AMTROL®

ENGINEERING HANDBOOK

For the Designer of Hydro-Pneumatic Applications in Engineered Mechanical Systems...

This design handbook concerns the design, selection and application of hydro-pneumatic components in commercial, industrial, institutional and high-rise residential mechanical piping systems.

Hydro-pneumatics—Can best be defined as the use of a gas, (usually air), in a liquid-filled piping system to control operating pressure, liquid expansion, and water hammer during system operation in applications for space heating and cooling; water heating (potable and process); and water supply applications (well systems and pressure booster systems).

Section A, beginning on page A-1, will deal with the fundamentals of hydro-pneumatics in liquid heating and cooling systems.

Section B, beginning on page B1, will cover sizing and design requirements, sizing procedures, installation and application variations of EXTROL hydro-pneumatic (expansion) tanks for hot water and chilled water hydronic systems.

Section C, beginning on page C-1, will deal with air removal procedures in large hot water heating and cooling systems.

The contents of each section are provided to aid the professional designer, specifier and estimator of engineered mechanical systems in selecting and applying hydro-pneumatic products as originally designed and produced by AMTROL Inc., to arrive at the most efficient and dependable system operation possible. In the future, the contents of this binder will be up-dated as new breakthroughs in the ever-changing mechanical systems field are made.

The Reference Section contains appropriate data tables as referenced throughout the text.

Section A

Hydro-pneumatics in hot water and chilled water systems

Chapter One

Pressurization of Hot Water Heating and Chilled Water Systems

The basic function of hydro-pneumatics in a closed piping system transmitting water or a similar liquid for heat transfer or cooling applications is over-all systems pressurization, and the maintenance of minimum-maximum pressures during all operating temperatures of the system.

Closed System vs. Open System

Closed System Advantages:

The advantages of a closed piping system in hot water heating and chilled water cooling as compared to an open system are many:

1. Minimum make-up water required.

1. Little, if any, make-up water is required to maintain a filled system as little or none is lost through evaporation or steaming as in an open system where the water content of the system is open, at some point, to the atmosphere.

2. No accumulation of oxygen from constant make-up water.

2. With very little fresh water added, there will be no meaningful accumulation of oxygen and other entrained corrosive agents in the system once it is initially purged. This extends the life expectancy of the system indefinitely. In an open system, on the other hand, entrained air and other corrosive agents accumulate rapidly through the constant entrance of make-up water.

3. Operating temperatures above 212°F.

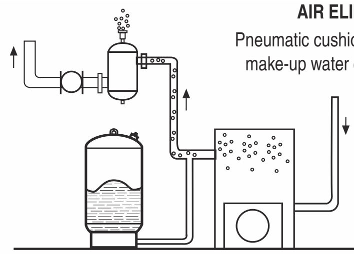
3. In a closed system the entire system is pressurized so that operating temperatures can be higher than 212°F without causing flashing or boiling in the system. This allows reduced sizing of piping, terminal units and heat generators. In an open system, maximum operating temperatures must be kept below

4. Operating economy and conservation of water and energy.

4. In a closed heating system, there is very little or no loss of water or heat as system water remains in the piping. During the evaporative process in a system open to the atmosphere, there is a constant loss of water and BTUs. Constantly entering cool make-up water requires frequent burning of fuel to maintain system temperatures.

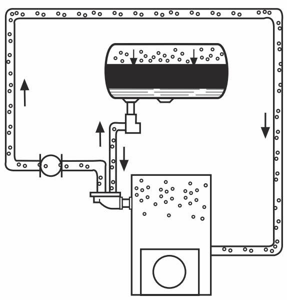
5. Positive pressure assures constant venting.

5. In a closed system, pressurization can maintain a positive pressure in the system at all times so that excess air can be eliminated through simple automatic air venting devices. In systems open to the atmosphere, pressures in a negative range can draw air into the system creating corrosive and circulation problems.



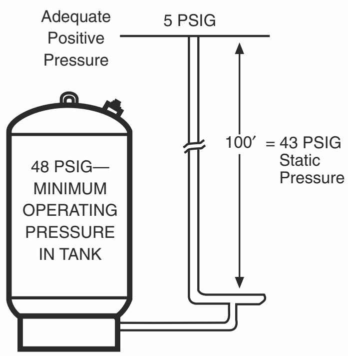
AIR ELIMINATION

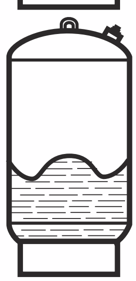
Pneumatic cushion permanent. Air in make-up water can be eliminated.



AIR CONTROL

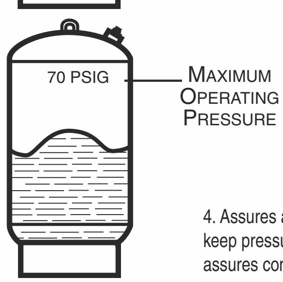
Pneumatic cushion constantly absorbed in system water and circulated throughout system. Must be collected and directed back to tank to maintain pressurization.





1. Pneumatic cushion eliminates need of charge water during system fill with a minimum operating pressure on system.

2. Accepts expanded water as system temperature rises.



3. Limits maximum system operating pressure to within safety setting.

4. Assures a positive pressure at all times to keep pressure above "flash" point of water and assures constant venting.

Air Elimination...Not Air Control

It has been thought for many years, that for ideal pressurization of a closed hot water heating or chilled water system, a perfectly and hermetically sealed piping system is mandatory for ideal air-free performance. While this is in theory ideal and should be the objective in any designer's mind, it is also acknowledged that such a completely pressure-tight, leak-proof state is understandably difficult to insure. Especially in a large, complicated engineered mechanical system such as found in the modern high-rise building.

Therefore, facing the reality that the ideal may not be reached, the designer of a closed piping system should incorporate into the basic system a provision for continuous and automatic elimination of excess air that will surely find its way into the system through fresh water which must be added to the system.

While the traditional approach to air, as the pressurizing agent in the system, has been one of controlling air (i.e., collecting and re-directing it to the compression tank), this manual will discuss the use of the EXTROL concept of permanently separating the air used for pressurizing the system from system water. Therefore, any other excess air may be collected and eliminated from the system as soon as it enters. This subject of air removal versus air control will be fully covered in Section C.

The Function of the Hydro-Pneumatic Tank

The primary device in pressurizing and maintaining pressure control in a closed system is the hydro-pneumatic tank, also known, traditionally, as the expansion or compression tank.

Its function in the pressurization process is as follows:

1. Through the use of a pneumatic cushion (air), it maintains positive minimum pressure throughout the system when it is initially filled.

2. During temperature rises, it provides an additional space in the system for the expanded volume of water that results. This is accomplished as the pneumatic cushion is compressed as system pressure increases, creating additional space for the increased volume of water. As the system temperature drops, the compressed pneumatic cushion forces water back into the system, maintaining a positive pressure on the system during all temperatures in the system's operating range.

3. Properly sized, the hydro-pneumatic tank will maintain maximum system pressures within the working pressure limitations of the system equipment and components.

4. By maintaining a positive pressure on the system throughout all the operating temperature range, the hydro-pneumatic tank enables the designer to constantly vent excess air through the use of automatically operating float type air vents.

PRESSURIZATION OF WATER HEATING AND CHILLED WATER SYSTEMS

The selection of the hydro-pneumatic tank will establish a base for determining the over-all pressurization of the systems. Before discussing the selection and sizing procedure, it would be advisable to evaluate the various pressure and temperature factors and their relation to pressure control.

A. Minimum Operating Pressure at the Tank

This is the pressure required, at the tank location, to keep the system completely filled, plus an additional pressure to insure positive pressure at the top of the system. In heating, this positive pressure must be above the flash point of the water at its maximum operating temperature.

In determining the minimum operating pressure, two factors must be considered:

1. Static Pressure at the Tank Location – This is the pressure caused by the physical weight of the water above the tank. Since 2.31 feet of water exerts a pressure of 1 pound per square inch gauge (psig) the height of the system above the hydro-pneumatic tank divided by 2.31 will give the static pressure at that point. This is the pressure required to fill the system to the top.

The height of the system above the plain steel hydro-pneumatic tank will be the pressure to which the system water will compress the air cushion in the tank when the system is filled.

In a high-rise system, for example, with the tank located in the basement of a building and 100 feet of water above it, the static pressure at the tank would be approximately 43 psig.

2. Adequate Positive Pressure – To the static pressure above, it is necessary to add an additional positive pressure at the top of the piping system when it is initially filled. This insures positive pressure at all times so that air can be constantly vented and negative pressures can not occur to draw air into the system through vents. A standard practice is to add at least 5 psig to accomplish this. (Figure A-1)

In a heating system, a positive pressure also is required at all system points to keep system operating pressures above the “flash” or boiling point of the water.

This flash point is determined from the basic steam table which indicates the minimum pressures at various temperatures, in the operating range, which must be maintained to keep water in a closed piping system from boiling, or “flashing” to a vapor (steam).

Without this positive pressurization, inadequate system pressures at the pump, when operating at higher temperatures, cause noise, loss of circulation, “flashing”, and usually result in serious damage to the pump.

In establishing an adequate positive pressure for a given system, it is good practice to add a safety margin of 15°F to the design temperature, and then refer to the steam table to determine the minimum positive pressure. This has been calculated and shown in Table 4 in the Reference Supplement (see page REF-6). Excerpts for purposes of illustration are also shown on the following page.

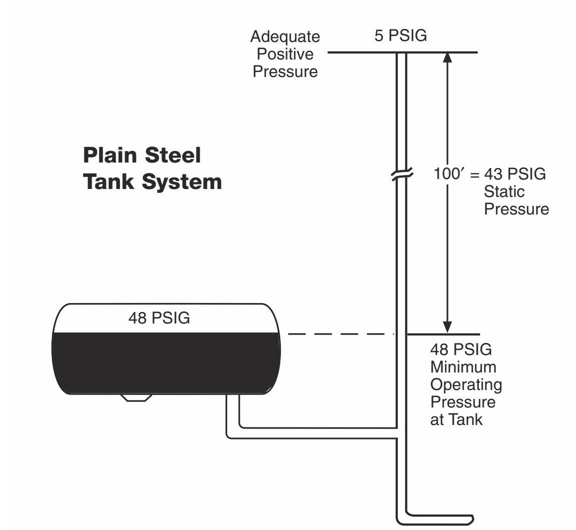


Fig. A-1

43 PSIG +	5 PSIG =	48 PSIG
Static Pressure	Adequate Positive Pressure	Minimum Operating Pressure at Tank

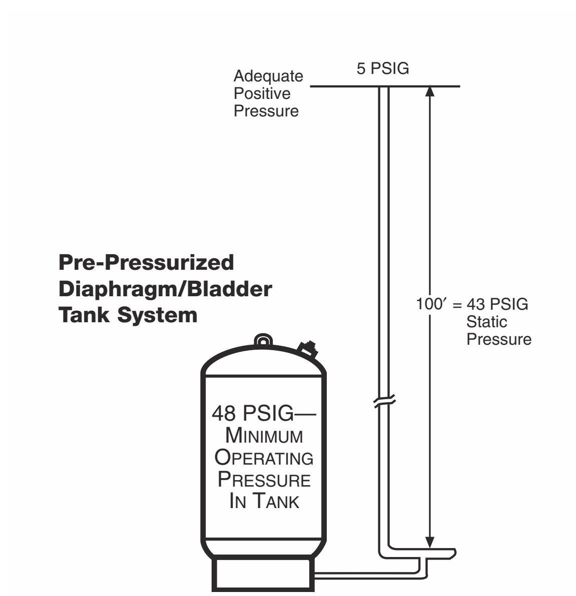


Fig. A-2

200°F Design Temperature

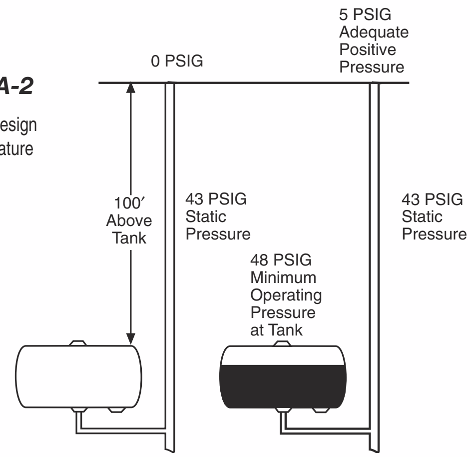


Fig. A-3

260°F Design Temperature

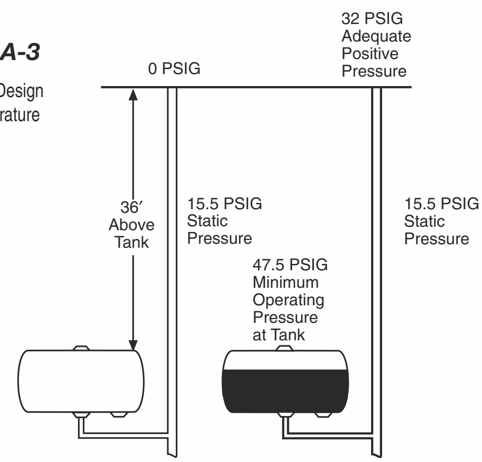


Table 4: Adequate positive pressures to prevent "flashing"

Based on tables in "Thermodynamic Properties of Steam" by Keenan & Keyes, published by John Wiley & Sons, Inc.

The following table includes a safety factor of 15°F.

Max. Operating Temperature in °F	Adequate positive pressure must be at least:
200°	2 PSIG *
205°	3 PSIG *
210°	5 PSIG
215°	7 PSIG
220°	9 PSIG
225°	11 PSIG
230°	13 PSIG
235°	16 PSIG
240°	19 PSIG
245°	21 PSIG
250°	25 PSIG
255°	28 PSIG
260°	32 PSIG

Recommended Adequate Positive Pressure to insure* ***Positive Air Venting* *at the top of the system should be* ***at least 5 psig****.*

The minimum operating pressure at the tank, then, must:

1) Be sufficient to fill the system to the top (static pressure).
2) Provide additional pressure to insure *positive venting* of excess air released during all operating stages of heating and cooling systems.
3) Provide additional pressure to insure an *adequate positive pressure* at all heating system points so that boiling will not take place.

In the example given at left, (Figure A-2), a static pressure of 43 psig is required in the tank at the bottom of the heating system to fill it completely.

If we operate the system at 200°F, the *adequate positive pressure* must be *at least 2 psig* at all *system points* to prevent the possibility of "flashing" or boiling. (See Table 4)

At the tank, then, a pressure gauge, installed in the line, will read 43 psig when the system is completely filled.

At this location, the static pressure is more than sufficient to prevent boiling or "flashing".

But at the highest point of the system (100' above the tank), a gauge will indicate near 0 psig as there is no water or static pressure above. At this system point, if a temperature override occurred, water could begin to boil and turn to steam.

Therefore, the minimum operating pressure at the tank must be – a *static pressure* of 43 psig at the bottom, plus *adequate positive pressure* of at least 2 psig to prevent the possibility of "flashing" at the system's highest points. However, in the example, we have already established an arbitrary positive pressure of 5 psig to assure positive venting. So we have more than enough positive pressure to prevent "flashing".

In this example, the *minimum operating pressure* in the tank (static plus adequate positive), is 48 psig.

Now let's examine a system with a static height of 36' above the tank and operating at 260°F. (Figure A-3). The *static pressure* required to fill the system would be 36 ÷ 2.31 or 15.5 psig.

Referring to Table 4, we find that an *adequate positive pressure* to prevent "flashing" at 260° F. is 32 psig.

Therefore, in a system 36' high and operating at 260°F., we would require a *positive pressure* of 32 psig at the top of the system, plus 15.5 psig static pressure, or 47.5 psig *minimum operating pressure* at the bottom of the system.

> *Note: If the compression tank is located at some distance from the suction side of the circulating pump, subtractive pressure effect will result at all system points between the compression tank and the suction side of the pump when the pump operates. The designer should consider this carefully, as pressure may fall below the adequate positive pressure required to prevent flashing. See Chapter 1, Section B for further details.*

In a plain steel tank, the minimum operating pressure is created by filling the system beyond the static pressure until the air trapped in the tank is compressed to the minimum operating pressure. In the EXTROL diaphragm-type tank, this minimum operating pressure (static plus minimum positive) is already in the permanent pre-charged air cushion.

B. Maximum Allowable Pressure at all System Points

The maximum pressure allowable at any given point in a hot water heating or chilled water cooling system is that pressure within the limits at which the boiler, chiller, and/or other system components are rated to operate. A relief valve, installed at the location of each major component, is set to equal the component's rated maximum working pressure. It prevents system pressures at that point from exceeding the pressure rating of the component by automatically opening, once the rated pressure is reached, releasing system water and thereby reducing the pressure below the danger point.

However, the relief valve is a safety device and is not to be used as a working component of the system. Therefore the designer must select a maximum *allowable* pressure for each component location to be protected. This maximum *allowable* pressure must be kept at some point *below* the relief valve setting, or rated maximum pressure of the component(s) to be protected.

In view of the fact that most relief valves will begin to "weep" before the safety relief pressure is reached, it is good design practice to select a maximum allowable pressure at some point below the safety relief valve "set" pressure tolerances which have been specified in the A.S.M.E. Boiler and Pressure Code, "Pressure Settings of Safety Devices".

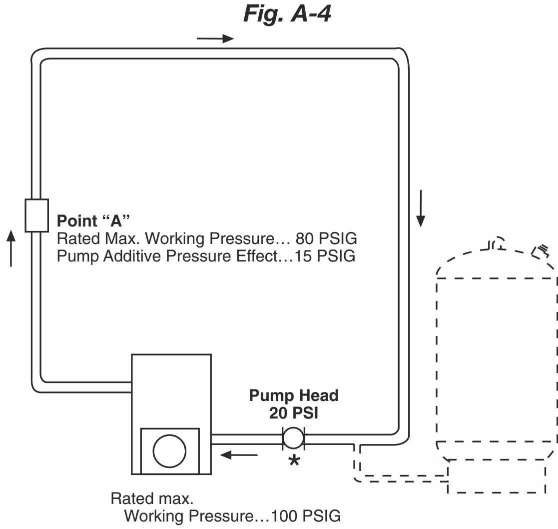
A recommended standard is at least 10% below the relief valve setting.

Pump Operation Also Affects Maximum Allowable Pressure

Before establishing the maximum allowable pressure at each component location, the designer must also consider the possible *additive pressure effect* at each component location that will be caused by operation of the circulating pump(s). Its magnitude depends upon the friction loss between the component's location and the connection of the compression tank in the direction of system flow. (Chapter One in both Section B and Section C will present a more detailed examination of pump operation effect on heating and cooling system pressures.)

Once the amount of additive pump pressure is determined, it should be subtracted from the relief valve setting minus 10% to arrive at the correct Maximum Allowable Pressure at each component location.

Fig. A-4



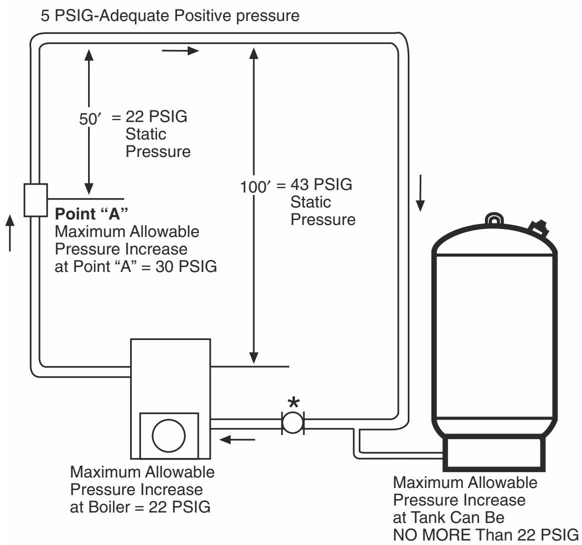
*Not recommended pump location. For example illustration only.

