, 16 feet wide, with an average depth of 5 feet?

Our equation now is:

$$\text{Btu/hr. Input} = 30 \times 16 \times 5 \times \frac{62}{.8} \times 1 = 186{,}000$$

A boiler with at least a 186,000 Btu/hr. input is required.

If, however, the boiler were rated at 70% efficiency, the equation for the same pool would be:

$$\text{Btu/hr. Input} = 30 \times 15 \times 5 \times \frac{62}{.7} \times 1 = 212{,}570$$

If the pool in question were the same dimensions but in a gymnasium in a school in Connecticut or outdoors in Georgia, the equation for the same boiler, at 70% efficiency would be:

$$\text{Btu/hr. Input} = 30 \times 16 \times 5 \times \frac{62}{.7} \times 1/2 = 106{,}290$$

The area method of "sizing the boiler" is based entirely on the cooling action of both the wind and surface evaporation. Depth of the pool, therefore, is not a factor.

Experience indicates that the pool will lose 12 Btu per hour per square foot of surface area for every degree of temperature difference between the water and the surrounding air temperature.

The area method equation is:

$$\text{Btu/hr. Input} = L \times W \times \frac{12}{\text{Boiler Efficiency}} \times (T_w - T_a)$$

Where L = Length

W = Width

T_w = Temperature of Water in $^{\circ}F$

T_a = Average air temperature in $^{\circ}F$

Example:

Assume the same pool, outdoors in Connecticut. The owner wants to use the pool between September 15th and October 15th. During this period the average air temperature get to $55^{\circ}F$. He wants to keep the pool water at $80^{\circ}F$. Again, a gas boiler with 80% efficiency has been decided on.

The equation:

$$\text{Btu/hr. Input} = 30 \times 16 \times \frac{12}{.8} \times (80-55)$$

or:

$$\frac{30 \times 16 \times 12 \times 25}{.8} = 180{,}000$$

With a boiler rated at 70% efficiency we would have:

$$\frac{30 \times 16 \times 12 \times 25}{.7} = 205{,}700$$

Again, let's move the same pool with a gas boiler operating at 70% efficiency, indoors. Assume the gym is kept in use from 8:00 a.m. to 10:00 p.m. with a space temperature of 70°. After 10:00 p.m., it is allowed, on the thermostat's night setting, to cool to 60°, with the morning cycle set for 7:00 a.m.

Average temperature under these conditions would be 66°. Then:

$$\text{Btu/hr. Input} = 30 \times 16 \times \frac{12}{.7} \times (80-66)$$

or:

$$\frac{30 \times 16 \times 12 \times 14}{.7} = 108{,}350$$

Thus, the two methods of calculating the boiler capacity in Btu/hr. for three different situations involving the same pool size are:

	Volume	Area
Outdoors, Connecticut, 80% efficiency	186,000	180,000
" " 70% "	212,570	205,700
Indoors, " 70% "	106,290	108,350

CHAPTER 31

MAINTENANCE

CHAPTER 31

MAINTENANCE

Regular, methodical maintenance of any heating system pays off in longer life and more economical operation.

Burners -- oil, gas and stokers -- should be checked, cleaned and adjusted according to manufacturer's recommendations at least once a year.

Controls, too, need an annual inspection. Changes in calibration can be detected and corrected. All electrical contacts should be checked and cleaned. Minor pitting can be cured by smoothing...severe pitting only by replacement.

New boilers, either hot water or steam whether in new systems or replacements in old systems, require daily inspection for the first several weeks of operation.

Water should be drained from the boiler at each inspection. If the water is muddy, draining should continue until it runs clear.

Draihing should be accomplished with the water hot and burner off. With dry base boilers more than one drain may be desirable.

The most common problem with new steam boilers is "foaming".

In new steam systems, the condition is aggravated by oil in the piping, left over from the pipe threading operation, which floats back to the boiler with the condensate.

Almost every time a new steam boiler is put into operation oil and sand will be found floating on the water in the boiler, particularly when the boiler is part of an entirely new system.

The strength of the film of oil...surface tension...is much greater than that of water. Instead of the bubble of steam breaking easily through the surface of clear water, it now blows a soap-like bubble.