Example Illustrating
Maximum Allowable Pressure

In the example previously given (Figure A-4), if we add a boiler rated at 100 psig and a system component rated at 80 psig at Point "A" as shown, and assuming a pump additive pressure effect of 20 psi at the boiler and 15 psi at Point "A", the following Maximum Allowable Pressures would be calculated:

Boiler Rated Maximum Working Pressure	
(Relief Valve Setting)	**100 psig**
Minus at least 10%	**– 10 psig**
	90 psig
Minus Pump Additive Pressure Effect	**– 20 psig**
MAXIMUM ALLOWABLE PRESSURE AT BOILER.	**70 psig**
Point "A" Rated maximum Working Pressure	
(Relief Valve Setting)	**80 psig**
minus at least 10 %	**– 8 psig**
	72 psig
Minus Pump Additive Pressure Effect	**– 15 psig**
MAXIMUM ALLOWABLE PRESSURE AT POINT "A"	**57 psig**

Fig. A-5



*Not recommended pump location. For example illustration only.

Example Illustrating

Maximum Allowable Pressure Increase on System

In the example given (Figure A-5), if we calculate the allowable pressure increase at both the boiler and at Point "A", we can determine the Maximum Allowable Pressure Increase for the entire system:

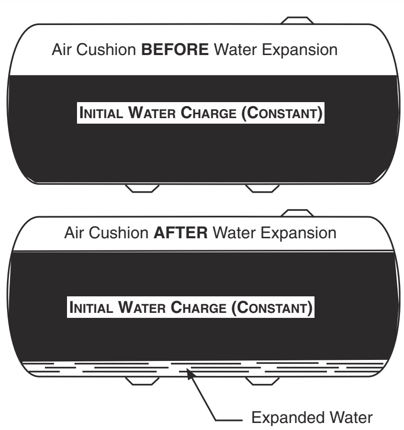
Maximum Allowable Pressure at Boiler	**70 psig**
Minus Minimum Pressure at Boiler (Fig. A-2)	**– 48 psig**
MAXIMUM PRESSURE INCREASE AT BOILER	**22 psig**
Maximum Allowable Pressure at Point "A"	**57 psig**
Minus Minimum Pressure at Point "A" (Fig. A-5)	**- 27 psig**
MAXIMUM ALLOWABLE PRESSURE INCREASE AT POINT "A	**30 psig**

The smallest – 22 psig – will be the maximum allowable pressure increase at all system points, including the compression tank.

Example Illustrating

Maximum Operating Pressure at Tank

Minimum Operating Pressure at TANK	**48 psig**
Plus Maximum Allowable Pressure Increase	**+ 22 psig**
MAXIMUM OPERATING PRESSURE AT TANK	**70 psig**



C. Maximum Allowable Pressure Increase on the System

This is a fixed pressure value applicable to all system points. Since a compression tank of the correct size will maintain the overall pressure range (min. to max.) on the entire system, the maximum allowable pressure increase for the system must be determined prior to compression tank sizing.

To determine the maximum allowable pressure increase for the entire system, the designer must calculate an allowable pressure increase for each component location to be protected and then choose the smallest increase as the system's *maximum* allowable pressure increase.

The calculation is simply made by subtracting the minimum pressure, at each location, from the maximum allowable pressure at each location. The result will indicate the allowable pressure increase for each component location. The *smallest* of these is the Maximum Allowable Pressure Increase for the entire system.

D. Maximum Operating Pressure at the Tank

The function of the hydro-pneumatic compression tank is to limit pressure increase throughout the system to within the range of the Maximum Allowable Pressure Increase.

For proper sizing of the tank to accomplish this, it is necessary to determine what the final or maximum operating pressure in the tank's air cushion will be when the pressure throughout the system has reached the upper limit of the Maximum Allowable Pressure Increase.

Once the Minimum Operating Pressure in the tank (Pages 4-5), and the Maximum Allowable Pressure Increase for the system are determined, the sum of these two pressure valves will be the Maximum Operating Pressure allowable in the tank.

E. Temperature Change and Water Expansion

Water expands when heated and contracts when cooled. It expands and contracts in proportion to the temperature changes in the system. Since water cannot be compressed, the expanded volume must be accommodated at some point in the system. Even in chilled water lines with a comparatively small temperature increase, expansion does occur and must be provided for in the closed system.

The air cushion in the hydro-pneumatic tank provides the space for expansion. As the temperature increases, and water expands, the resulting pressure increase compresses and reduces the volume of the air cushion accommodating the increased volume of system water.

In selecting the properly sized tank, it is necessary to determine the amount of expanded water which will have to be accommodated in the tank. This determination is made by multiplying the coefficient of expansion between minimum and maximum system temperature by the total water content of the system (including boiler/chiller or heat exchanger, radiation and terminal heat distributors, and piping or tubing).

Amount of Expanded Water Based on System Water Content and Maximum Average Design Temperature.

The total volume of system water will expand as system temperature increases. To determine the exact additional volume of expanded water, the System Water Content is multiplied by a Net Expansion Factor which is derived from the Smithsonian Tables for Relative Volume of Water and an acceptable co-efficient to express the expansion of metallic system components.

For a design temperature of 200°F. the net Expansion Factor is 0.0351. (Based on an initial temperature of 40°F.)

F. Compression Tank Sizing is Based on Pressure Increase at the Tank Location and the Amount of Expanded Water.

With the Minimum Operating Pressure and Maximum Operating Pressure at the tank determined, and the amount of expanded water the system will generate properly calculated, proper tank sizing can be accomplished.

Tank operating pressure range determines acceptance factor

Using the Minimum Operating Pressure and the Maximum Operating Pressure in the tank, an acceptance factor can be computed to determine the total tank volume required to contain the initial water charge (in plain steel tanks only), the air cushion at Maximum Operating Pressure, and the volume of expanded water at the Maximum Design Temperature. In diaphragm-type tanks, no initial charge water is required.

A formula (derived form Boyle's Law) to use is as follows:

For plain steel tanks - $\frac{P_a}{P_f} - \frac{P_a}{P_o} =$ Acceptance Factor

For diaphragm-type tanks- $1 - \frac{P_f}{P_o} =$ Acceptance Factor

Where:

P_a = Pressure in the plain steel tank when it is installed empty, usually atmospheric.

P_f = Initial, or the Minimum Operating Pressure at the tank's location, in pounds per square inch, absolute.

P_o = Final, or Maximum Operating Pressure at the tank's location, in psia.

Expanded water divided by acceptance factor will give total tank volume required.

G. Importance of Thorough Evaluation of Pressure Values.

Since the function of the hydro-pneumatic tank is to control pressure within the maximum allowable pressure increase at all system points, it can be seen why a careful evaluation of pressures and their relationships should be conducted prior to compression tank sizing.

This evaluation will also aid the designer in his choice of location for the compression tank and other components, in order to achieve the most critical and economical sizing.

The necessary steps in conducting this pressure evaluation will be covered in detail in Chapter 1 of both Section B for heating, and Section C for cooling.

System Water content . . . . 2640 Gal.
Maximum average design temperature. . . . . 200°F

To Calculate Total Tank Volume Required

1. Determine amount of expanded water
   2640 x 0.0351 = 92.6 gallons, expanded water
2. Determine acceptance factor

   For PLAIN STEEL TANKS:

   $$\frac{P_a}{P_f} - \frac{P_a}{P_o} = \frac{14.7}{62.7} - \frac{14.7}{84.7} = 0.0608 \text{ Acceptance Factor}$$

   FOR DIAPHRAGM-TYPE TANKS:

   $$1-\frac{P_f}{P_o} = 1 - \frac{62.7}{84.7} = 0.260 \text{ Acceptance Factor}$$

3. Divide expanded water by acceptance factor

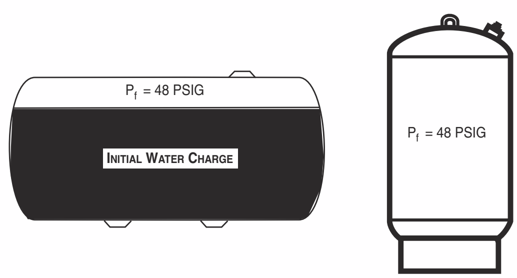
   FOR PLAIN STEEL TANKS:

   92.6 ÷ 0.0608 = 1520 Gallons, Total Tank Volume

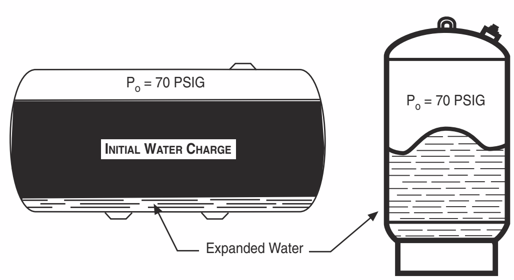
   FOR DIAPHRAGM-TYPE TANKS:

   92.6 ÷ 0.260 = 356 Gallons, Total Tank Volume

Minimum Operating Pressure at Tank



Maximum Operating Pressure at Tank



Hydro-Pneumatics in hot water and chilled water systems

Chapter Two

Selecting the type of hydro-pneumatic expansion tank

In today's engineered building field, space, time and energy consumption have become critical factors that must be considered when designing and installing large heating and cooling systems using a liquid (usually water) as the transfer medium.

When sizing and selecting the hydro-pneumatic tank for such systems, the designer must accomplish full system pressurization and accommodation of the system's expanded water throughout the operating temperature range.

But he must accomplish this with as small and light a tank as possible to reduce the space/time/energy factors involved.

Cost (initial + installation) + maintenance + energy
life-cycle

Space Factor - Less equipment space means more revenue producing space

Large bulky tanks, with the required support materials, take up valuable space in equipment rooms. By reducing the space requirements of hydro-pneumatic tank installations, the designer can give the owner more space for the production of revenue or use by the building inhabitants. This space factor has become very critical in the over-all design of engineered systems.

Time Factor - Lower installation time means lower total costs

With the soaring cost of labor, the time of installation required can affect the total over-all cost of the job. In many cases, a system component may be comparatively inexpensive to purchase. But the installation time required to install it and place it in operation may result in a total installed cost that is more expensive than a similar component with a higher initial cost.

This is especially true in the case of hydro-pneumatic tank installations where large heavyweight tanks require a large amount of installation time, additional support materials, and accessory equipment.

Energy Factor - Critical sizing to reduce energy consumption

Perhaps the most critical factor in the mechanical systems field is the concern of energy consumption of all mechanical systems. Designers can no longer afford the luxury of conservative sizing and over-sizing components. Critical standards of performance efficiency require critical sizing, eliminating all areas of possible wasteful use of energy.

In the case of hydro-pneumatic tank installations, large tanks with enormous initial water charges add to the energy consumption, as they may double the system water content. Heat loss through uninsulated bare steel tanks is another area of energy waste.

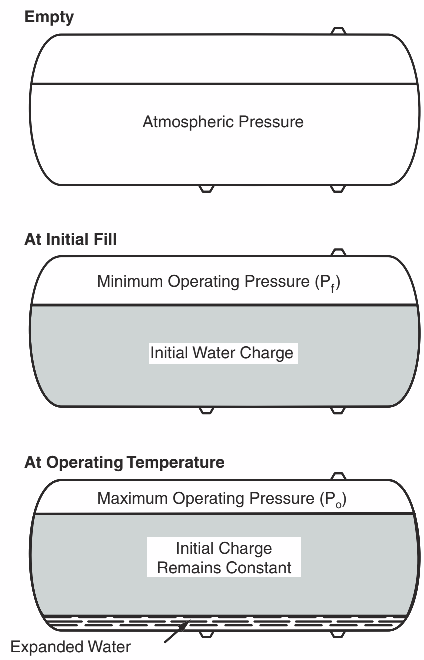
Life-Cycle Cost Factor - Buildings Must Last

Modern buildings are being designed and erected to last longer than buildings erected in the past. In the complex economics of investing in a building, owner groups insist on extended life-cycles for buildings.

This means that mechanical systems must be designed that will afford the lowest possible maintenance requirements over a period extended to twenty-five or thirty years.

In the area of hydro-pneumatic tanks, periodic maintenance and inspection, waterlogging, bleeding down and recharging, are all costs that adversely affect the life-cycle cost of the system. And should be avoided in initial design and selection, if at all possible.

Before selecting the type of hydro-pneumatic tank for the large engineered system, it would be advisable to discuss the space/time/energy requirements of hydro-pneumatic tanks. There are currently three basic types of tanks used, plain steel tanks, diaphragm-type pre-pressurized tanks and replaceable bladder pre-pressurized tanks.



A. Plain Steel Tanks

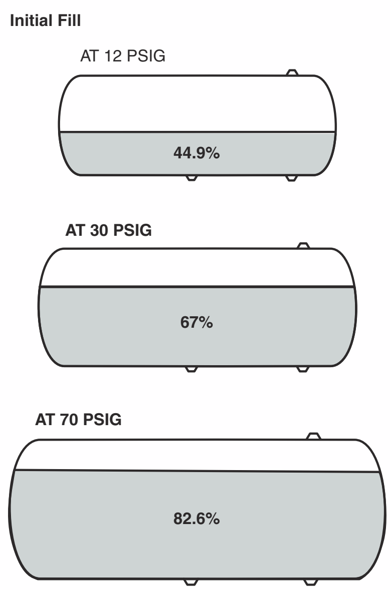
This type of hydro-pneumatic tank, also called the expansion or compression tank, is generally of cylindrical black steel, constructed and stamped according to A.S.M.E. standards for unfired pressure vessels. In larger systems they may be installed in multiples instead of a large single tank.

Space & Weight Requirements High

The plain steel tank operates on a basis of being installed empty. The tank contains air at atmospheric pressure. When the system is initially filled, system water enters the tank, compresses the air until the system is filled and the air is at the minimum operating pressure (P_f) at the tank location.

This amount of water to compress the air is in the initial water charge. Based on the minimum operating pressure (P_f) at the tank, it could be as high as 80% of the total tank volume. This initial water charge must always remain in the tank to keep the system constantly pressurized. It should not be confused with the expanded volume of water that results when the system operating temperature increases. The expanded water is an *additional volume* which the tank must be sized to accept. By keeping in mind the axiom that "the higher the initial pressure, the larger the tank," we can establish a ratio of tank size to initial pressure for the plain steel tank.

This ratio of the portion of tank volume required for initial charge water to total tank volume increases sharply as initial pressure increase.



At 12 psig, for example, 44.9% of the total tank volume of a given plain steel tank is required for initial charge water. At a 30 psig initial pressure, 67% of the tank volume is required. And at 70 psig, 82.6% is required.

In small, non-engineered residential systems, these ratios are of little consequence. A 12 gallon tank requires a little over five gallons of charge water, and is not much bigger than a 7 gallon pre-pressurized expansion tank that requires no charge water at all.

But in large engineered systems, with high initial pressures, high operating temperatures, and/or extremely large total system water volumes, these ratios can mean that a plain steel tank can take up as much as seven and a half times the space, and weigh ten times as much as a pre-pressurized hydro-pneumatic tank.

For example (Figure A-6):

Assume a system with 100' of static head above the tank, a minimum operating pressure (P_f) of 48 psig and a maximum operating pressure of (P_o) 70 psig. The system contains 2640 gallons and the maximum design temperature is 200°F.

A single plain steel tank to service this system, installed close to and above the boiler at the system's lowest point, would be approximately 6' in diameter and 10' long. With the system operational and at design temperature, the weight of the tank, initial water charge and expanded water would be over 14,500 lbs. This type of installation would require special structures to support the tank.

B. Diaphragm-type or replaceable-bladder pre-pressurized hydro-pneumatic tanks

The diaphragm-type tank differs sharply from the plain steel tank in design and construction. More than just a tank, it consists of a steel shell enclosing a sealed-in heavy-duty flexible diaphragm which divides the interior into two permanently separated chambers, one to contain expanded system water, and the other containing pre-pressurized air cushion. When the tank is installed the diaphragm is fully flexed downward so that the entire tank volume can be filled with air. This air is pre-charged to the minimum operating pressure of the tank before it is installed.

When the system is filled, the pre-charged air cushion keeps water from entering the tank. As the system pressure increases above the pre-charged pressure (P_f), expanded water enters the tank. The only volume of the tank used for water is that required to accept the expanded water. This concept is also true for the replaceable bladder pre-pressurized tanks.

Space & weight requirements low

It can be easily understood that by eliminating the need of a large volume of initial system water in order to charge the tank's air cushion, the diaphragm-type tank can be but a fraction of the size of the steel hydro-pneumatic tank. For high initial pressures, the pre-charge is proportionately higher. While over-all tank volume may increase slightly to provide for a larger total air volume, the ratio of increase is considerably less when compared to that of the plain steel tank.

In the example given, a diaphragm-type tank that is floor mounted to service the same system that a 6' by 10' 4150 pound tank services, that would be floor mounted, would require only 7 square feet of floor space and stand 100” high. When installed it weighs 968 compared to 13,800 pounds. When the design temperature is reached, its weight would be 1737 pounds.

Fig. A-6

Example System:

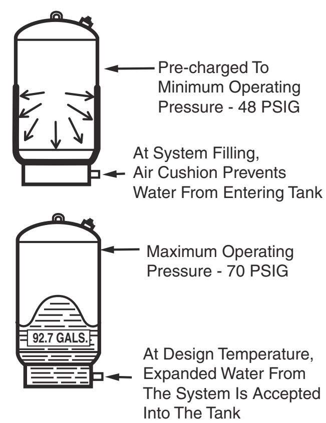
Total System Water	2640 Gal.
Static Head at Tank	100 Ft.
Max. Average Design Temp.	200°F
Minimum Operating Pressure (P_f)	48 PSIG
Maximum Operating Pressure (P_o)	70 PSIG

Plain Steel Tank

Size (Total Volume)	1520 Gal.
Weight Empty	4150 Lbs.
Weight at Minimum operating pressure (P_f)	13,800 Lbs.
Weight at Maximum Operating Pressure (P_o)	14,569 Lbs.

Diaphragm-Type Tank

Size (Total volume)	356 Gal.
Weight Empty	968 Lbs.
Weight at minimum Operating pressure (P_f)	968 Lbs.
Weight at maximum Operating Pressure (P_o)	1,737 Lbs.

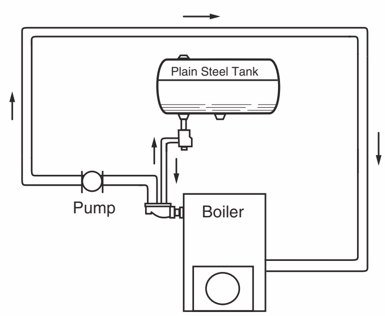


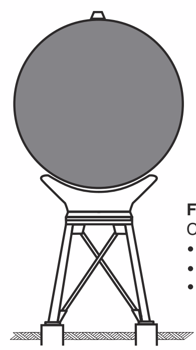
Space Ratio of Hydro-Pneumatic Tanks

% minimum Tank volume (Gallons) Diaphragm to Plain Steel Tanks.

Minimum Operating Pressure In Tank	Space Ratio % of Plain Steel Tank Volume
12 PSIG	.55
20 PSIG	.42
30 PSIG	.32
40 PSIG	.26
50 PSIG	.23
60 PSIG	.19
70 PSIG	.17
80 PSIG	.15
90 PSIG	.14
100 PSIG	.13

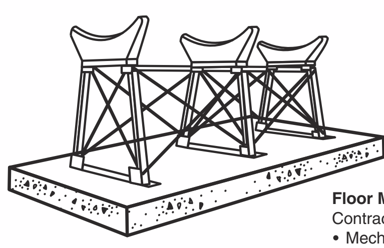
Time and Installation Cost Factors for Hydro-pneumatic Tanks



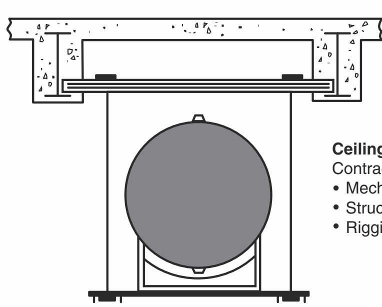


Total Weight
Over 7 Tons

Floor Mounting
Contract Jurisdiction:
- Mechanical
- Structural
- Rigging



Floor Mounting Pad
Contract Jurisdiction:
- Mechanical
- Structural
- Rigging
- Masonry



Ceiling Hung
Contract Jurisdiction:
- Mechanical
- Structural
- Rigging

Total Weight
Over 7 Tons

Plain steel tanks

In general, plain steel hydro-pneumatic tanks are installed close to and above the boiler or chiller. This location is required to provide proper air control to insure maintenance of the tank's air cushion and prolong the period before water-logging of the tank takes place. (see Section C, "Air Removal") It is also necessary to limit operating pressure on the boiler or chiller to within its rated working pressure.

When installing the plain steel tank as outlined above, certain time and material cost factors must be considered by the designer during the original stages of system design:

1. Floor Supporting - In some cases the designer may elect to support a single large plain steel tank, or a battery of smaller ones, from the equipment room floor so that the air cushion is above the heat generator.

Support Rack - The supporting device is usually a pre-assembled stand or rack furnished with tank saddles. It allows for an installation height of seven to ten feet. In many cases, the support stand is constructed on the job-site, by assembling and field welding I-beam or angle iron material. In either case, the designer must consider the material cost factor, as well as the labor factor when evaluating the total installed cost in modern life-cycle costing procedures.

Floor Mounting Pad - The total operating weight of the hydro-pneumatic system will usually require a specially reinforced mounting pad on the equipment room floor. In the example of a 6' by 10' plain steel tank, the required floor support rack will add approximately 2,500 lbs. to the hydro-pneumatic system weight loading on the floor. The total operating weight would be over seven tons, much more than can be safely supported by standard flooring. The floor must be reinforced during the time of the erection of the structural framework, or by pouring a reinforced concrete slab.

For true cost evaluation, the designer must consider these over-looked time and material factors when selecting and sizing the hydro-pneumatic compression tank.

2. Ceiling Hung - The designer may elect to support the compression tank from the equipment room ceiling, in order to maintain proper elevation of the compression tank air cushion.

Ceiling Anchor Plate - In this type of installation, the tank and the water it contains are supported by a girder framework which must be suspended from a specially reinforced anchor plate. The plate is usually installed as an integral member of the structural framework of the building. The time and material cost factors in this case will be reflected in the structural erection phase of construction, rather than in the mechanical contract. But it must, nevertheless, be considered by the designer when planning the mechanical system.

Hanger Support Assembly - The tank will be suspended by either a job-site erected assembly or a pre-assembled purchased tank support framework. In either case, time and material cost evaluations must be considered.

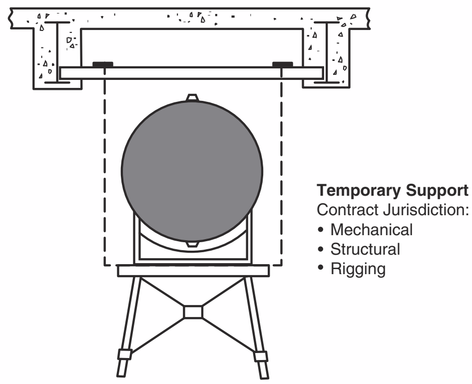
Temporary Tank Support - in addition to the above, an evaluation should be made for the provision of temporary tank supports in the installation position while the permanent ceiling hung assembly is being installed.

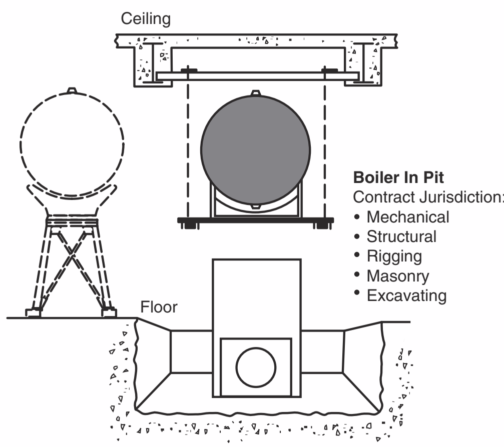
3. Boiler in Pit - in cases of limited headroom in the equipment room, the designer will have to specify an excavation to allow installation of the boiler below the tank's air cushion. Cost evaluations must include the cost of excavation, masonry work to line the boiler pit, versus the support cost factors for time and material for either floor or ceiling support of the hydro-pneumatic tank.

4. Cross-Jurisdictional Labor Cost Factors - No matter which installation the designer elects for the plain steel tank, handling it in the installation process will usually involve time and material costs outside of the traditional jurisdiction of the mechanical contractor's labor classifications. The designer must consider the cost impact of this on the total installed cost when selecting the type of hydro-pneumatic tank. For example, the 6' by 10' plain steel tank will require rigging into place. This means the cost of a crane rental, plus the time of a rigging crew for a minimum of a day, or longer if the erection schedule is delayed. This additional labor supplied by other sources are usually outside the mechanical contractor's jurisdiction.

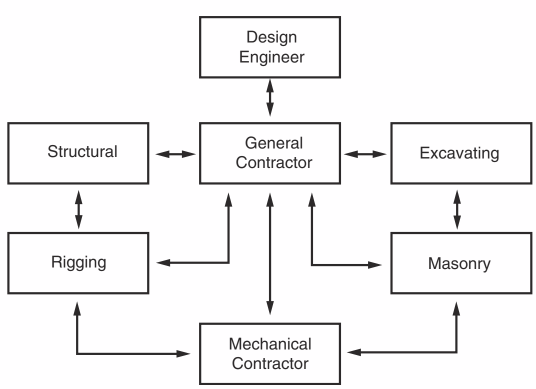
Pitting the boiler will require material and time costs of an excavating contractor, plus a masonry contractor to supply pit lining.

Much of this cross-jurisdictional labor and material must be done prior to the actual installation of the mechanical systems (i.e. boiler pitting, floor or ceiling mounting pads), and will be covered contractually in more than one of the subcontractors' contracts. While the general contractor will assume over-all responsibility for job scheduling, the designer should take into consideration the full cost impact of these often overlooked factors when initially selecting the type of hydro-pneumatic compression tank.

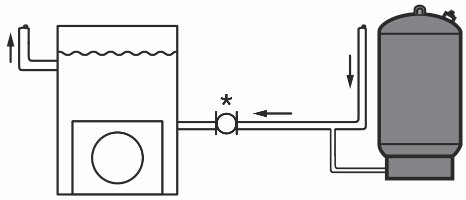




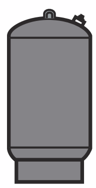
Cross - Jurisdictional Labor Increases Time And Material Costs



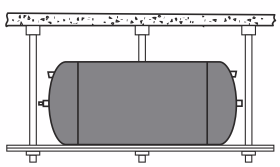
Time and installation cost factors for hydro-pneumatic tanks



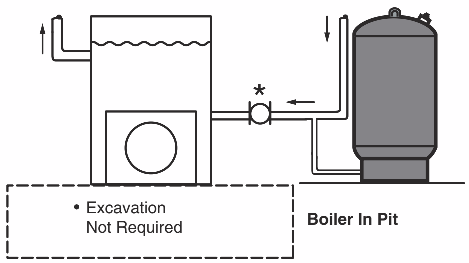
Not Required To Be
Installed Above
Waterline
Or Near Boiler



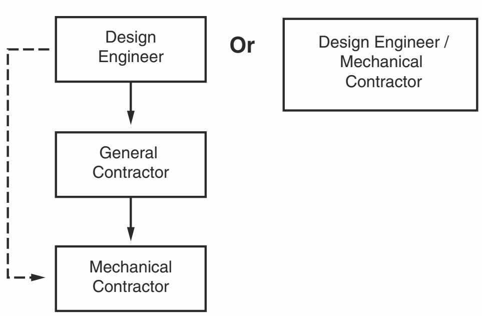
Floor Mounting
Contract Jurisdiction:
• Mechanical



Ceiling Hung
Contract Jurisdiction:
• Mechanical



Time And Material Costs Low



* Not recommended pump location. For example illustration only.

Diaphragm-type tanks

This family of hydro-pneumatic compression tanks are not required to be installed near or above the waterline of the heat generator or chiller to maintain the air cushion. Since the pneumatic cushion is permanently sealed in and away from system water, the diaphragm-type tank will never waterlog. Therefore the designer may locate it directly on the equipment room floor with its system connection made at the suction side of the circulating pump. (See Chapter One, Sections B and C, "Installations".)

1. Floor Mounting - Larger diaphragm-type tanks usually are supplied with an integral floor skirt or stand. Since it is but a fraction of the operating weight of a comparatively sized plain steel tank, it may be installed free standing. There is no need for support racks or assemblies. And usually there is no need for re-inforced floor mounting pads.

2. Ceiling Hung Models - Smaller diaphragm-type tanks are designed for suspended installation. This is easily accomplished by means of standard or stock pipe suspension hangers attached to the normal building members, with scrap lengths of piping utilized for the supporting framework. There is no need for temporary tank support, as the smaller diaphragm-type tanks are easily handled and can be slid into position once the suspension assembly is installed.

3. Boiler in Pit - Is not required for diaphragm-type tanks.

4. No Cross-Jurisdictional Cost Factors - Diaphragm-type tanks can be easily handled and installed without specialized outside rigging or welding crews. Even the largest diaphragm-type tanks can be unloaded, transported to the installation location and installed within the traditional mechanical contractor's job

Air Control VS. Air Elimination and Space/Time Factors

The other space/time factor to consider in selecting the type of hydro-pneumatic tank relates to the problem of air in the operating closed piping system.

Plain Steel Tank = Air Control

Because of the design of the plain steel tank (i.e., empty when installed), air, the pressurizing agent in the tank, is constantly in contact with system water. During operation of the system, system water circulates between the tank and the system piping. Air is constantly absorbed by the system water and is circulated in an "entrained" state throughout the system. If not provided for in the system design, this entrained air will collect in the system, causing circulation blockage and corrosion. More importantly, as more and more of the air in the tank is absorbed, more water is added to the tank to take its place. Finally, the tank becomes water-logged. The system has lost all pressurization and expansion control. It must be drained and re-charged with air. A considerable increase in time and life-cycle cost.

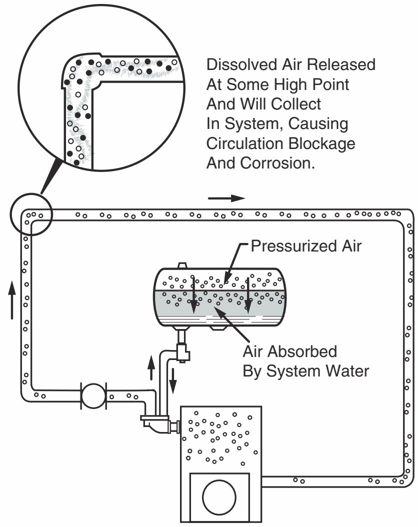
To avoid this, the user of the steel tank has to resort to extensive and complicated air control devices on the heat generator and on the tank, plus air separation devices to insure separation of entrained air from system water at a convenient point where it can be collected and directed back to the plain steel tank.

The size, weight and initial material cost of this air control equipment must be considered in the space/time evaluation of the total cost of a plain steel tank application in a large heating or cooling system. See Section C.

Diaphragm-Type Tanks = Air Elimination

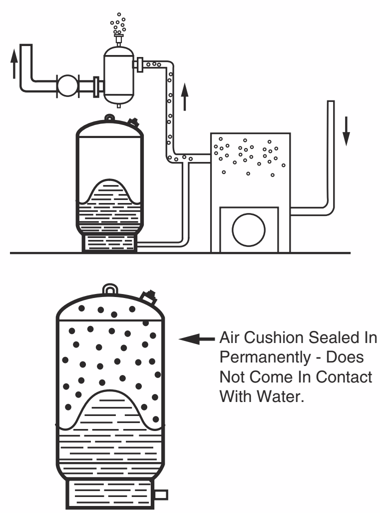
Because the air cushion in the diaphragm-type tank is sealed permanently by the diaphragm, it is never in contact with system water. It can never be absorbed and circulated through the system. It is impossible for a diaphragm-type tank to become water-logged. System pressurization and expansion control will always remain 100% efficient.

The only system air introduced is through the occasional entrance of air in system makeup water. This is determined by the amount of loss of system water through leakage. This air is not required to maintain the air cushion in the diaphragm-type tank. Therefore, it can be easily and simply separated and collected at a convenient point in the system and automatically vented to the atmosphere. There is no need for additional complicated air control devices. The space/time and cost factors are considerably less.



Air Control (Plain Steel Tank)

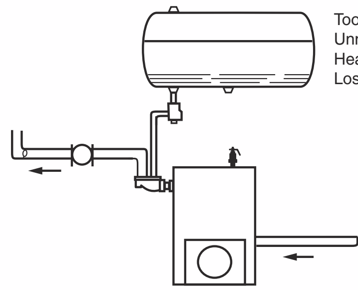
1. Complex air control system mandatory
2. Complex tank fittings required
3. Additional piping required
4. Additional construction costs



Air Elimination (Diaphragm Tank)

1. Air separation equipment
2. Air vents

Energy Factors of Hydro-Pneumatic (Expansion) Tanks



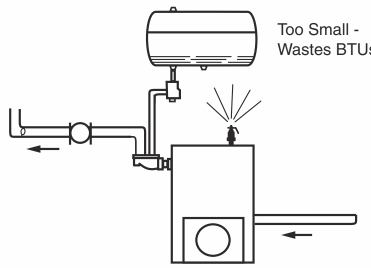
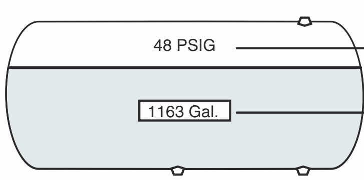


Figure A-6

Example System:

Total system water	2640 Gal.
Static head at tank	100 Ft.
Max. average design temp	200°F.
Minimum operating pressure (P_f)	48 PSIG
Maximum operating pressure (P_o)	70 PSIG



With a concern over rising fuel costs and questionable fuel availability, and the advent of pre-emptive legislation making efficient energy usage mandatory, the designer of mechanical systems is faced with severe energy-use considerations when designing a system and selecting energy-use related components. While not primarily energy consuming system components, hydro-pneumatic tanks can contribute to energy waste in a system.

1. If they are too large, the system will lose BTUs in the heating of unnecessary system water.

2. If they are too small, the system will waste BTUs through frequent activation of the relief valve.

3. If they waterlog, control of system pressure is lost and the subsequent relief valve operation will waste BTUs.

4. During normal operation, uninsulated plain steel tanks waste BTUs through heat loss to the surrounding boiler or equipment room where it is not required.

One of the prime prerequisites in accommodating energy-use considerations in selecting and sizing hydro-pneumatic tanks is critical sizing. Conservative sizing, the use of short cut sizing tables which result in grossly over-sized tanks, and short term full performance capability, are luxuries in which the designer can no longer indulge.

The hydro-pneumatic tank selection must be based on critical and highly accurate sizing procedures that will result in minimum tank size. The tank selected should add no extra water to the system. And should provide full pressurization and acceptance of expanded system water over the life of the system.

In view of these ideals, then, let us examine the energy-use factors of the two types of hydro-pneumatic tanks.

Energy-Use Factors of Plain Steel Tanks

Adds to system water content

Plain steel tanks require large amounts of initial charge water, as much as 82% of their total tank volume.

In the example system (Figure A-6), the 1520 gallon plain steel tank required 1163 gallons of initial charge water to create a minimum operating pressure of 48 psig in the tank. This initial water charge increased the total system water content of 2640 gallons by over 44%. There are now 3803 gallons of water to heat, not 2640 gallons as originally calculated for system water content. The waste of BTUs expended in heating the additional and non-useful initial charge water is one of the largest sources of wasted energy in large heating systems. (While the initial charge water in the tank will not be maintained at design temperature, it will absorb BTUs unnecessarily.)

In some cases, plain steel tank sizes could double system water content.

Constantly Decreasing Performance Efficiency Wastes Energy

By its inherent performance characteristics, (air and water mixing), the plain steel tank is prone to waterlogging in a comparatively short period of time.

As air in the tank is absorbed by system water, it is circulated to other parts of the system where it is separated and vented to the atmosphere. Additional water enters the tank to replace the space vacated by air. Eventually the tank approaches a water-logged state. Control of system pressure is lost. The relief valve then opens on every cycle of the system, losing BTUs and energy consuming components, (burners and pumps), are required to operate more frequently, wasting still more energy.

This decreasing performance begins from the first operation of the system. System pressurization begins to deteriorate almost as soon as the system is placed into operation.

To combat this, advocates of the plain steel tank insist on air control devices and systems. These systems are designed to control the rate of air absorption and collect system air and re-direct it back to the hydro-pneumatic tank. While this is a delaying action against eventual water-logging, it does not completely prevent it, or the loss of performance efficiency.

The ultimate solution is to drain the tank, refill with new charge water and start all over again. This, naturally, is a further waste of energy in BTUs required to bring this new water volume up to operating temperatures.

BTUs wasted by heat loss through tank skin surfaces

Another source of energy waste related to plain steel tanks is the loss of BTUs by heat transfer through the tank's uninsulated surface to ambient atmosphere surrounding the tank. This ambient area is usually in a boiler or equipment room where ambient temperatures are already in the high range. The additional BTUs transferred from the tank are wasted. In the case of large systems with larger plain steel tanks, this loss of BTUs can be considerable.

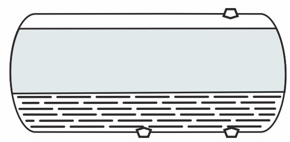
In actual temperature readings of installed and operating plain steel tanks, tank temperatures from 130°F. to 150°F. were found to be common.

If, for example, in the sample system (Figure A-7), we assume a tank temperature of 130°F., the 1520 gallon tank would waste 22,680 BTUs per hour through its tank surface. Enough energy to heat a good-sized apartment.

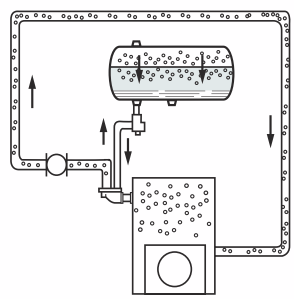
Recent procedures aimed at reducing energy waste are calling for the insulation of all bare steel components in the heating system, including plain steel hydro-pneumatic tanks.

However, while this does reduce the rate of heat loss through tank surfaces, the resulting heat gain inside the tank creates adverse affects. Increased water temperature in the tank generates a greater amount of expanded water than in uninsulated tanks. The resulting increase in the air temperature in the tank changes the pressure relationship, reducing the operating range, or allowable pressure increase of the system. (See Charles' Law Effect, Chapter 3, Section A.)

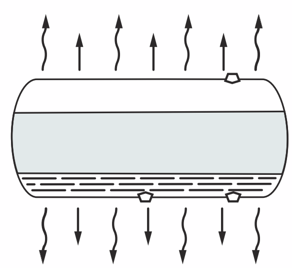
If not considered when originally sizing the tank, this would create a condition for which the tank very well might be substantially <u>under-sized.</u>



Water-logged Tanks
Will Waste BTUs



Air Control
Pneumatic cushion constantly absorbed in system water and circulated throughout system must be collected and directed back to tank to maintain pressurization.



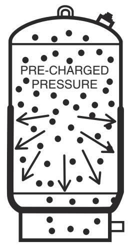
Uninsulated Tanks
Waste BTUs Through
Heat Loss

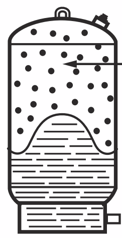
Heat Loss of Plain Steel Tanks

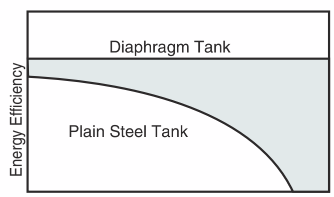
Uninsulated in 80°F. Ambient Room Temperature

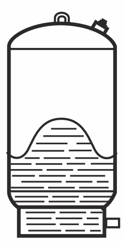
Figure A-7

Capacity Gals.	Dimensions (inches)	Heat loss in BTUs/Hour — Average Tank Temperature 120°F	130°F	140°F	150°F
150	24 x 84	4070	5220	6420	7660
270	30 x 96	6150	7880	9690	10,960
435	36 x 108	7910	10,130	12,450	14,800
650	42 x 120	10,270	13,150	16,160	19,300
1020	48 x 144	13,840	18,280	21,790	26,030
1520	54 x 168	17,710	22,680	27,810	33,280









Energy Factors of Pre-Pressurized Diaphragm-Type Hydro-Pneumatic Expansion Tanks

Adds no water to total system volume

The diaphragm-type hydro-pneumatic tank needs no initial water charge to create system pressurization. The air-side of its permanently sealed-in diaphragm is pre-charged to the minimum operating pressure (P_f). This pre-charged air cushion is permanent, completely separated from system water, and will never require additional recharging.