These bubbles form a froth which blankets the entire surface of the water, which holds back the pressure being built up in the hot water until it becomes strong enough to lift large portions of the blanket and blow it out of the boiler. Often some of the boiler water is lifted with the foam and carried along into the system. This action is called "priming" and results in a rapid lowering of the water level and burner shut-down by the low-water cut-off.

With oil burning boilers, too frequent action by the low-water cut-off has a secondary effect on the stack switch.

Rapid cycling causes the stack switch to go to safety position, preventing the burner from re-starting. An inexperienced service man might replace the stack switch which is operating perfectly and miss the real reason for its action.

In addition to frequent action by the low-water cut-off, there are a number of other "indicators" of foaming.

1. Because of the blanket of foam the boiler does not generate enough steam to fill the radiation.

2. Because of the periodic priming action, water in the gage glass, following the water in the boiler, surges up and down, violently, in wide swings.

3. When the safety valve is opened both water and steam will blow out instead of steam alone.

4. A small amount of boiler water drained into a sauce pan and put on a hot plate will foam up and boil over.

To eliminate foaming, it is common practice with large steam boilers to drain condensate into a sewer for the first couple of weeks of operation.

Periodic "saucepan" tests of the condensate will tell when it is clear of oil and sand and safe to return to the boiler.

The smaller water content of gas boilers, of course, aggravates the foaming problem.

Once a boiler foams and primes, it should be cleaned by the blow-off method.

The chemical treatment is easy but never sure.

The blow-off method is sure but it may have to be done several times... weekly for about three weeks.

American-Standard oil and coal-fired steam boilers are provided with blow-off tappings.

To give a G-60 boiler a surface blow-off the safety-valve tapping may be used. On the G-40 use tapping B. See Technical Catalog Bulletin for G-40.

Remove the safety valve or plug as the case may be and install a nipple, a gate valve and some drain piping.

The water level is raised to the blow off tapping and the boiler fired until steam pressure is developed. The gate valve is then opened and the outrush of steam sweeps the foam off the top of the water and into a drain.

After the pressure has dropped, water is fed into the boiler slowly and the burner kept operating. Any dirt and oil is floated out the blow-off. This process may have to continue for a considerable period depending on the water conditions.

The blow-off is now closed and steam pressure allowed to build up to 10 psig. The gate valve is opened once more to blow-off any additional foam that may have formed.

The burners are shut off and the boiler drained.

After refilling the boiler, start the burner and raise the steam pressure to 10 psig. Open boiler drain to flush any dirty water out of the boiler water legs. In dry base boilers a drain valve should be installed in each leg or on the return manifold.

The water must not be allowed to rise above the blow-off valve as it would trap the dirt and grease above the valve allowing clean water to drain off.

A boiler cannot be cleaned of grease and oil if it is not hot. Merely draining the boiler to clean it will result in surface grease and dirt clinging to its sides, and rising to the surface when the boiler is refilled.

In hot water systems, excess dirt and oil sometimes cause the liberation of substantial amounts of marsh gas which slows water circulation. This situation can usually be handled by a chemical cleaner in accordance with instructions of the cleaner manufacturer.

Boilers not in Operation

In winter, the boiler, returns and system -- if hot water -- should be completely drained. Flues should be cleaned and all access doors left open.

In summer, the boiler flues, especially on coal and oil fired boilers, must be completely cleaned. Gas fired boiler flues stay clean when burners are in proper adjustment and ample air is available for combustion. Leave all access doors open.

Leave water in a hot water boiler and system during summer months.

Fill the steam boiler completely with water during the summer. This stops internal corrosion at the water line during shut down. Be sure to drain excess water out before firing the boiler in the fall.

COMPETITIVE COMPARISONS

CHAPTER 32

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CHAPTER 32

COMPETITIVE COMPARISONS

Competitive comparisons, while not often made specific, are one of the salesman's strongest assets...if he knows them.

This chapter provides these comparisons...as of the date noted on each chart.

Products will change and as they are changed, it is important that you revise these charts so that you are always ready to:

1. Stress your product's advantages.

2. Be aware of competitive claims and have the best answers to them.

Since convenience and cost of assembly are important in any sale, we note several basic assembly factors.

1. Wet base boilers are easier to assemble than dry base boilers because each section has its own base (American-Standard A-5)

   N. B. However certain American-Standard dry base boilers (A-7 and G-60) are furnished with bases to facilitate assembly.

2. American-Standard boiler sections pull tightly together, eliminating the need of asbestos wicking between sections.

3. Smaller sections are more easily handled and assembled than large double sections.

Table 32-1

COMPARATIVE DATA ON G-40
GAS-FIRED BOILERS

Competitive Features	AR & SS G-40	National U.S. 16C	Peerless Series 50	Burnham Holiday 70	Weil-McLain Type H	Bryant 222 (W)	Bryant 223 (S)	H. B. Smith Century 11
Common section for or less tankless	Yes	No	No	No	No	No	No	No
Common tankless heater with steam & water.	Yes	Yes	No	Yes	Yes	No	No	Yes
Available in sections factory assembled.	Yes (up to G-406)	Only 3 & 4 sect.	No	No	No	Yes	Yes	No
Available for dual fuel.	Yes	Yes	No	No	No	Yes	Yes	No
Non-protruding draft hood	Yes	No	No	No	No	No	No	No
Optional jacket extension covering left hand end section.	Yes	Yes	No	No	No	No	Yes	Yes
Available for 50 lbs. PSI	Yes	No	Yes	No	Yes	No	No	No
Integral air eliminator	Yes	No	No	No	No	No	No	No
Deluxe jacket standard	Yes	No	Yes	Yes	Yes	No	No	No
No separate base required	Yes	No	No	No	Yes	No	No	Yes
Internal water heater-capacities steam & water	both 3-9gpm	both $3\frac{1}{2}$-6gpm	----	3-3/4-15gpm for water. 3-7-1/2gpm for steam.	3-10 1/2 gpm	----	----	both 2.2 - 90 gpm

Table 32-3

A-5 Feature Comparisons

Features	AR & SS "A-5"	Burnham "PaceKing"	"25B"	National "27-37 Series"	H. B. Smith 200 &250	Weil-McLain A-72 & A-82
Water Line	46 1/4"	52 1/2"	45"	45 1/4" to 51"	47 3/8" to 57"	47 1/4" to 55"
Wet Base construction	Yes	No	Yes	No	No	No
Section Assembly	A-5 utilizes one set of tie rods & bottom slip nipples	Same	Same	Same	Require short individual tie rods	Requires short individual tie rods, asbestos rope needed to prevent air leakage.
Jacket Extension	Available	Not avail.	Avail.	Not Avail.	Not Avail.	Jacket extension avail on A-72 only.
Boiler-Burner Unit	Yes	No	No	No	No	Yes
Common internal tankless heater with manifold arrangement	Yes	No	No	No	No	Provisions for sing or double domestic water heaters - A-82 only.
Water working pressure	50 lbs. reg.	30 lbs.	30 lbs.	30 lbs.	40 lbs.	30 lbs.
Number & Tapping sizes				"27" "37"		"72" "82"
a) supply	2-4"	4-4"	2-3"	2-3" 2-4"	2-3"	2-3" 2-5"
b) return	4-2 1/2"	2-4"	3-3"	2-3" 4-4"	2-3"	2-3" 2-4"
Internal Water Heater Capacities	4-16gpm (both)	water 4½-15gpm steam 3½-9½gpm	3½-6gpm (both)	"27" "37" 3-7gpm (both)	"200" water 4-8gpm steam 4-6gpm; "250" water 4-12gpm steam 4-6gpm	"72" 4½-6gpm (both); "82" 7-9gpm (dual 16-1

Table 32-2

G-60 Feature Comparison

Features	AR & SS G-60	National-U.S. "66 Series"	Peerless "150 Series"	Weil-McLain Type J	Bryant Model 630
Deluxe Jacket Standard	Yes	Yes	No	Yes	No
Water Working Pressure	50lbs. reg. 80lbs. opt. added charge	30lbs. reg. 50lbs. opt. N.C.	30lbs. reg. 80lbs. opt. Added charge	30lbs. reg. 75lbs. opt. N.C.	30lbs. reg. no options
Height	66 1/8"	69 1/4"	55 1/2"	65 3/4"	67 1/2"
Water Line	44"	45"	41 1/2"	45 3/4"	44"
Boiler Base	Cast Iron	Steel	Cast Iron	Steel framed	Steel
Section Availability	6 to 41	4 to 23	5 to 37	5 to 37	10 to 36
Flue Connection	12" - 30"	all 14" except Model 5(12")	10" - 30"	10",12",14" only.	all 14" excluding Model 10 (12")
Assembly Boiler	G-60 utilizes one set of tie rods & top & bottom nipples on both ends of section	3 nipple sect. cons.	2-nipple sect. cons.	requires short individual tie rods, 2 nipple sect. cons.	2-nipple sect. cons.
No. & size of tappings a) flow b) return	5-6" 5-6"	2-6" 2-6"	2-6" 2-6"	5-6" 2-4"---3-3"	2-6" 2-6"
Available for Dual Fuel	Yes	Yes	No	Yes	No