Because there is no initial water charging required, the diaphragm-type pre-pressurized tank adds no water to the total system volume.

As a result, no excess BTUH input is required. Only the actual system water content is heated when the system is operating.

Delivers full performance efficiency for the life of the system

Since the pre-pressurized air cushion in these types of hydro-pneumatic tank is permanent and never is lost to system water absorption, the diaphragm-type tank will always deliver full rated performance. It can never waterlog and produces a straight line curve when performance is plotted over the life-cycle of the building.

No significant energy waste through tank surface heat loss

When the design temperature is reached, the only water in the diaphragm tank is expanded water. There is comparatively little water in the tank (92.6 gallons compared to 1256 gallons in the 1520 gallon plain steel tank in Figure A-7). Since the tank is floor-mounted, there is negligible thermal circulation between tank water and system water.

Consequently there is little heat loss to the surrounding atmosphere. This would strongly indicate that diaphragm-type tanks require no insulation at all.

Section A

Hydro-pneumatics in hot water and chilled water systems

Chapter Three

BOYLE'S LAW AND THE HYDRO-PNEUMATIC (EXPANSION) TANK

Before discussing the sizing procedures of hydro-pneumatic tanks, it would be advisable to evaluate the differing principles of the plain steel and diaphragm-type tanks in terms of Boyle's Law of Perfect Gases in order to demonstrate the reasons for the space/time ratio indicated in the previous chapter.

BOYLE'S LAW...THE BASIS FOR TANK SIZING

The function of the hydro-pneumatic tank is explained by Boyle's Law of Perfect Gases:

$$PV=P_1V_1=P_2V_2$$

Which states that the product of the pressure (P) and the volume (V) of a perfect gas (air) will remain constant so long as the temperature is constant (isothermal relationship).

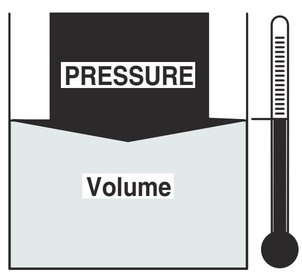
If we assume that the air in a hydro-pneumatic tank remains at constant temperature, then we can assume that the pressure and volume of the air will vary inversely if either is changed (i.e., if the pressure increases, the volume decreases).

$$P_aV_a=P_fV_f=P_oV_o$$

The pressure changes in the tank (minimum operating pressure to maximum operating pressure) will then determine the air volume required in the tank at any given time once the tank is pressurized to its maximum operating pressure.

In the following analysis of Boyle's Law, we will assume the following sample system design values:

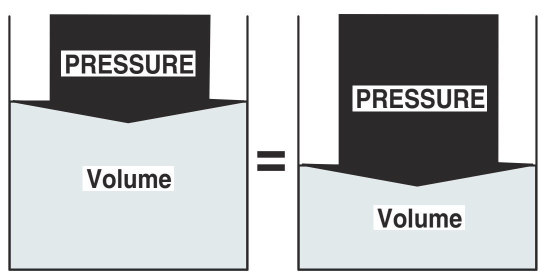
Total system water . . . .**2640 gal.**
Static head at tank . . . .**100 ft.**
Maximum average design temp . . . .**200°F**
Minimum operating pressure (P_f) . . . .**48 psig**
Minimum tank vol. required:
    Plain steel tank . . . .**1520 gal.**
    Diaphragm-type tank . . . .**356.4 gal.**
Maximum operating pressure (P_o) . . . .**70 psig**



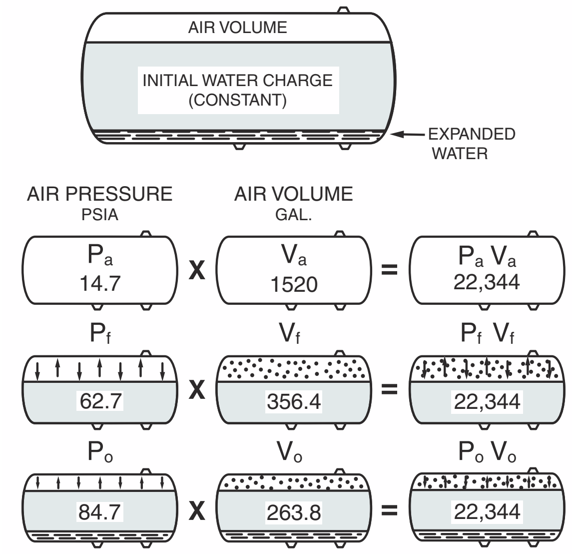
$PV=P_1V_1=P_2V_2$

Isothermal Relationship

The product of Pressure and Volume will remain constant so long as the Temperature is constant.



Boyle's Law Applied to Plain Steel Tanks



Example System:

Total system water . . . . .2640 gal.

Static head at tank . . . . .100 ft.

Max. average design temp . . . . .200°F

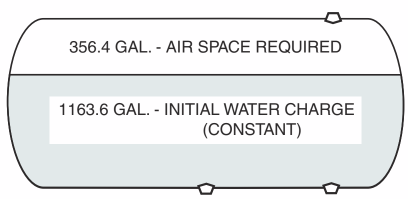
Minimum operating pressure (P_f) . . . .48 psig

Maximum operating pressure (P_o) . . . . .70 psig

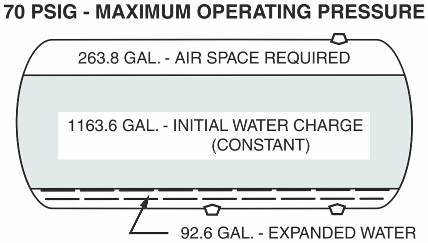
1520 Gal. — Total Tank Volume

$$P_aV_a = P_fV_f$$
$$14.7 \times 1520 = 62.7 \times V_f$$
$$V_f = 356.4$$
$$1520 - 356.4 = 1163.6$$

48 PSIG - MINIMUM OPERATING PRESSURE



70 PSIG - MAXIMUM OPERATING PRESSURE



When considering plain steel tanks, we must remember that the tank is installed empty. Theoretically, then, we can restate Boyle's Law as follows in regard to determining the air volume required in a steel tank:

$$P_aV_a=P_fV_f$$
$$P_aV_a=P_oV_o$$

Where: Pa=The pressure in the tank when it is installed. (In plain steel tanks, this is usually atmospheric pressure or 14.7 psia.)

V_a=The total volume of tank when empty.

P_f=The pressure on the air in the tank when the system is filled and the tank is at its minimum operating pressure.

V_f=The volume of air space in the tank required to contain the air cushion at the minimum operating pressure.

P_o=The pressure on the air cushion when the system is at its maximum operating pressure.

V_o=The volume of air space in the tank required to contain the air cushion at the maximum operating pressure.

We know at this point the pressure values for P_a, P_f and P_o. We will express all pressures in psi absolute (psia). Pa is atmospheric pressure, the pressure of the air in the tank when it is empty. Since the system height is 100' we will arrive at a static pressure of 43.3 psig, plus adequate positive pressure of 4.7 psig at the top of the system, which will give us a minimum operating pressure in the tank of 48 psig. We will assume a final maximum operating pressure of 70 psig.

Therefore:

P_a=14.7 psia

P_f=48 + 14.7 = 62.7 psia

P_o=70 + 14.7 = 84.7 psia

Given a total tank volume of 1520 gallons, we can now compute the portion of that tank required to contain the air cushion when the system is pressurized to 48 psig:

$$P_aV_a=P_fV_f$$
$$14.7 \times 1520 = 62.7 \times V_f$$
$$V_f = 356.4 \text{ gallons}$$

In the plain steel tank, then, a volume of 1163.6 gallons of system water is required to enter the tank, trap and compress the air to the minimum operating pressure of 48 psig. This initial volume of water in the steel tank is considered the initial charge water. It enters the tank when the system is filling and must always remain in the tank in order to maintain optimum system pressurization.

By using the formula $P_aV_a=P_oV_o$, we can determine the amount of air space required to contain the air cushion when the tank is at the maximum operating pressure of 70 psig.

This is computed as follows:

$$P_aV_a = P_oV_o$$

$$14.7 \times 1520 = 84.7 \times V_o$$

$$V_o = 263.8 \text{ gallons}$$

Therefore, in a 1520 gallon plain steel tank at a maximum operating pressure of 70 psig, there will be an air volume of 263.8 gallons.

Expanded Water

To review at this point, by using Boyle's Law, we have determined that in a 1520 gallon plain steel tank:

1. A volume of 1163.6 gallons of system water is required to pressurize the air trapped during the filling of the system and compress it to the tank's minimum operating pressure of 48 psig (P_f).

2. A 356.4 gallon volume of air space is required in the tank to contain the air cushion at the minimum operating pressure of 48 psig (P_f).

3. A 263.8 gallon volume of air space is required in the tank to contain the air cushion when it is at the maximum operating pressure of 70 psig.

We know that according to Boyle's Law, the initial water charge of 1163.6 gallons will always remain in the tank.

So if we subtract the air volume at maximum operating pressure (263.8 gallons) from the air volume of the air cushion at minimum pressure (356.4 gallons), we can determine that there will be an additional volume of space (92.6 gallons) in the tank when it is at final or maximum operating pressure and the system temperature is raised to 200°F.

This additional volume is the tank's acceptance volume, and it will accept the expanded system water that is a physical result of raising the temperature of the system from its cold position to its maximum average design temperature of 200°F.

To compute the amount of expanded water that is generated when a system's temperature increases, we can use the standard density and volume tables relating to water. Assuming that unity is established at 4°C (39.2°F), we can accurately determine the amount of expansion of the water volume for any degree of temperature above 39.2°F.

However, we know that the metal components of the system will also expand as the system temperature increases. This means that the piping, boiler (chiller, in cooling applications), and expansion tank will expand, creating more space to contain the expanding water.

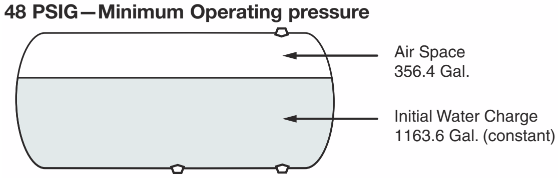
To compute this expansion, we must use a factor for the expansion of metal. This can be shown by the formula:

$$3(6.8 \times 10^{-6})\ t$$

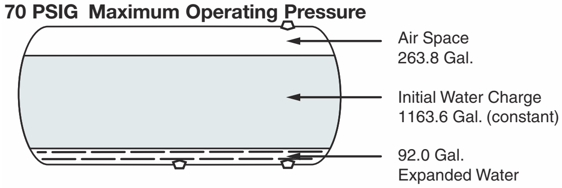
Where "3" is a volumetric conversion constant; "6.8×10^{-6}" is an acceptable coefficient for the linear expansion of metallic system components; and "t" represents the temperature differential, in degrees fahrenheit, between the fill and operating range. In our sample system, with a fill temperature of 40°F and a design temperature of 200°F, this differential would be 160°F.

1,520 Gal.—Total Tank Volume

48 PSIG—Minimum Operating pressure



70 PSIG Maximum Operating Pressure



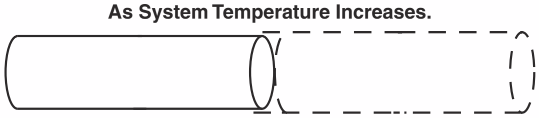
356.4 Gal. (V_f)
- 263.8 Gal. (V_o)
92.6 Gal. (Expanded Water)

Relative Density and Volume of Water

The mass of one cubic centimeter of water at 4°C is taken as unity. The values given are numerically equal to the absolute density in grams per milliliter.

Temp. °F	Temp. °C	Density	Volume	Temp. °F	Temp. °C	Density	Volume
	-10	0.99815	1.00186	95.0	+35	0.99406	1.00598
	-9	843	157	96.8	36	371	633
	-8	869	131	98.6	37	336	669
	-7	892	108	100.4	38	299	706
	-6	912	088	102.2	39	262	743
	-5	0.99930	1.00070	104.0	40	0.99224	1.00782
	-4	945	055	105.8	41	186	821
	-3	958	042	107.6	42	147	861
	-2	970	032	109.4	43	107	901
	-1	979	021	111.2	44	066	943
	0	0.99987	1.00013	113.0	45	0.99025	1.00985
	1	993	007	114.8	46	0.98982	1.01028
	2	997	003	116.6	47	940	072
	3	999	001	118.4	48	896	116
39.2	4	1.00000	1.00000	120.2	49	852	162
41.0	5	0.99999	1.00001	122.0	50	0.98807	1.01207
42.8	6	997	003	123.8	51	762	254
44.6	7	993	007	125.6	52	715	304
46.4	8	988	012	127.4	53	669	349
48.2	9	981	019	129.2	54	621	398
50.0	10	0.99973	1.00027	131.0	55	0.98573	1.01448
51.8	11	963	037	140.0	60	324	705
53.6	12	952	048	149.0	65	059	979
55.4	13	940	060	158.0	70	0.97781	1.02270
57.2	14	927	073	167.0	75	489	576
59.0	15	0.99913	1.00087	176.0	80	0.97183	1.02899
60.8	16	897	103	185.0	85	0.96865	1.03237
62.6	17	880	120	194.0	90	200°F. = .03836	
64.4	18	862	138	203.0	95		
66.2	19	843	157	212.0	100	0.95838	1.04343
68.0	20	0.99823	1.00177	230.0	110	0.9510	1.0515
69.8	21	802	198	248.0	120	0.9434	1.0601
71.6	22	780	221	266.0	130	0.9352	1.0693
73.4	23	756	244	284.0	140	0.9264	1.0794
75.2	24	732	268	302.0	150	0.9173	1.0902
77.0	25	0.99707	1.00294	320.0	160	0.9075	1.1019
78.8	26	681	320	338.0	170	0.8973	1.1145
80.6	27	654	347	356.	180	0.8866	1.1279
82.4	28	626	375	374.0	190	0.8750	1.1429
84.2	29	597	405	392.0	200	0.8628	1.1590
86.0	30	0.99567	1.00435	410.0	210	0.850	1.177
87.8	31	537	466	428.0	220	0.837	1.195
89.6	32	505	497	446.0	230	0.823	1.215
91.4	33	473	530	464.0	240	0.809	1.236
93.2	34	440	563	482.0	250	0.794	1.259

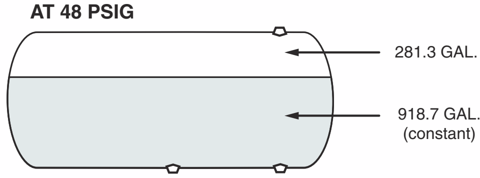
Metal Components Of System Expand
As System Temperature Increases.



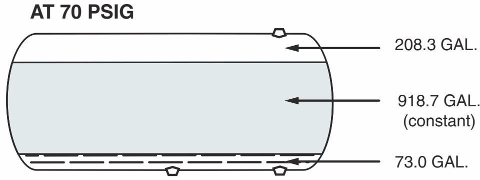
Metal Component Expansion Creates More Space
To Contain Expanding Water In System.

1,200 GAL. - Total Tank Volume

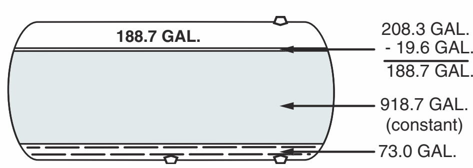
AT 48 PSIG



AT 70 PSIG



92.6 Gal. – Acceptance vol. of system (1520 gal. tank)
-73.0 Gal.– Acceptance vol. of 1200 gal. tank
19.6 Gal. – Difference in acceptance volumes



The factor for determining the expansion of metal components is:

$$3(6.8 \times 10^{-6})\ 160 = .00326$$

if we subtract this factor from the factor for computing the *gross* water expansion, we will arrive at a factor that can be used for determining the net water expansion in a system with a fill temperature of 40°F and a design temperature of 200°F.:

$$.03836 - .00326 = .03510$$

The *net*, or effective water expansion for this system with a total water content of 2640 gallons, will be:

$$2640 \times .03510 = 92.6 \text{ gallons}$$

The ASHRAE Formula, "(.00041t - .0466) V_s", accomplishes the same calculation with a universal constant factor. This formula is based on an average *net* expansion factor for a temperature range between 160° and 280° only, assuming that the fill temperature will always be 40°F.

It may not be used for accurate computations for **(a)**, fill temperatures at other than 40°F.; and **(b)**, operating temperatures above 160°F and below 280°F.

In this example of a 2,640 gallon system volume, raised from 40°F to 200°F, both methods of calculating water expansion will arrive at approximately the same result, namely, 92.6 gallons of expanded water. (The ASHRAE Formula will be in error by +0.9%.)

Therefore, the 1,520 gallon plain steel tank, in this system, will both pressurize the system and accommodate the amount of expanded water generated when it is at the design or operating temperature. This is the minimum tank volume required. We could not use a smaller tank.

For example:

If we used a 1,200 gallon tank instead of the 1,520 gallon tank:

V_f = 281.3 gallons of air space in the tank when the system is at a minimum operating pressure of 48 psig.

V_o = 208.3 gallons of air space in the tank when it is pressurized to maximum operating pressure of 70 psig.

$V_f – V_o$ = 73.0 gallons, the acceptance volume for expanded water.

We know that the system will generate 92.6 gallons of expanded water. Therefore, with a 1,200 gallon tank there will be 19.6 gallons of expanded water left over when the acceptance volume of 73.0 gallons is reached. This 19.6 gallons of expanded water will use 19.6 gallons of the air space of 208.3 gallons when the air cushion is pressurized to 70 psig. This means that the actual air volume of the tank, at its maximum operating pressure and when the system is at 200°F, will only be 188.7 gallons. Using Boyle's Law:

$$P_aV_a = P_oV_o$$
$$14.7 \times 1200 = P_o \times 188.7$$
$$P_o = 93.5 \text{ psia}$$
$$93.5 - 14.7 = 78.8 \text{ psig}$$

We find that the final pressure is not 70 psig, but 78.8 psig. This is still below the safety valve setting of 100 psig but substantially above the designed maximum operating pressure of 70 psig. In the example (Figure A-4, Chapter One), the additive pump pressure effect of 20 psig at the boiler would boost final pressure to 98.8 psig. This would be extremely close to activation of the relief valve. An override in system temperature would increase the amount of expanded water. The final pressure would also increase as the air volume in the tank decreases through water-logging until the safety margin was passed.

For Example:

An increase of 20°F over the maximum average design operating temperature of 200°F. would increase the amount of expanded water by 22 gallons. This would decrease the final volume (V_o) of air space to 166.7 gallons. And the final pressure in the tank would then be 91.1 psig.

In actual sizing procedures for plain steel type hydro-pneumatic tanks, the nearest standard-size tank above the minimum tank volume computed is chosen.

By going up to a standard size, not down, the designer is ensuring that the system will be properly pressurized and protected.

However, going beyond this and choosing a tank too large will needlessly add to cost in terms of space and time requirements.

Relief valve setting	**100 psig**
Minus pump additive pressure	**-20 psig**
	80 psig
Minus 10% safety margin	**-10 psig**
(of relief valve setting)	
Maximum allowable pressure	**70 psig**

At 200°F

1,520 gallons tank:

P_o – Maximum Operating Pressure = 70 psig
…10 psig Safety Margin

1,200 gallons tank:

P_o – Maximum Operating Pressure = 78.8 psig
…1.2 psig Safety Margin

At 220°F

1,520 gallons tank:

P_o – Maximum Operating Pressure = 77.7 psig
…2.3 psig Safety Margin

1,200 gallons tank:

P_o – Maximum operating pressure = 91.1 psig
…11.1 psig Over Allowable Pressure

Boyle's Law Applied to Diaphragm-Type Tanks

The concept of the EXTROL® diaphragm hydro-pneumatic tank incorporates a sealed-in air cushion which is pre-pressurized to the minimum operating pressure. No initial charge water is required.