Table 32-4

A-7 Feature Comparison

Feature	AR & SS "A-7"	National-U.S. "47 Series"	Weil-McLain "HO-40"	H. B. Smith 340 and 440
Domestic Water Heater Arrangement.	Furnished with external heater tapping.	May be equipped with an integral type tankless heater or a storage tank heater.	Not available	Available with manifold heater on special order.
Water Working Pressure.	50 lbs. regular-no options.	30 lbs. regular.	30 lbs.regular-50 lbs.option N.C.	40 lbs. regular 80 lbs. option - Added Charge.
Jacket Availability.	Available with sturdy 20-gauge steel jacket.	Available with jacket.	Available with jacket.	No jacket available.
Assembly Boiler.	A-7 utilizes one set of tie rods & top & bottom slip nipples.	Same as "A-5".	Requires short individual tie rods, multiplicity of parts in excess of slip nipples.	Requires special tools & special upper and lower manifold connections for all sections.
Dry or Wet Top Nipple.	Top nipple part for steam partially submerged in water Boiler rear section tapped for external water heater, eliminating header connections from sections for external heater. 3" tappings furnished regularly for this purpose.	Wet top nipple - moderate tapping and manifolding of sections.	Dry top nipple requiring expensive heater manifold & special tappings.	Dry top nipple required special manifold & heaters.
Height	70 7/8"	69 3/8"	79 1/4"	78" and 76"
Width	49 3/4"	49 1/2"	50"	51" and 64"
Length	41 1/2" - 94"	42 1/4" - 105 1/4"	42 3/4" - 114 3/4"	49" - -- 85"

Table 32-4 (Cont'd)

A-7 Feature Comparison

Features	AR & SS "A-7"	National-U.S. "47 Series"	Weil-McLain "HO-40"	H. B. Smith 340 and 440
Water Line.	59 1/2"	59 1/2"	67"	56" to 61 1/2"
Flue Connection.	16"	16"	18"	18" & 20"
Steel Pedestal Base Option.	Yes	Yes	Yes	Yes
Boiler-Burner Unit	Yes	Yes	Yes	Yes
Tapping Sizes a) Flow b) Return.	5-4" 2-6"	2-6" 4-4"	4-5" 2-5"	"340" "440" 3-5" 3-6" 2-4" 2-5"

GETTING THE ORDER

CHAPTER 33

CHAPTER 33

GETTING THE ORDER

Which, of course, is the sole reason for preparing, writing, printing... AND READING...this book.

Product knowledge is a salesman's byword...the basis for his most potent selling arguments.

The more technical, the more flexible, the more varied, the more it resembles a complex system than a single component, the more important... and potent...knowledge of the product becomes.

This book will not give you all the answers. No single book can. Dozens of sources were used to gather this material.

It was designed to give you a basic understanding of the problems...a method for solving them...and an appreciation of the extent of product knowledge necessary.

The final step in any large heating system sale will often be the formal submission of a proposal with specifications.

This is, of course, a very important step in getting the order and this chapter contains some sample proposals to act as a guide.

Even more important are the inspections, conferences and conversations that precede proposal preparation, and in many cases...particularly in the replacement market...are the only steps to the sale.

Here it is that product and system knowledge pays off. Do you understand the problems? Can you help solve them? Can you suggest better values... either by improving the quality of the system or by finding a lower cost way of accomplishing part or all of the job?

The more money a man spends for a product...the more he wants to be sure that it, the company behind it, and the salesman in front of it are right.

And a large heating system represents a lot of money. Product and installation costs are high, but they're only a fraction of the money he'll be spending on it for the next 20 or 30 years -- or longer.

And so, differences in price in competitive proposals -- occasionally even reasonably large differences -- are sometimes less important than the quality of the salesman's product knowledge.