In the diaphragm-type tank, the only water to enter the tank will be expanded water as the temperature rises.

In Boyle's Law as stated above for plain steel tanks, P_a will have the same value as P_f when stated for diaphragm-type tanks. Likewise, V_a will have the same value as V_f. In other words, when the diaphragm-type tank is empty, it is already at the minimum operating pressure. And the total tank volume is used to contain the air at this pressure.

Therefore, Boyle's Law, applied to diaphragm-type tanks, can be stated:

$$P_f V_f = P_o V_o$$

Given a diaphragm-type tank with a total volume of 356.4 gallons that is pre-charged to the minimum operating pressure of 48 psig, we can compute the volume required to hold the air at the maximum operating pressure of 70 psig.

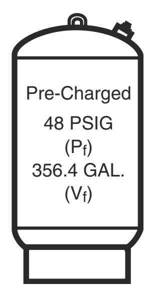
$$62.7 \times 356.4 = 84.7 \times V_o$$
$$V_o = 263.8 \text{ gallons}$$

Subtracting:

$$V_f - V_o = \text{Acceptance volume}$$
$$356.4 - 263.8 = 92.6 \text{ gallons}$$

The 356.4 gallon diaphragm-type tank will accept the 92.6 gallons of expanded water and still pressurize the system. *It will do the same job that requires a 1,520 gallon plain steel tank.*

356.4 Gal. – Total Tank Volume



356.4 Gal. – Total Volume, V_f
–263.8 Gal. – Air Volume, V_o
92.6 Gal. – Acceptance Volume

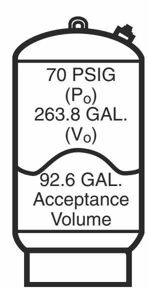
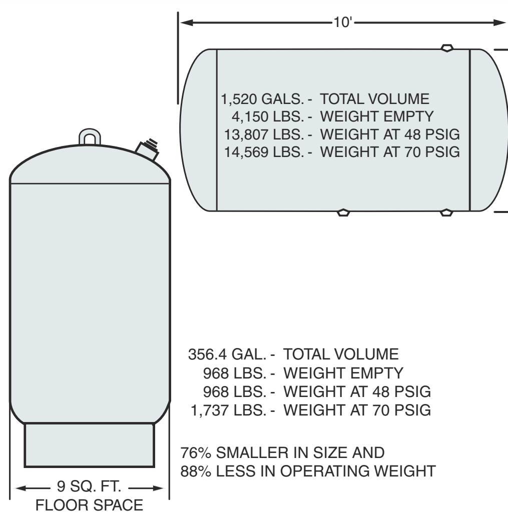


Figure A-6 (Reference page A2-3)

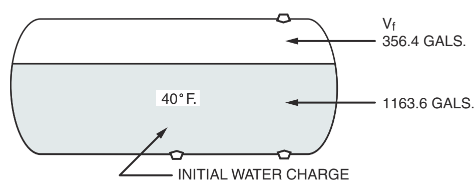
Example System:

Total system water	**2,640 gal.**
Static head at tank	**100 ft.**
Max. average design temp	**200°F**
Minimum operating pressure (P_f)	**48 psig**
Maximum operating pressure (P_o)	**70 psig**

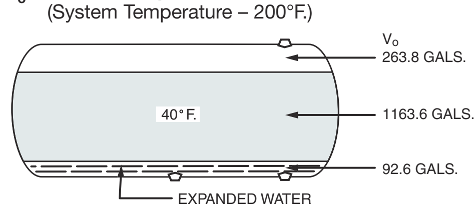


1,520 Gal. – Total Tank Volume

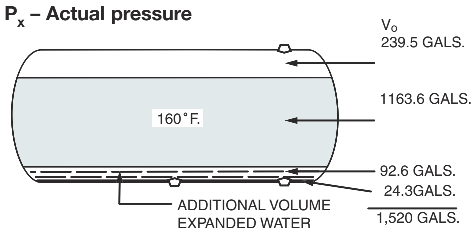
At P_f – Minimum Operating Pressure



At P_o – Theoretical pressure
(System Temperature – 200°F.)



At P_x – Actual pressure



Advantages of the diaphragm-type hydro-pneumatic tank

By using Boyle's Law to evaluate the plain steel and diaphragm tanks, we can compare the characteristics of each as they are applied to the sample system (Figure A-6):

	Plain steel	**Diaphragm**
P_a	14.7 psi	–
V_a	1520 gallons	356.4 gallons
Initial water volume	1163.6 gallons	–
P_f	48 psig	48 psig
V_f	356.4 gallons	356.4 gallons
P_o	70 psig	70 psig
V_o	263.8 gallons	263.8 gallons
Acceptance volume	92.6 gallons	92.6 gallons
Weight of tank empty	4,150 lbs.	968 lbs.
Weight at 48 psig	13,807 lbs.	968 lbs.
Weight at 70 psig	14,569 lbs.	1,737 lbs.

It can be readily seen that by eliminating the need for large amounts of system water to initially charge the tank to the minimum system pressure, the diaphragm-type tank is but a fraction of the size and weight of an equivalent plain steel tank. In our example, the diaphragm-type tank is **23%** of the size of the steel tank…and **12%** of its operating weight.

Increased tank water and air temperatures

While the water volume in the plain steel tank, (initial water charge plus expanded water), will probably never approach design temperature levels, (except in insulated tanks), it will be considerably higher than the system's original fill temperature. This increase in the temperature of the water in the tank has a twofold effect on pressure and volume changes in the plain steel tank:

1. Increased temperature of tank water generates additional expansion

There is an additional amount of expanded water that results when the temperature of the charge water in the tank increases.

This has not been calculated so far.

Assume that the temperature of the water in the tank increases from 40°F to 160°F when the system reaches its operating temperature of 200°F.

Then the 1163.6 gallons of initial charge water, which is in addition to the 2,640 gallons system volume, will generate an additional volume of 24.3 gallons net expanded water.

Then:

$$P_oV_o = P_xV_x$$

Where:

- P_o = The theoretical pressure in the tank when it is at its operating range.
- V_o = The theoretical volume of air space in the tank when it is at its maximum operating range.
- P_x = The actual pressure in the tank.
- V_x = The actual volume.

Since the 24.3 gallons of additional expanded water, generated by the increase in temperature of the charge water in the tank, must be accommodated in the tank, the volume of air space in the tank (V_o) will be 24.3 gallons less, or $V_x = V_o - 24.3$.

Then:

$$P_oV_o = P_xV_x$$

$$84.7 \times 263.8 = P_x \times (263.8 - 24.3)$$

$$P_x = 93.3 - 14.7$$

$$P_x = 78.6 \text{ psig}$$

Therefore, the effect of increased temperature of the initial charge water in the tank, while not reaching design temperature, has, however, generated enough additional expanded water to raise the actual final tank pressure well above the maximum allowable pressure of 70 psig.

2. Increased Tank Air Temperatures Raise Pressure

So far in our evaluation of Boyle's Law of Perfect Gases, we have assumed that in all cases the temperature of the air in the hydro-pneumatic tank is constant.

However, in reality, there is an increase of the air temperature in a plain steel tank as a result of heat transfer from hot water in the tank to the air.

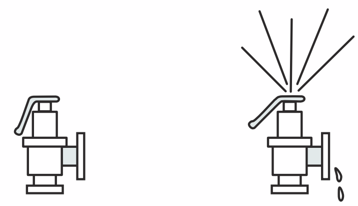
In a plain steel tank, water is constantly in contact with tank air. There is also a tendency for system water to constantly circulate in the tank through gravity flow. Because of the large amount of water in the tank (initial water charge plus expanded water), the resulting heat transfer to tank air is considerable. When temperatures vary, we must use a combination of Boyle's and Charles' Laws:

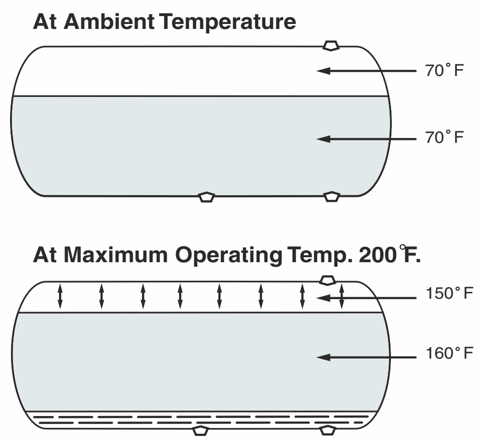
$$\frac{PV}{T} = \frac{P_1V_1}{T_1}$$

We will assume that in the 1,520 gallon tank in our sample system, the ambient temperature of the air in the tank was 70°F, and that this temperature was increased to 150°F when the tank reached its maximum operating pressure of 70 psig and operating temperature of 200°F. With a combination of Boyle's and Charles' Laws we can determine the resulting pressure change:

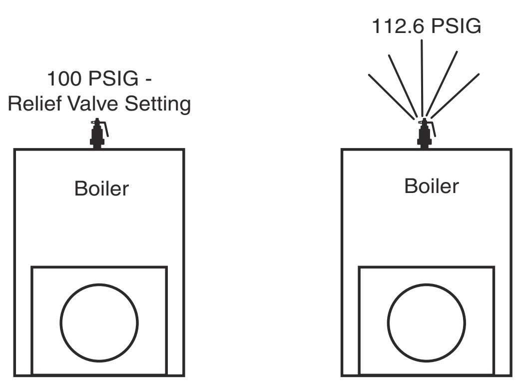
$$\frac{P_xV_x}{T} = \frac{P_{x_1}V_{x_1}}{T_1}$$

Relief Valve Setting	**100 psig**
Minus Pump Additive Pressure	**-20 psig**
	80 psig
Minus 10% Safety Margin (of relief valve setting)	**-10 psig**
Maximum Allowable Pressure	**70 psig**





Increased Tank air Temperature Raises Pressure



92.6 PSIG - Max. Operating Pressure
+ 20.0 PSIG - Pump Additive Pressure
112.6 PSIG - Actual Final Pressure at Boiler

Where:

P_x = The actual pressure in the tank at design maximum operating pressure expressed in psi absolute.

P_{x_1} = The actual pressure in the tank resulting from an increase in air temperature in the tank.

V_x = The final actual air volume when the system is at maximum operating pressure. (Since there is no further pressure increase on the air volume from system water, this value will remain constant and can be eliminated from the equation.)

T = The ambient temperature of the air in the tank (70°F) expressed in degrees absolute. (°F + 460).

T_1 = The temperature of the air in the tank as the result of heat transfer from system water, expressed in degrees absolute.

Computing the equation, we can determine the pressure change resulting from the temperature change:

$$\frac{P_x}{T} = \frac{P_{x_1}}{T_1}$$

$$\frac{93.3}{(70 + 460)} = \frac{P_{x_1}}{(150 + 460)}$$

$$.1760 = \frac{P_{x_1}}{610}$$

$$P_{x_1} = 107.3 - 14.7$$

$$P_{x_1} = 92.6 \text{ psig}$$

So, because of the effect of increased temperatures in the charge water of the plain steel tank (additional water expansion), and increased air temperatures in the tank, we find that instead of a final maximum operating pressure of 70 psig, we have an actual final pressure of 92.6 psig, well above the maximum allowable pressure of 70 psig.

In addition to this increased pressure, resulting from increased air temperature, other factors such as vapor pressure will also contribute to the pressure increases not originally calculated. In all likelihood, the actual final pressure (P_{x_1}) would be above the relief valve setting.

The answer is to select a larger tank or increase the maximum operating pressure. Equipping the tank with a special device to restrict gravity circulation of hot system water in the tank will help prevent but not eliminate the problem.

No Significant Temperature Rise in Diaphragm-Type Tanks

In the floor-mounted diaphragm-type hydro-pneumatic tank, air temperatures are not subject to significant increases. No system water enters the tank until expanded water is generated. When its maximum pressure is reached, only a comparatively small amount of water is in the tank.

For example, the only water in the diaphragm-type tank is expanded water. This water has already been calculated for in the tank sizing. There is no further amount of expanded water. So there is no need for additional acceptance volume in the tank.

And, if we assume that in the system above, tank air in a diaphragm-type tank increases in temperature from the ambient 70°F to 85°F, then:

$$\frac{P_o}{T} = \frac{P_{o_1}}{T_1}$$

$$\frac{84.7}{(70 + 460)} = \frac{P_{o_1}}{(85 + 460)}$$

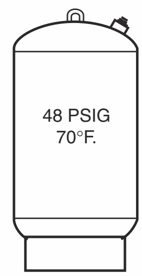
$$P_{o_1} = 87.1 - 14.7$$

$$P_{o_1} = 72.4 \text{ psig (well within 10\% safety margin)}$$

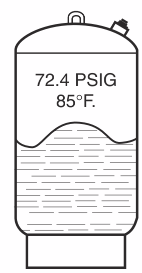
As there is little or no increase in air temperature, there is little change in the final pressure.

Consequently, the diaphragm-type tank will not require larger sizing or extra tank fittings.

At Ambient Temperature



At Maximum OperatingTemp. 200°F.



70 PSIG - Max. Allowable Pressure
72.4 PSIG - Final Pressure (within 10% safety margin)

Section B

Design factors, application and sizing procedures for EXTROL hydro-pneumatic compression tanks in low-temperature hot water heating systems

Chapter One

SYSTEM DESIGN PRESSURE FACTORS AFFECTING CRITICAL COMPRESSION TANK SIZING

- **Pump operating pressure effects on system pressurization**
- **Relative location of compression tank, circulating pump and other components**
- **Determination of design pressure values at each component location**

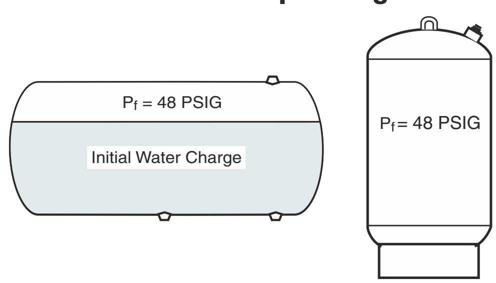
Before the accurate sizing of the compression tank can be accomplished, certain system design criteria must be determined by the designer.

Minimum to Maximum Operating Pressure at Tank Location Determines Tank Size

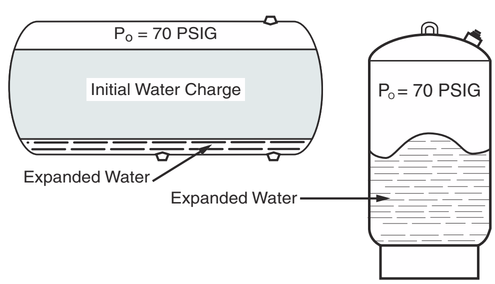
Sizing of the tank will be based on the Minimum Operating Pressure and the Maximum Operating Pressure which the designer calculates to be present at the tank's location. (See Chapter One, Section A.)

Since a correctly sized hydro-pneumatic tank will maintain system pressures at all points within a fixed maximum allowable pressure (min. to max.), the designer must make a careful evaluation of all system pressure values, at all component locations, during all stages of system operation.

48 PSIG — Minimum Operating Pressure



70 PSIG — Maximum Operating Pressure



I. Pump Operating Pressure Effect

Pump Operation Results in a Change in System Pressure Effects

The first criterion to evaluate is the pressure effect of pump operation on all components in the system.

The function of the circulating pump on a closed piping system is to cause flow through the piping, components and distribution terminals so as to achieve circulation of hot water. Operation of the pump, then, circulates hot water away from the boiler, or heat generator (supply), through the piping, components and heating terminal fixtures, and finally back to the boiler (return) to be reheated.

This ability of the pump to move water through the piping is known as pump head. It can be measured in feet of head or in pounds per square inch pressure (feet ÷ 2.31).

A pump is properly selected when the pump head is sufficient to overcome the calculated amount of friction which the piping, components and heat distribution terminals present to impede the flow of water. This is known as the friction loss of the system. It is also measured in feet or in pounds per square inch.

If, for example, the friction loss of a piping system is calculated to be 20 psig, a pump with the ability to produce a 20 psig head is required to circulate the system water.

When the pump comes on, a 20 psig force is delivered at the discharge side of the pump. This pressure head is gradually dissipated by the system friction loss as water flows through the system. Therefore, at the suction side of the pump, the 20 “pound” head will be reduced to 0 psig.

This gradual reduction in the pump head is proportional to the distance from the discharge side of the pump measured in the direction of water flow. The farther away from the discharge side of the pump, the lower the added pressure effect on the system.

This pressure effect is *additive* to system pressures at all component locations throughout the system. For example, if we place a 20 psig head circulating pump on the return line of a boiler, it will be pumping directly into the boiler. When the pump operates, an additional pressure of 20 psig will be put upon the boiler pressure already existent.

If we put a system component at the top of the system, midway in the piping circuit, and assuming uniform pipe sizes, an additive pressure of approximately 10 psig will be *felt* by that component when the pump operates.

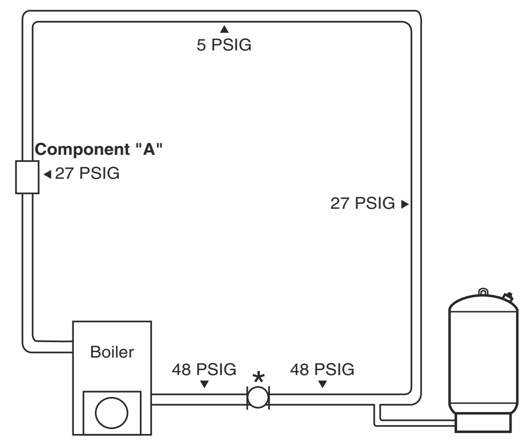
This operating pressure effect on components must be considered when establishing minimum and maximum system pressure values.

Compression Tank Connection Determines Type of Pump Operating Pressure Effect

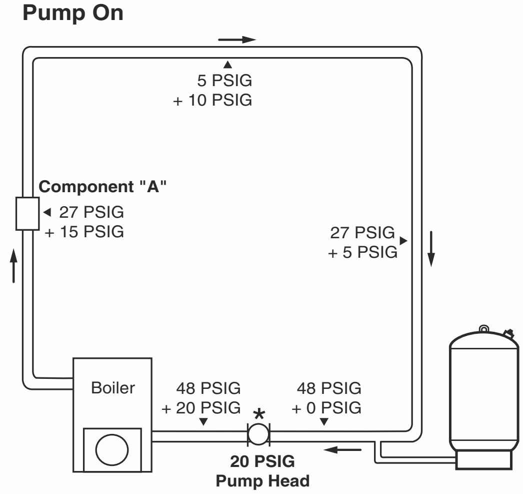
So far, we have seen that pump operation creates an additive pressure effect on system components.

However, the location of the compression tank connection to the system relative to the location of the pump can result in *subtractive* pressure effects.

Pump Off



Pump On



*Not recommended pump locaion. For example illustration only.

Compression Tank Connection Is a "Point of No Pressure Change"

System pressure at the tank connection can only be changed by increasing or decreasing water volume in the tank.

This is shown by Boyle's Law of Perfect Gases (see Chapter Three, Section A) …the pressure changes in the air cushion of a hydro-pneumatic tank are inversely proportional to volume changes. The only way to increase pressure in the air cushion of a hydro-pneumatic tank is to compress the air, decreasing its volume. And the only way to accomplish this is to add more water to the tank.

Since the operation of the circulating pump only causes water to *flow* and does not add water to the tank, there can be no decrease in air volume in the tank. Subsequently, there is no pressure change at that point in the system. It is a "point of no pressure change."

If we then place this point of compression tank connection mid-way in the system at equal distance from both the discharge side and the suction side of the pump, we will alter the pump operating pressure effect on all other system points.

The pump operating pressure effect from the discharge side of the pump to the "point of no pressure change" will be *additive*. From the tank connection to the suction side of the pump, the pump pressure effect will be *subtractive*.

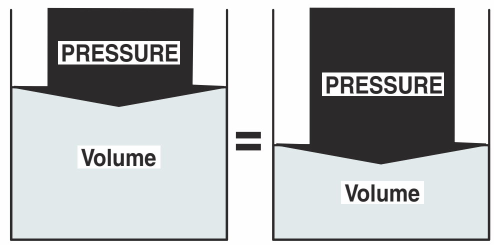
Additive Pressure Effects, then, caused by pump operation, will be felt as a pressure increase on all system components located between the discharge side of the pump and the compression tank connection. The *magnitude* of the increase will be equal to the friction loss of the system between the component and the compression tank location measured in the direction of flow. (Or the friction loss from component to compression tank connection measured in a counter-flow direction and then subtracted from the total system friction loss or pump head.)

Subtractive Pressure Effects will be felt as a pressure *decrease* at all piping points and components located between the compression tank connection and the suction side of the pump. The magnitude of the decrease will be equal to the system friction loss from the compression tank connection to the component measured in the direction of flow.

Connection of Compression Tank as Close as Possible to Suction Side of the Pump Ensures That All Operating Pressure Effects Are Additive

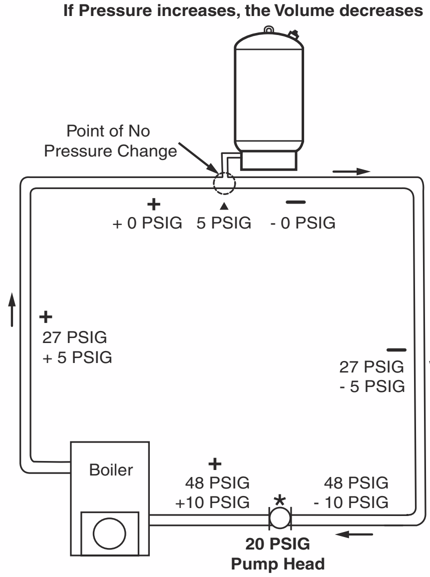
In the case of subtractive pump pressure effects, minimum system pressures at points between the tank connection and the suction side of the pump could result in pressures below the *anti-flash* point of the system, and harmful system boiling could occur.

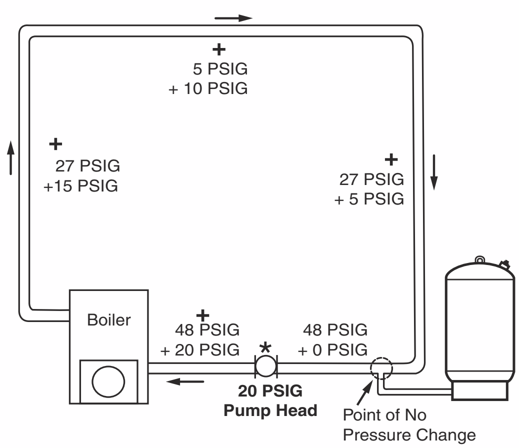
Because of this, it has been traditionally accepted engineering practice to locate the *connection of the compression tank as close to the suction side of the pump as possible*. This will ensure that all pressure effects resulting from pump operation will be *additive*, with the magnitude of pressure effect on each component equal to the friction loss of the system from that component location to the compression tank connection, measured in the direction of flow.



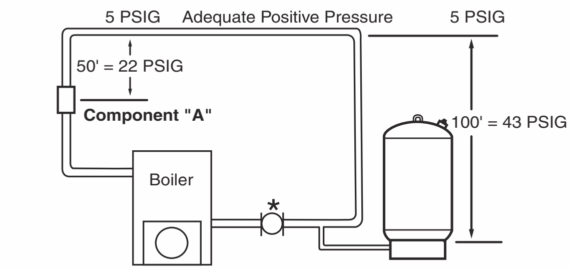
$$P_a V_a = P_f V_f = P_o V_o$$

If Pressure increases, the Volume decreases



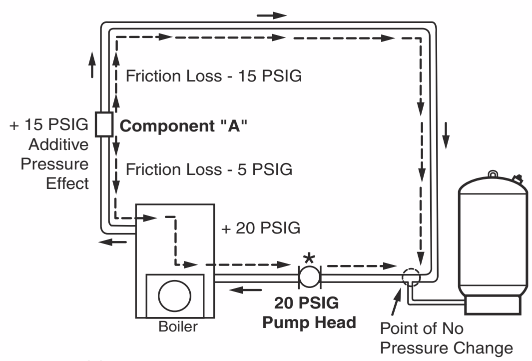


* Not recommended pump location. For example illustration only.



Example: (1)

In a system 100' high with a maximum average design temperature of 200°F, an Adequate Positive Pressure to insure venting and prevent flashing at the top of the system is 5 psig. This is applicable to each component location.



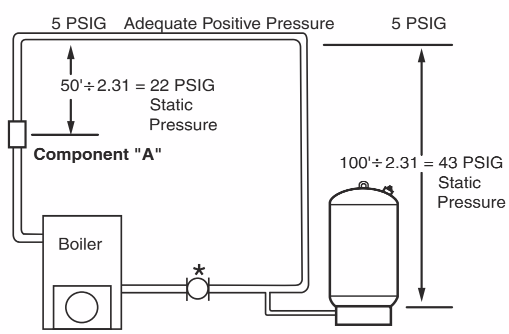
Example: (2)

In the system above, a boiler is located in the basement, as shown, and a system component is located half-way up the supply side of the system, as shown.

The friction loss for the system is calculated to be 20 psig. Additive pressure of 20 psig would be felt at the boiler when pump operates.

Assuming all pipe sizes are uniform, the friction loss from Component "A" to the compression tank in direction of flow would be 15 psig.

Or, the friction loss from Component "A" to the compression tank in a counter-flow direction would be 5 psig. This subtracted from the pump head of 20 psig would be 15 psig.



Example: (3)

In the system above, the static pressure at the boiler location would be 100' divided by by 2.31 or approximately 43 psig.

The static pressure at Component "A" would be 50' divided by 2.31 or approximately 22 psig.

**Not recommended pump location. For example illustration only.*

II. Determination of Design Pressure Factors at Each Component Location

Once the pump operating pressure effect upon each component is understood, it is possible to evaluate and measure it and other pressure factors which will determine the minimum and maximum operating pressures at the location of the compression tank.

There are five pressure values to be determined in this evaluation:

1. Adequate Positive Pressure

This is a fixed pressure value which is applicable to each component location. It is the positive pressure necessary to maintain positive venting and prevent flashing. (See Chapter one, Section A). It is determined by referring to Table 4 (Page Ref-6) in the Reference Supplement Section and reading the Adequate Positive Pressure opposite the system maximum design temperature.

2. Additive Pump Pressure Effect at Each Component Location

This is the pressure increase at each component location that results when the pump operates, as discussed above.

To determine it for each component location, calculate the friction loss between each component and the compression tank connection, measuring in the direction of system flow.

Or, if the component is closer to pump discharge, calculate the friction loss from it to the compression tank connection in a counter-flow direction, and subtract it from the pump head.

3. Static Pressure at Each Component Location

This is the pressure of the height of water (weight) above each component location.

To determine it, in gauge pressure, divide the height of the system piping in feet above each component location by 2.31.

4. Maximum Allowable Pressure Increase at Each Component Location

This is the maximum increase in pressure at each component location from minimum (static plus adequate positive) to maximum allowable pressure.

The maximum allowable pressure is that pressure beyond which the designer does not want pressures to rise. It must be at some point below the rated working pressure or relief valve setting of each component, and must also reflect the additive pump operating pressure effect at each component location. (See Chapter One, Section A)

To determine the Maximum Allowable Pressure Increase at each component location, the designer merely has to make four subtractions of known pressure values, beginning with the relief valve setting of the component:

Relief Valve Setting (or rated working pressure).

1. MINUS at least 10% of that setting.

2. MINUS the Additive Pump Operating Pressure Effect at that location.

3. MINUS the Adequate Positive Pressure for the system.

4. MINUS the Static Pressure at that location.

The result will be the Maximum Allowable Pressure Increase for the component.

5. Maximum Allowable Pressure Increase for System

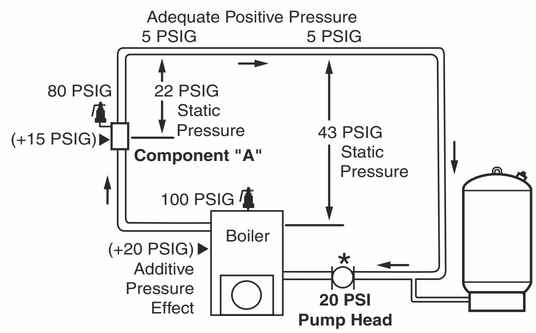
This is a fixed pressure value applicable to all component locations, including the compression tank. It is equal to the *smallest*, or at least allowable increase of all component locations.

III. Determination of Minimum and Maximum Operating Pressures at Compression Tank Location

Now that the Maximum Allowable Pressure Increase for the system has been determined, the designer can easily establish both the Minimum Operating Pressure and the Maximum Operating Pressure at the compression tank location:

1. Locating Compression Tank in the System

The designer selects the most advantageous location for the compression tank, with its connection to the system piping made as close as possible to the suction side of the pump.



Example: (4)

In the system shown, the boiler is rated at 100 psig. Component "A" is rated at 80 psig. Relief valves are installed on both. The Maximum Allowable Pressure Increase would be calculated as follows:

Boiler

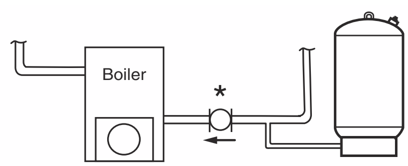
Relief Valve Setting	100 psig
1. MINUS at least 10% of relief valve setting	-10 psig
	90 psig
2. MINUS the Additive Pump Operating Pressure at boiler	-20 psig
	70 psig
3. MINUS the Adequate Positive Pressure for system	-5 psig
	65 psig
4. MINUS the Static Pressure at boiler	-43 psig
Maximum Allowable Pressure Increase at Boiler	22 psig

Component "A"

Relief Valve Setting	80 psig
1. MINUS at least 10% of relief valve setting	-8 psig
	72 psig
2. MINUS Pump Operating Pressure at Component "A"	-15 psig
	57 psig
3. MINUS the Additive Positive Pressure for System	-5 psig
	52 psig
4. MINUS the Static Pressure at Component "A"	-22 psig
Maximum Allowable Pressure Increase at Component "A"	30 psig

Example: (5)

In the system, the boiler location, with a Maximum Allowable Pressure Increase of 22 psig, will establish the Maximum Allowable Increase for the entire system, including the compression tank.

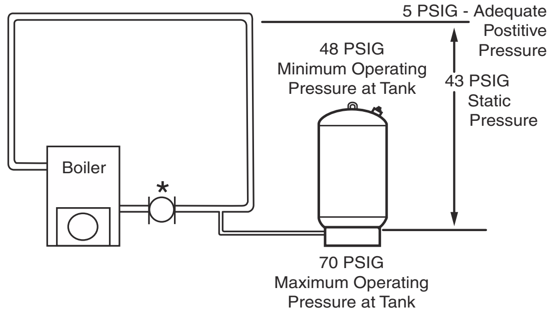


Example: (III-1)

In the system, the compression tank will be installed at the bottom of the system, adjacent to the circulating pump, connected to system piping at the suction side of the pump.

**Not recommended pump location. For example illustration only.*

Figure B-1



Example: (2)

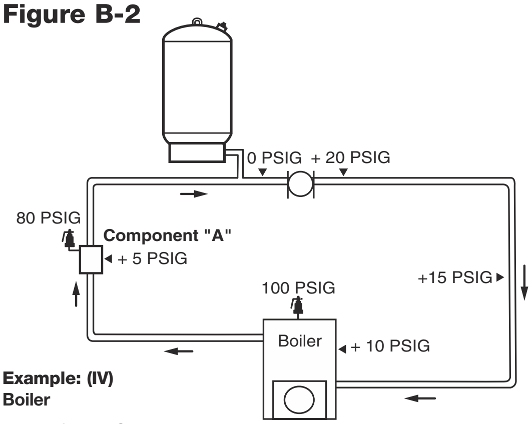
Static pressure at tank location (100' divided by 2.31) . . . . .43 psig
PLUS Adequate Positive Pressure for system . . . .+ 5 psig
Minimum Operating Pressure at Tank . . . . .48 psig

Example: (3)

Minimum Operating Pressure at Tank . . . . .48 psig
PLUS Maximum Allowable Pressure Increase . . . .+ 22 psig
Maximum Operating Pressure at Tank Location . . . . .70 psig

**Not recommended pump location. For example illustration only.*

Figure B-2



Example: (IV)
Boiler

Relief Valve Setting . . . . 100 psig
1. MINUS at least 10% of relief valve setting . . . . -10 psig
90 psig
2. MINUS Additive Pump Operating Pressure . . . . 10 psig
80 psig
3. MINUS Adequate Positive Pressure . . . . -5 psig
75 psig
4. MINUS the Static Pressure at boiler . . . . -43 psig
Maximum Allowable Pressure Increase at Boiler . . . . 32 psig

Component "A"

Relief Valve Setting . . . . 80 psig
1. MINUS at least 10% of relief valve setting . . . . -8 psig
72 psig
2. MINUS Pump Operating Pressure . . . . -5 psig
67 psig
3. MINUS Adequate Positive Pressure for system . . . . -5 psig
62 psig
4. MINUS the Static Pressure at Component "A". . . . . -22 psig
Maximum Allowable Pressure Increase at Component "A" . 40 psig
Minimum Operating Pressure at Tank Location:
Static (0) plus Adequate Positive Pressure. . . . . 5 psig
Maximum Operating Pressure at Tank Location:
Minimum (5) plus Maximum Allowable Pressure Increase of 32 psig 37 psig

2. Determine Minimum Operating Pressure at This Location

This is the Static Pressure at this location plus the Adequate Positive Pressure for the system.

3. Determine Maximum Operating Pressure at This Location

To the Minimum Operating Pressure, add the Maximum Allowable Pressure Increase for the system. This will result in the correct Maximum Operating Pressure required in the compression tank for proper sizing.

Minimum and Maximum Operating Pressure at Tank Will Determine Acceptance Factor

Once the designer has determined the Minimum to Maximum Operating Pressure range for the required compression tank, he or she can easily compute the acceptance factor, which when divided into the amount of expanded water will result in the correct total tank volume required in an EXTROL hydro-pneumatic tank.

In the case of the example shown in Figure B-1, the Acceptance Factor would be 0.260, meaning that 26% of the total tank volume required would contain the amount of expanded water when the maximum operating pressure of 70 psig has been reached.

Chapter Two in this Section contains detailed instructions for both accurately estimating total tank size and critical sizing procedures.

IV. Varying Tank and Pump Locations to Reduce Compression Tank Size

With the EXTROL diaphragm-type hydro-pneumatic tanks, the designer has great flexibility in locating the compression tank. Since the air cushion is permanently pre-pressurized, it requires no external air control connections or fittings, and can be installed anywhere in the system, providing care is taken to ensure constant additive pump pressure throughout.

For example, in Figure B-1, the EXTROL compression tank was located at the bottom of the system, adjacent to the pump and boiler.

At this location, the tank was subjected to a high static pressure as the full weight of system water was above it. In addition, the full magnitude of Additive Pump Operating Pressure was put upon the boiler as the pump was located to pump into the boiler. This reduces the Maximum Allowable Pressure Increase for the system to very narrow limits.

If we locate the tank and the circulating pump at the top of the system (Figure B-2) (with the tank connected close to the suction side of the pump), we can alter the pressure factors in our favor:

1. The Static Pressure on the tank would be eliminated.

2. The Additive Operating Pressure on the boiler would be cut in half.

Now, an evaluation of pressure factors would lead to a smaller minimum operating pressure at the tank and a wider allowable pressure increase.

The acceptance factor for a minimum to maximum operating pressure range of 5 to 37 psig is 0.619; or 61.9% of the total tank size required will accept expanded water at maximum operating pressure. A much smaller tank is required.

V. Varying Tank and Component Locations to Reduce Additive Operating Pressure Effects and Tank Size

The designer also has flexibility in locating all system components to reduce additive operating pump pressures so that system operating pressures can be kept to within desired rated working pressure limits.

In the sample system (Figure B-2, page B1-6), the boiler installed at the bottom of the system has to be rated at 100 psig because of: (1) high static pressure, and (2) high additive operating pressure caused by the pump pumping directly into the boiler. This additive operating pump pressure effect also reduced the Allowable Pressure Increase for the system to very narrow limits. Hence, a compression tank with a total volume of 356.4 gallons is required.

If we locate the boiler at the point midway in the system elevation (Figure B-3), connect the EXTROL compression tank on its return, and install the pump on its supply to pump away from the boiler, we can reduce additive operating effects and still reduce the size of the EXTROL required. (We would also locate Component A at the top of the system as shown.)

1. Static pressure at both the boiler and tank location will be cut in half.

2. Additive operating pressure on the boiler will be eliminated. We can now safely use a boiler with a rated working pressure of only 80 psig, instead of 100 psig.

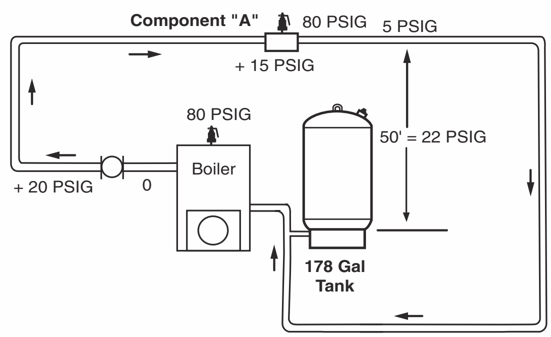
In this variation of tank and component location, not only will a lower rated boiler be acceptable, but 52% of the required EXTROL tank size can be used for expanded water, as compared to only 26% when boiler and tank were in the basement. Instead of a 356 gallon tank, the designer can use a 178-gallon tank.

Unlimited Flexibility in Design Pressure Factors

The designer has unlimited flexibility in determining design pressure factors which will affect compression tank sizing. The innovative designer may wish to fully explore all possibilities in altering: operating pump pressure effects (*additive and subtractive*): varying tank and all system component locations; using remote air tanks, compressor-actuated tanks etc.

Figure B-3 - Sample System:

V_s = 2640 gals.
T = 200°F.
P_f = 48 psig
P_o = 70 psig
Tank Volume = 356 gals.



Example: (V)
Boiler

Relief Valve Setting	80 psig
1. MINUS at least 10% of relief valve setting	-8 psig
	72 psig
2. MINUS Additive Pump Operating Pressure	0 psig
	72 psig
3. MINUS Adequate Positive Pressure	-5 psig
	67 psig
4. MINUS the Static Pressure at boiler	-22 psig
Maximum allowable pressure increase at boiler location	45 psig

At Component "A"

Relief Valve Setting	80 psig
1. MINUS at least 10% of relief valve setting	-8 psig
	72 psig
2. MINUS Additive Pump Operating Pressure	-15 psig
	57 psig
3. MINUS Adequate Positive Pressure	-5 psig
	52 psig
4. MINUS the Static Pressure at Component "A" location	-0 psig
Maximum allowable pressure increase at Component "A"	52 psig

The Maximum Allowable Pressure Increase for the System Would be 45 psig.