The important thing he needs is reassurance that your system, your product, will do the job, without overdoing it.

An undersized system will create serious trouble for him -- and you -- and American-Standard.

An oversized system will take money out of his pocket, and in the long run, out of yours -- and American-Standard.

In the early stages, off-the-cuff estimates on costs, Btu/hr. can cover a reasonably wide range.

Sizing a pipe, a pump, a boiler or an expansion tank cannot cover any range at all.

Be precise.

Make your answers to his questions precise.

And make them only when you are absolutely, one hundred percent guaranteed sure.

If you're almost absolutely, ninety-nine and forty-four one-hundredths percent, warranteed sure...don't answer. Tell him you don't know the answer but that you'll get it for him, fast.

Then get it for him...fast!

When you've got all the facts and figures, use the following samples as a guide and make up your proposal.

Sample Specification for Gas Fired Boilers

Contractor shall install where indicated on plans_____gas fired water steam boiler(s) having a net I=B=R rating of_____for automatic operation on_____ gas. Boiler(s) shall be constructed in accordance with ASME Code for Low Pressure Heating Boilers, and bear its symbol, Section IV. Boiler(s) shall be A. G. A. approved for a maximum input of_____BTU per hour. Boiler(s) shall be constructed of cast iron sections with vertical flues with one atmospheric drill port or removable ribbon burner per flue way.

Manufacturer shall supply each boiler with main automatic gas valve, safety pilot with thermocouple as required by A. G. A., Gas pressure regulator (except for LP), high limit control, water or steam boiler trim as required by ASME, integral canopy and drafthood. (For G-2 and G-40 boilers an air eliminator as an integral part of the boiler and tankless coil of_____ GPM capacity)

Additional equipment required as follows:
(Select one of the following specific equipment desired)

Manufacturer will furnish, in addition to the above, basic additional controls necessary to meet F. M. & F. I. A. requirements including electronic controls to provide electronic supervision and all primary gas pilots and electric ignition on all pilots and 100 per cent safety shutoff. Boiler(s) shall, after equipment is properly installed by contractor, be fully automatic in operation with intermittent pilot ignition on all pilots.

_____or_____

Manufacturer will furnish boiler, in addition to the above electronic controls, to provide the following: (select one of the following)

1. Manual ignition non 100 per cent shutoff continuous pilot
2. Manual ignition 100 per cent shutoff continuous pilot
3. 100 per cent shutoff with electric ignition on primary boiler pilot providing equipment component check of each boiler cycle.

Manufacturer will furnish factory wired and assembled electronic control panel suitable for wall hanging with main power on and off switch, pilot and gas valve switch, indicator lights for power on, pilot off, gas valve on.

Boiler shall be as manufactured by American-Radiator & Standard Sanitary Corporation, New York, New York, Model No._____or approved equal.

Sample Specification for Oil Fired Boilers

Contractor shall install where indicated on plans_____oil fired steam or water boiler(s) having a net I=B=R rating of_____at a firing rate of _____gph. Boiler(s) shall be constructed in accordance with ASME Code for Low Pressure Heating Boilers, Section IV. Boiler(s) shall have a gross I=B=R output of_____BTU/hr.

Manufacturer shall furnish each boiler with steam or water trim in accordance with ASME requirements, with UL approved oil burner designed and tested for boiler described above. Controls shall include high limit control, primary control, fuel pump, draft stabilizer. Where required, manufacturer will furnish special pedestal base to raise boiler to correct height to provide proper combustion area required by burner.

Additional equipment required for firing rate of 7 gph or more:
(Select one of following specific equipment desired)

Boiler manufacturer will furnish oil burner equipped with flame safeguard equipment with basic controls to meet F. M. and F. I. A. approval with control panel to provide 15 second trial for ignition and 2 to 4 second flame failure shutdown. Photocell mounted in burner. Oil valve, motor transformer and Photocell to be factory wired.

_____or_____

Boiler manufacturer will furnish oil burner equipped with flame safeguard equipment as required by UL to provide 15 second trial for ignition and 15 second flame failure shutdown. Photocell mounted in burner. Oil valve, motor, transformer and Photocell to be factory wired.

Boiler & Burner shall be as manufactured and equipped by American Radiator & Standard Sanitary Corporation, New York, New York, Model No._____or approved equal.