Compression Tank Min. – Max. Operating Range
Minimum Operating Pressure (P_f) at tank location:

Static Pressure	22 psig
PLUS Adequate Positive Pressure	+5 psig
	27 psig
PLUS Maximum Allowable Pressure Increase	+45 psig
Maximum Operating Pressure (P_o)	72 psig

Section B

Design factors, application and sizing procedures for EXTROL® hydro-pneumatic compression tanks in low-temperature hot water heating systems

Chapter Two

SIZING THE HYDRO-PNEUMATIC TANK FOR LOW-TEMPERATURE WATER SYSTEMS

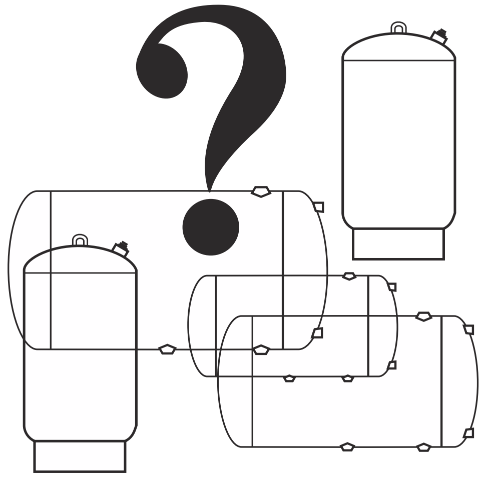
When selecting and sizing hydro-pneumatic tanks for pressurization and expansion control in engineered space heating systems involving hot water as the heat transfer source, the designer must consider the important space, time and energy factors covered in Chapter Two, Section "A."

Accurate Critical Sizing a Must

As a result of these priorities, the most important consideration in selecting and applying hydro-pneumatic tanks in large systems is one of critical sizing.

Designers of mechanical systems cannot afford the luxury of oversizing system components to achieve a safety margin, nor can they take the chance of undersizing through the use of inaccurate averaging approaches traditionally used for sizing in the past.

To meet the critical sizing requirements, the designer must be able to provide an adequate tank volume to guarantee full system pressurization at all times, plus the accommodation of the accurately calculated amount of expanded water which the system will generate. But, the designer must provide this with a tank size of minimum volume and weight so that a minimal amount of space and time will be consumed, and no waste of energy will be encountered.



Traditional Sizing Methods Do Not Allow Accurate Critical Sizing

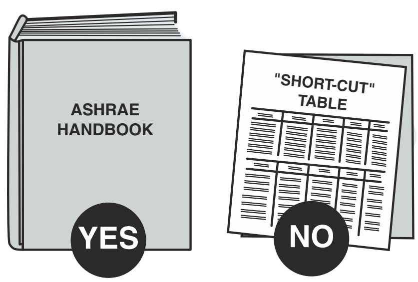
The traditional sizing methods which have been employed by system designers in the past have served well for *rule of thumb* approximations. Many times, the designer has used these methods to arrive at a general size and then added his or her own safety margin to select a tank of larger size than calculated.

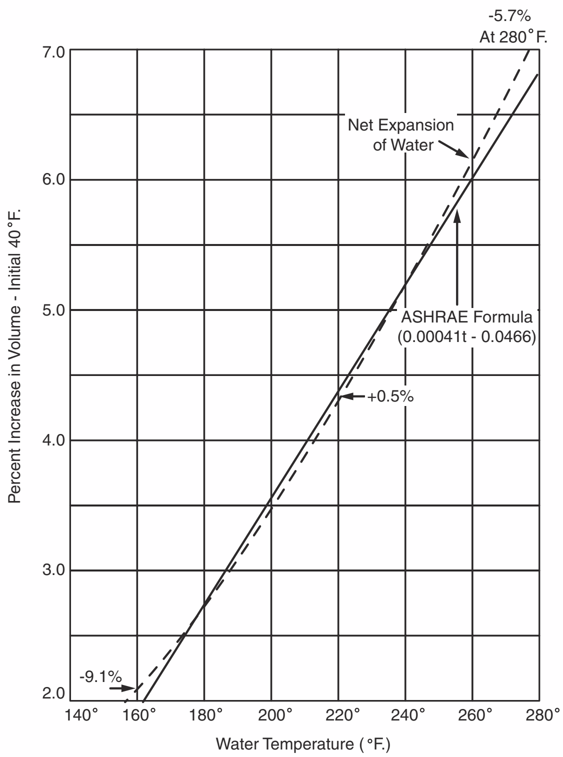
While this practice has resulted in tanks sufficiently large and therefore with a good safety margin in the past, it will not meet the critical sizing requirements that are being employed in modern system design.

There are two basic sizing methods which have been used by designers in the past. The first is the sizing formula published by ASHRAE. It is a formula for accurately estimating tank sizes for systems with operating temperatures of 160°F to 280°F.

The second basic method used has been the various versions of tables and factors published by manufacturers for *short cut* sizing of tanks.

Let us examine both of these methods in light of critical sizing requirements.





Net Factors

Expanded Water

237 gal. (ASHRAE)
-226 gal. (Actual)
11 gal. Additional

Size of Plain Steel Tank

4202 gal. (ASHRAE)
-4007 gal. (Actual)
195 gal. Oversize

Size of Diaphragm-Type Tank

826 gal. (ASHRAE)
-787 gal. (Actual)
39 gal. Oversize

A. ASHRAE Formula

Sizing plain steel hydro-pneumatic tanks

$$V_t = \frac{(.00041t - .04666)V_s}{\frac{P_a}{P_t} - \frac{P_a}{P_o}}$$

Where:
- V_t = Minimum tank volume size in gallons.
- V_s = Total water content of system in gallons.
- t = Maximum average design temperature.
- P_a = Pressure in the tank when water first enters, usually atmospheric.
- P_f = Minimum operating pressure at the tank expressed in pounds per square inch absolute (PSIA).
- P_o = Maximum operating pressure at the tank expressed in pounds per square inch absolute (PSIA).

1. Calculating expanded water with the ASHRAE formula

The top line of the ASHRAE formula is an averaging equation based on the ASHRAE curve for net expansion of water. It may be used for temperatures that fall between 160°F and 280°F. If we plot the equation as a curve and compare it with the actual curve as shown in the ASHRAE Guide, we can see that at the extreme upper and lower portions of the temperature range, the percentage of errors increases.

For example:

Expansion Factor for Net Expansion of Water*

Temperature	ASHRAE formula	Actual	% Error
160°	0.0190	0.0209	-9.1%
220°	0.0436	0.0434	+0.5%
280°	0.0682	0.0723	-5.7%

*Based on the actual gross water expansion as expressed in the "Tables for Density of Water," (page B3-2) less the expansion of metallic system components calculated by an acceptable coefficient of expansion.

2. Fill temperatures other than 40°F

The ASHRAE equation also assumes that the system is initially filled with water at 40°F. This means that systems filled with water higher in temperature than 40°F will actually generate less expanded water than calculated by the ASHRAE equation. In cases of critical sizing in systems with large system volumes, this could easily mean hundreds of gallons of tank volume not actually required.

For example:
If we assume a system with:
210 °F design temperature
6,000 gallons system volume
60 psig minimum operating pressure at tank
90 psig maximum operating pressure at tank
70 °F temperature of water at filling

then:

	ASHRAE Net Factor 0.0395	Actual Net Factor 0.0376
Expanded water	237 gal.	226 gal.
Size of plain steel tank	4202 gal.	4007 gal.
Size of diaphragm-type tank	826 gal.	787 gal.

It can be seen that, while the ASHRAE equation is very accurate for average sizing, if the designer is faced with extremely critical sizing requirements, the ASHRAE equation will not provide the accuracy required.

B. Manufacturers' Selection Tables and Short Cut Methods

Over the years, many short cut methods, nomographs and selection tables have been evolved by component manufacturers to give system designers a quick approximation of tank size. These are generally for sizing plain steel tanks.

In many instances, designers who have used these methods have developed their own margins of safety to use in conjunction with short cut methods, so that tanks would not be undersized.

In view of critical sizing requirements, however, while the ASHRAE formula does give a high degree of accuracy for average sizing, the various manufacturers' selection tables and short cut methods are, at their best, approximations only.

BTU Conversion to Gallonage Is Cause of Errors

Most manufacturers' selection tables, nomographs, etc., are based on an approximation of system water volume derived from a conversion factor to convert the total BTUH load of the system into gallons of water.

Since boiler designs and styles cover a wide range of water content, and because of the great variety of types of terminal units, it is virtually impossible to arrive at an accurate computation of total system water content with this method.

For example, referring to the actual listed water contents of major boiler manufacturers, water content for boilers with outputs in the neighborhood of 4 million BTUH run as low as 279 gallons to as high as 1,069 gallons.

Such a discrepancy could result, then, in tanks that are extremely undersized or oversized.

Selection Based on Small Residential Systems

The other reason why short cut selection tables are suspect is that most of them are based on small system characteristics (12 psig – 30 psig). For small systems with small system volumes, approximation methods such as these are adequate. The percentage of error, even if major, will not seriously affect the performance of tanks in small single family systems.

But, when using *short cut* tank sizing methods for larger engineered systems, the designer is faced with much greater system volumes, and possibly higher static pressures. To compensate for this, variation factors are added to these short cut tables as table footnotes.

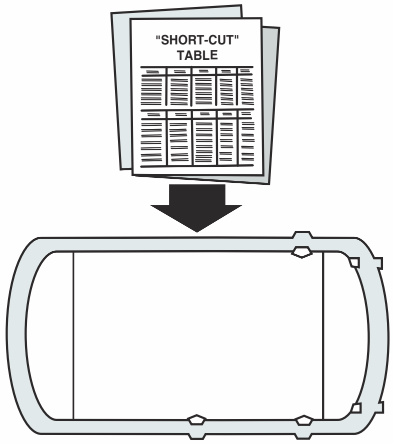
However, these factors, which are multiplying in nature, only amplify the inaccuracies already built into the basic approximation selection tables. The results become more inaccurate as the system gets bigger.

For example, using a specific *short cut* table with correcting factors (in use at publication date), here's what results show when sizing a tank for the system characteristics listed at the beginning of this chapter.

Sizing Methods

	Short cut **Table**	**ASHRAE Formula**	**Critical Sizing (Actual)**
Plain steel tank	**3,344 gal.**	**4,202 gal.**	**4,007 gal.**
Diaphragm-type tank	**Can't be sized**	**826 gal.**	**787gal.**

For these obvious examples of inaccuracy, it is strongly recommended that the designer not use short cut selection tables and short cut methods for critical sizing of hydro-pneumatic tanks.

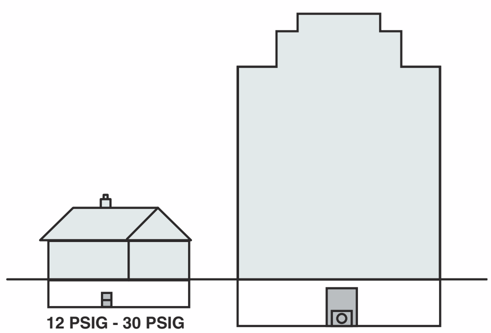


Oversized for Margin of Safety

Manufacturer's Selection Table for 4,000,000 BTUH

(BTUH=Gallons of system water)

Boiler "A"	1,069 gal.	
Boiler "B"	- 279 gal.	
	790 gal.	More than actual contents of system

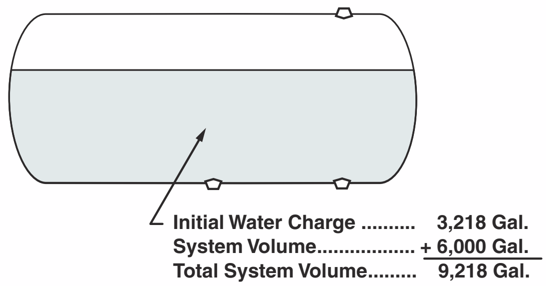


Plain Steel Tank

Critical Sizing	4,007 gal.	
Short Cut Table	-3,344 gal.	
	663 gal.	Undersize
ASHRAE Formula	4,202 gal.	
Short Cut Table	-3,344 gal.	
	858 gal.	Undersize

C. Recalculation Required on Large Plain Steel Tanks

4,007 Gal. Plain Steel Tank



Another consideration the designer should be aware of when selecting large plain steel tanks under critical sizing requirements is that the extremely large amounts of initial charge water required will affect the method of sizing the tank.

As shown in Chapter Three, Section A, large tanks which require large initial water charges substantially add to the total system water content. In the example system previously mentioned, the critically sized 4,007-gallon plain steel tank will require 3,218 gallons of initial charge water. Added to the 6,000-gallon system volume, the designer is faced with a total system volume of 9,218 gallons.

While temperatures of the tank water (3,218 gallons) will not approach design temperature (unless the tank is insulated), temperatures will be considerably higher than when the system was filled. More water expansion will be generated than originally calculated, and the tank could very well be too small. The designer should then recalculate, assuming a tank water temperature to arrive at a critical but safe size.

Example:
Assume tank water temperature to be 150°F
(Actual expansion factor for 70°F - 150°F is 0.01636.)
Then:

Total expanded water = 226 + (0.01636 x 3,218)
= 226 + 52.6
= 278.6 gallons

As a result of this recalculation of expanded water, the actual minimum tank volume required in a plain steel tank would be 4,940 gallons.

The recalculation requires a determination of the initial charge water gallonage by applying Boyle's Law:

$$PV = P_1V_1$$

Once the preliminary sizing has been accomplished and a value for the minimum tank size has been calculated, the designer can apply Boyle's Law to determine the amount of initial charge water:

$$P_aV_a = P_fV_f$$

In the example already given, the determination of 3,218 gallons was calculated as follows:

$14.7 \times 4{,}007 = 74.7 \times V_f$

$\frac{58{,}902.9}{74.7} = V_f$ (volume of air in tank at 60 psig)

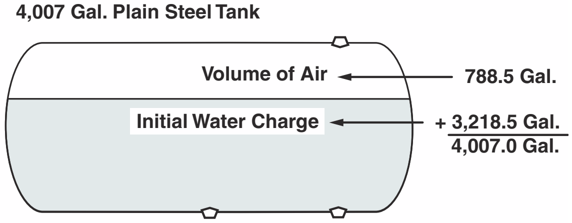
V_f =788.5 gallons of air

Then:

$V_a - V_f$ = Volume of water in tank at 60 psig

4,007 – 788.5 = 3,218.5 gallons of additional water in system

4,007 Gal. Plain Steel Tank



Once the determination is completed, the designer would proceed to recalculate the additional amount of expanded water and re-size the tank to accommodate the larger total amount of expanded water that would be generated as the system reaches and maintains design temperature levels.

Since this will result in a still larger amount of initial water charge (hence a still greater total system water volume), a still larger tank will be required.

Summation of Sizing Methods

In summation, it is recommended that the designer use either the ASHRAE formula method for accurately estimating tank sizes or the Critical sizing method for computing the exact minimum tank volume required.

Since the ultimate goal in selection and sizing of hydro-pneumatic tanks is one of the minimum size and weight, the sizing methods (ASHRAE and Critical) that follow will be for diaphragm-type or bladder-type tanks only. They cannot be used for sizing plain steel expansion tanks.

Preliminary Sizing Data Required

Before actual sizing computations are begun, certain basic data must be determined:

A. Total System Volume

This value must be determined before sizing by the ASHRAE formula method or the Critical sizing method can be used. While in some smaller systems, average content tables, based on the BTUH carrying capacity, or load, of the system can be used for quick approximation of total system volume, in the case of larger systems, this value is critical in accurately computing the amount of expanded water the system will generate. The only accurate method of determining this is a complete compilation of all system component water contents.

Water Content of Boilers – Refer to manufacturer's literature.

Water Content of Unit Heaters, Fan Coil Units and Convectors – Since these contain small amounts of water and are uniform in volumetric ratios, a BTUH output conversion factor will be sufficiently accurate. Refer to Table 1, Ref-6 for these factors.

Water Content of Commercial Finned Tube, Baseboard Radiation, and Piping – Refer to Table 2,Ref-6 "Volume of Water in Pipe Tubing." Consider commercial finned tube as steel pipe and baseboard radiation as copper tubing.

Water Content of Heat Exchangers – Refer to Table 3,Ref-6.

B. Determine Tank Location

Refer to Chapter 15, ASHRAE Handbook, or to Chapter One, Section B of this Engineering Handbook.

C. Determine System Pressure Values at Tank Location

Refer to Chapter 15, ASHRAE Handbook, or to Chapter One, Sections A and B of this Engineering Handbook.

D. Determine Maximum Average Design Temperature (t)

ASHRAE and Critical Sizing Methods for Diaphragm-Type and Bladder-Type Tanks Only

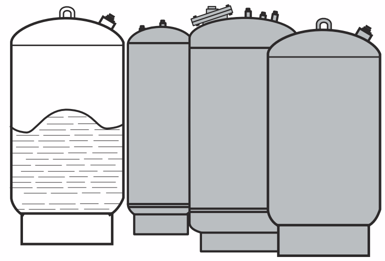


Table 1 – Water Content... Unit Heaters, Fan Coil Units and Convectors

(BTUH to gallons conversion factors)

	Gal./10,000 BTUH At 200°F	Gal./10,000 BTUH At 180°F
Convectors	0.64	–
Unit heaters	–	0.2
Fan coil units	–	0.2

Table 2 – Volume of Water in Pipe and Tubing

Gallons per lineal foot

Nominal Pipe Size in Inches	Steel Pipe Gal/Foot	Copper Tube Gal/Foot
$\frac{1}{2}$	0.016	0.012
$\frac{3}{4}$	0.028	0.025
1	0.045	0.043
$1\frac{1}{4}$	0.078	0.065
$1\frac{1}{2}$	0.105	0.092

Table 3 – Water Content of Heat Exchangers

Shell Dia. (nominal pipe size in inches)	Gal/Foot of Shell Length In Shell	Gal/Foot of Shell Length In Tubes
4	0.4	0.2
6	1.0	0.5
8	1.8	0.9
10	2.4	1.2
12	4.0	2.2

Table 7A – Examples of

Total Tank Volumes and Expanded Water Volumes of "AX"-Series or "L"-Series Model EXTROL Hydro-Pneumatic Tanks.

Use for calculated total tank volumes of 132 gallons or less

Total Volume Gal.	Expanded Water Vol. Gal.	Model no.
7.8	2.5	AX-15
10.9	2.5	AX-20
21.7	5.0	AX-40
22.0	11.0	85-LBC
44.0	27.0	165-LBC
132.0	53.0	500-LBC

Table 7B – Examples of

Total Tank Volumes of "L"-Series EXTROL Diaphragm Hydro-Pneumatic Tanks. Use for Calculated Total Tank Volumes of 158 Gallons or more.

Total Volume Gal.	Model no.
158.0	600-L
211.0	800-L
264.0	1000-L
317.0	1200-L
370.0	1400-L
422.0	1600-L
528.0	2000-L
660.0	2500-L
792.0	3000-L
925.0	3500-L
1,057.0	4000-L

E. Importance of Listed Acceptance Volumes in Smaller EXTROL® Diaphragm-Type Hydro-Pneumatic Tanks

Sizing AX Model EXTROLS – When the total tank volume calculated is 132 gallons or less, the amount of expanded water calculated must be equal or less than the acceptance volumes listed in Table 7A. If both the calculated total tank volume and the amount of expanded water are not met by the listed volume and acceptance volume, select the next larger AX or LBC model that meets both volume and acceptance.

Sizing L–Series EXTROLS – If the calculated total tank volume is larger than 158 gallons, refer to Table 7B and select for total tank volume only.

I. Sizing EXTROL® Diaphragm-Type Hydro-Pneumatic Tanks by the Formula Method (ASHRAE)

This formula is published by ASHRAE for use in sizing diaphragm-type expansion tanks (see Chapter 15, ASHRAE handbook). It may be used to calculate the approximate size of expansion tanks for systems with design temperatures that lie in the range of 160°F to 280°F, and with the system fill water temperature assumed to be 40°F. It is accurate from +0.5% to -9%. The formula is stated:

$$V_t = \frac{(0.00041_t - 0.04666)\, V_s}{1 - \frac{P_f}{P_o}}$$

Where: V_t = The minimum tank volume.

t = Maximum average design temperature.

V_s = Total system water content.

P_f = The initial or minimum operating pressure at the tank expressed in pounds per square inch, absolute (Psia).

P_o = The final or maximum operating pressure at the tank expressed in Psia.

Converting Pressure values to Psia

Psig + 14.7 = Psia

Sizing example using the formula method:

System water volume (V_s): . . . . .1,135 gal.

Maximum average design temperature (t): . . . . .210°F

Minimum operating pressure at the tank (P_f): . . . . .35 psig

Maximum operating pressure at the tank (P_o): . . . . .65 psig

System fill water temperature (T_f): .40°F

Computations:

1. $$V_t = \frac{(0.00041 \times 210 - 0.0466)\ 1{,}135}{1 - \dfrac{35 + 14.7}{65 + 14.7}}$$

2. $$V_t = \frac{\text{44.8 gallons expanded water}}{\text{0.376 acceptance factor}}$$

3. V_t = 119.1 gallons, minimum EXTROL® total volume

4. Table 7A shows that *AX-240, AX-240V* have a total volume of *132 gallons* and will accept up to *46 gallons* of expanded water. Either an AX-240 or an AX-240V (vertical style) would be the correct size.

II. Sizing EXTROL® Diaphragm-Type Hydro-Pneumatic Tanks by Critical Sizing Method

This sizing method is recommended when Critical sizing is required and/or when system fill temperatures are higher than 40°F. **It involves three steps:**

A. Determining Expanded Water

Refer to Table 5, "Net Expansion of Water" also found in the Reference Supplement (Page Ref-7).

On the horizontal base line, find the initial or fill temperature (T_f). On the vertical base line, find the final, or maximum average design temperature (t). At the intersection of the two columns, read the net expansion factor. Multiply the total system water content (V_s) by the expansion factor to determine the exact amount of expanded water.

B. Determining Acceptance factor

Refer to Table 6, "Acceptance factors for initial and final pressures" in Reference Supplement (Page Ref-10).

On the horizontal base line, "Initial, or Minimum Operating Pressures (P_f)," find the correct pressure value. On the vertical base line, "Final, or maximum operating pressure (P_o)", find the correct pressure value. At the intersection of the two columns, read the acceptance factor.

Table 7A –

Total Tank Volumes and Expanded Water Volumes of AX-Series or "L"-Series Model EXTROL® Hydro-Pneumatic Tanks. Use for Calculated Total Tank Volumes of 132 Gallons or Less.

Total Volume Gal.	Expanded Water Vol. Gal.	Model No.
7.8	2.5	AX-15, AX-15
10.9	2.5	AX-20, AX-20V
21.7	5.0	AX-40, AX-40V
33.6	11.5	AX-60, AX-60V
44.4	23.0	AX-80, AX-80V
55.7	23.0	AX-100, AX-100V
67.0	23.0	AX-120, AX-120V
77.0	23.0	AX-144, AX-144V
90.7	34.5	AX-180, AX-180V
110.7	34.5	AX-200, AX-200V
131.7	46.0	AX-240, AX-240V
132.0	53.0	500-LBC

Table 5 – Net Expansion of Water

Factors for calculating Net Expansion of Water

Gross expansion minus system piping expansion.

(Based on expansion of steel.)

Final Temp °F	Initial Temperature °F 40	45	50	55	60	65	70	75	80	85
50°	.00006	.00008	-							
55°	.00025	.00027	.00019	-						
60°	.00055	.00057	.00049	.00030	-					
65°	.00093	.00095	.00087	.00068	.00038	-				
70°	.00149	.00151	.00143	.00124	.00094	.00056	-			
75°	.00194	.00196	.00188	.00169	.00139	.00101	.00045	-		
80°	.00260	.00262	.00254	.00235	.00205	.00167	.00111	.00066		
85°	.00326	.00328	.00320	.00301	.00271	.00233	.00177	.00132	.00066	
90°	.00405	.00407	.00399	.00380	.00350	.00312	.00256	.00211	.00145	
95°	.00485	.00487	.00479	.00460	.00430	.00392	.00336	.00291	.00225	.00159

Table 6 – Acceptance Factors $\left(1 - \frac{P_f}{P_o}\right)$

(Use gauge pressure)

P_o Maximum Operating Pressure PSIG	P_f-Minimum operating pressure at tank (PSIG) 5	10	12	15	20	25	30	35	40
10	0.202	-							
12	0.262	0.075	-						
15	0.337	0.168	0.101	-					
20	0.432	0.288	0.231	0.144	-				
25	0.504	0.378	0.328	0.252	0.126	-			
27	0.527	0.408	0.360	0.288	0.168	-			
30	0.560	0.447	0.403	0.336	0.224	0.112	-		
35	0.604	0.503	0.463	0.403	0.302	0.202	0.101	-	-
40	0.640	0.548	0.512	0.457	0.366	0.274	0.183	0.091	-
45	0.670	0.586	0.553	0.503	0.419	0.335	0.251	0.168	0.838

Table 7B –
Total Tank Volumes of "L"-Series EXTROL® Hydro-Pneumatic Tanks. Use for Calculated Total Tank Volumes of More than 132 Gallons.

Total Volume Gal.	Model No.
158	600-L
211	800-L
264	1000-L
317	1200-L
370	1400-L
423	1600-L
528	2000-L
660	2500-L
792	3000-L
925	3500-L
1057	4000-L

C. Computing EXTROL® Size

Divide the amount of expanded water by the acceptance factor to determine the minimum total tank volume required (V_t).

Or

$$V_t = \frac{\text{Amount of expanded water in gallons}}{\text{Acceptance factor}}$$

Sizing Example Using Critical Sizing Method:

System Water Volume (V_s): . . . . .4400 gal.
Maximum Average Design Temperature (t): . . . . .230°F
Minimum Operating Pressure at the Tank (P_f): . . . . . .50 psig
Maximum Operating Pressure at the Tank (P_o): . . . .110 psig
System Fill Water Temperature (T_f): . . . . .70°F

1. From Table 5 Ref-7, find the intersecting point of vertical column, "70°F" and horizontal column, "230°F" and read "0.0461."
2. 0.0461 x 4,400 = 202.8 gallons of expanded water.
3. From Table 6 Ref-10, find the intersecting point of vertical column,"50 psig" and horizontal column,"110 psig" and read "0.481."
4. $V_t = \frac{202.8}{0.481}$ = 421.6 gallons, minimum EXTROL total volume
5. Table 7B shows that 1600-L EXTROL has a total volume of 423 gallons. This would be the correct size.

Chapter Three

Standard Installation Procedures for EXTROL® Hydro-Pneumatic Pre-Pressurized Compression Tanks

A. Sizing the EXTROL System Connection Lines

An important consideration in a hot water heating system is careful selection of the pipe size of the line connecting the compression tank to the system.

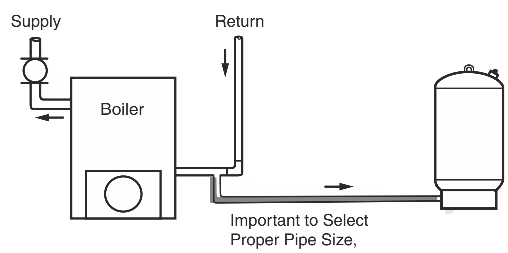
Size of system connection line depends upon the rate of flow of expanding water from system to tank.

In sizing the compression tank, it is the amount of expanded water that is the critical factor. However, when calculating the size of the piping connecting the tank to the system, it is the rate at which the expanding water will flow from the system to the tank that is the critical selection base.

Size of connection line must be large enough to keep the pressure drop low.

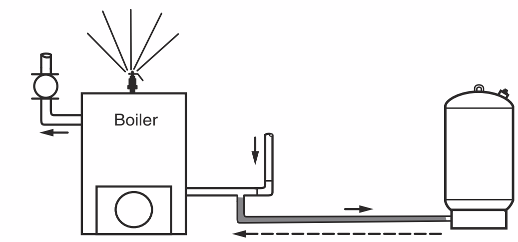
Usually, the rate of flow of expanded water in the connection line is low. In such cases, even the smallest pipe size (1/2") will not result in a significant pressure drop or friction loss.

However, in larger installations where higher temperatures and critical operating pressure ranges are employed, excessive rates of flow often occur in compression tank connection lines. If the connection line were undersized, the resulting pressure drop could be high enough to activate the relief valve even though the maximum operating pressure in the tank is below the 10% safety margin established in the initial system design.



Relief valve setting .... 100 psig
Minus 10% .... -10 psig

Max. operating pressure .... 90 psig
Additional pressure .... +11 psig
From pressure drop .... 101 psig



Undersized connection Line to System Results in Pressure Drop. Relief Valve is Opened.

Relative Density and Volume of Water

The mass of one cubic centimeter of water at 4°C is taken as a unit. The values given are numerically equal to the absolute density in grams per milliliter.

(Smithsonian Tables, compiled from various authors)

Temp. °F	Temp. °C	Density	Volume	Temp. °F	Temp. °C	Density	Volume
	-10	0.99815	1.00186	95.0	+35	0.99406	1.00598
	-9	843	157	96.8	36	371	633
	-8	869	131	98.6	37	336	669
	-7	892	108	100.4	38	299	706
	-6	912	088	102.2	39	262	743
	-5	0.99930	1.00070	104.0	40	0.99224	1.00782
	-4	945	055	105.8	41	186	821
	-3	958	042	107.6	42	147	861
	-2	970	032	109.4	43	107	901
	-1	979	021	111.2	44	066	943
	0	0.99987	1.00013	113.0	45	0.99025	1.00985
	1	993	007	114.8	46	0.98982	1.01028
	2	997	003	116.6	47	940	072
	3	999	001	118.4	48	896	116
39.2	4	1.00000	1.00000	120.2	49	852	162
41.0	5	0.99999	1.00001	122.0	50	0.98807	1.01207
42.8	6	997	003	123.8	51	762	254
44.6	7	993	007	125.6	52	715	304
46.4	8	988	012	127.4	53	669	349
48.2	9	981	019	129.2	54	621	398
50.0	10	0.99973	1.00027	131.0	55	0.98573	1.01448
51.8	11	963	037	140.0	60	324	705
53.6	12	952	048	149.0	65	059	979
55.4	13	940	060	158.0	70	0.97781	1.02270
57.2	14	927	073	167.0	75	489	576
59.0	15	0.99913	1.00087	176.0	80	0.97183	1.02899
60.8	16	897	103	185.0	85	0.96865	1.03237
62.6	17	880	120	194.0	90	534	590
64.4	18	862	138	203.0	95	192	959
66.2	19	843	157	212.0	100	0.95838	1.04343
68.0	20	0.99823	1.00177	230.0	110	0.9510	1.0515
69.8	21	802	198	248.0	120	0.9434	1.0601
71.6	22	780	221	266.0	130	0.9352	1.0693
73.4	23	756	244	284.0	140	0.9264	1.0794
75.2	24	732	268	302.0	150	0.9173	1.0902
77.0	25	0.99707	1.00294	320.0	160	0.9075	1.1019
78.8	26	681	320	338.0	170	0.8973	1.1145
80.6	27	654	347	356.	180	0.8866	1.1279
82.4	28	626	375	374.0	190	0.8750	1.1429
84.2	29	597	405	392.0	200	0.8628	1.1590
86.0	30	0.99567	1.00435	410.0	210	0.850	1.177
87.8	31	537	466	428.0	220	0.837	1.195
89.6	32	505	497	446.0	230	0.823	1.215
91.4	33	473	530	464.0	240	0.809	1.236
93.2	34	440	563	482.0	250	0.794	1.259

For example:

Assume a system with the following characteristics:

Total system volume (V_s) . . . . .7,000 gallons
Max. average design temperature(t) . . . . .270°F
Minimum operating pressure (P_f) . . . . .65 psig
Maximum operating pressure (P_o) . . . . .90 psig
Relief valve setting . . . . .100 psig
Expansion tank connection pipe size . . . . .1/2"
Length of tank connection pipe .30 Ft

When this system reaches its maximum average design temperature of 270°F, the 7,000 gallons will have generated an additional 468 gallons of expanded water. At the final stage of the temperature rise, 260°F to 270°F, this expanded water will flow through the pipe connecting the expansion tank to the system at the rate of 10.7 gpm.

At this rate of flow, the pressure drop in the 30-foot length of 1/2" connection piping will be 25 feet or 11 psig.

During this stage of system operation, the pressure will be at its maximum operating pressure (P_o) or 90 psig at the tank location. The additional 11 psig will cause pressure to rise above the relief valve setting of 100 psig and the valve will open.

For this reason, it is sound engineering practice to select a connection pipe size large enough to ensure negligible pressure drop.

The higher the temperature the greater the rate of flow.

The rate of expansion of system water increases as the temperature rises to the design operating range. The higher the temperature in the system piping, the faster the expansion takes place. Hence, the greater the rate of flow through the connection line from the system to the compression tank.

For Example: Between 90°F and 100°F, system water will expand in volume 0.17%. Between 260°F and 270°F, the expansion of volume will be 0.51%.

The expansion factor for a 10°F rise between any two points in the system temperature range as it rises from cold to maximum temperature is based on volume factors taken from the tables and an acceptable factor for the expansion of metallic system components. (See Chapter Three, Section A and Chapter Two, Section B.)

By multiplying this expansion factor for any given 10°F increment in the operating range by the total system volume, we can arrive at the amount of expanded water that the system will generate as the temperature rises through the 10°F increment.

For Example: (Figure. B-4)

In the system with a total system water content (V_s) of 7,000 gallons:

1. When the temperature rises from 260°F to 270°F, the volume of system water will expand by 0.5% or

$$0.0051 \times 7{,}000 = 35.7 \text{ gallons}$$

Therefore, 35.7 gallons of expanded water will flow through the piping connecting the system to the expansion tank when the system temperature rises from 260°F to 270°F.

To determine the rate of flow through the connection piping, we must calculate the length of time required for the expansion to take place.*

$$\frac{V_s \times W \times C \times \Delta T \times 60}{\text{BTUH}} = \text{minutes}$$

Where:

V_s = System water content in gallons

W = Density of water in lbs., at Maximum average design temperature

C = Specific heat of water BTUH /(lb)(°F), at maximum average design temperature

ΔT = 10°F increment (260°F to 270°F)

BTUH = Output of the heat generator in BTU's per hour.

2. If we assume that in the sample system above, a heat generator transfers 10 million BTU per hour to the system, then:

$$\frac{7{,}000 \times 7.8 \times 1.02 \times 10°F \times 60}{10{,}000{,}000} = 3.34 \text{ minutes}$$

Then the 35.7 gallons of expanded water that is generated when the system temperature rises from 260°F to 270°F will flow through the connection piping in 3.34 minutes.

3. Once the amount of expanded water is known and the length of time required to generate it is know, the rate of flow through the connection piping can be calculated:

$$\frac{\text{Amount of expanded water, in gallons}}{\text{Time required to generate it}} = \text{rate of flow, gpm}$$

Or:

$$\frac{35.7}{3.34} = 10.7 \text{ gpm}$$

Then, in the sample system above (Figure B-4), the expanded water generated as the system temperature rises from 260°F to 270°F will flow from the system through the connection piping to the expansion tank at a rate of 10.7 gallons per minute.

Since the higher the temperature, the greater the rate of flow, the selection of the connection pipe size should be based on the last 10°F increment of temperature rise.

The amount of total system water content has no bearing on the rate of flow of expanded water.

In the example given above (Figure B-4), the amount of water in the system will have no bearing on the rate of flow through the connection piping.

* Calculated at maximum rate of system temperature increase.

Table 5 – Net Expansion of Water

Factors for calculating Net Expansion of Water

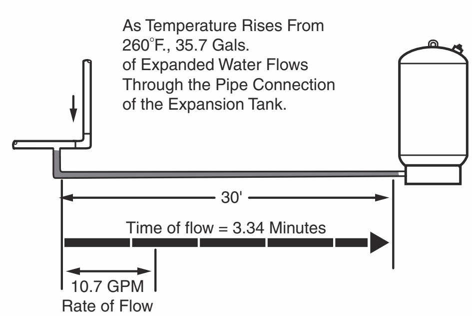
Gross expansion minus system piping expansion.
(Based on expansion of steel.)

Final Temp	Initial Temperature °F									
°F	235	240	245	250	255	260	265	270	275	280
240°	.00229									
245°	.00458	.00229								
250°	.00693	.00464	.00235							
255°	.00938	.00709	.00480	.00245						
260°	.01183	.00954	.00725	.00490	.00245					
265°	.01429	.01200	.00971	.00736	.00491	.00246				
270°	.01694	.01465	.01236	.01001	.00756	.00511	.00265			
275°	.01965	.01736	.01507	.01272	.01027	.00782	.00536	.00271		
280°	.02235	.02006	.01777	.01542	.01297	.01052	.00806	.00541	.00270	
285°	.02509	.02280	.02051	.01816	.01571	.01326	.01080	.00815	.00544	.00274
290°	.02799	.02570	.02341	.02106	.01861	.01616	.01370	.01105	.00834	.00564
295°	.03089	.02860	.02631	.02396	.02151	.01906	.01660	.01395	.01124	.00854
300°	.03379	.03150	.02921	.02686	.02441	.02196	.01950	.01685	.01414	.01144

Table 10 – Properties of Water

Temp. °F	Pressure PSIA	Density LB./GAL.	Specific Heat BTUH/LB/°F	Enthalpy BTUH/LB
180	-	-	1.003	-
190	-	-	1.004	157.94
200	-	8.04	1.006	167.99
210	-	8.01	1.007	178.05
220	17.19	7.97	1.009	188.13
230	20.78	7.94	1.010	198.23
240	24.97	7.90	1.012	208.34
250	29.83	7.86	1.015	218.48
260	35.43	7.82	1.017	228.64
270	41.86	7.79	1.020	238.84
280	49.20	7.75	1.022	249.06
290	57.56	7.71	1.025	259.31
300	67.01	7.66	1.028	269.59

Figure. B-4



V_s=7,000 Gallons

$$0.0051 \times 7{,}000 = 35.7 \text{ gal.}$$

$$\frac{7{,}000 \times 7.8 \times 1.02 \times 10°F \times 60}{10{,}000{,}000} = 3.34 \text{ minutes}$$

$$\frac{35.7}{3.34} = 10.7 \text{ gpm (rate of flow)}$$

V_s=3,500 Gallons

$$0.0051 \times 3{,}500 = 17.8 \text{ gal.}$$

$$\frac{3{,}500 \times 7.8 \times 1.02 \times 10°F \times 60}{10{,}000{,}000} = 1.67 \text{ minutes}$$

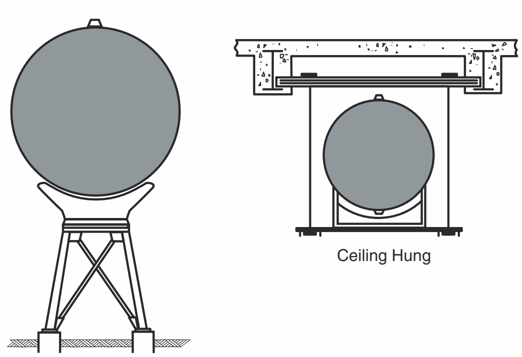
$$\frac{17.8}{1.67} = 10.7 \text{ gpm (rate of flow)}$$

Rate of Flow Is the Same Regardless of the Amount of Water in the System

Table 8 – Minimum pipe size from tank to system (in inches)

	Equivalent length up to 10' Max. average design temp. °F					Equivalent length 11' to 30' Max. average design temp. °F					Equivalent length 31' to 100' Max. average design temp. °F				
MBH	100°	150°	200°	250°	300°	100°	150°	200°	250°	300°	100°	150°	200°	250°	300°
1000	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	3/4	3/4	1/2	3/4	3/4	3/4	1
2000	1/2	1/2	1/2	1/2	3/4	1/2	3/4	3/4	3/4	1	3/4	3/4	1	1	1 1/4
3000	1/2	1/2	3/4	3/4	3/4	3/4	3/4	1	1	1	3/4	1	1	1 1/4	1 1/4
4000	1/2	3/4	3/4	3/4	1	3/4	1	1	1	1 1/4	1	1	1 1/4	1 1/4	1 1/4
5000	1/2	3/4	3/4	1	1	3/4	1	1	1 1/4	1 1/4	1	1 1/4	1 1/4	1 1/4	1 1/2
6000	1/2	3/4	1	1	1	3/4	1	1 1/4	1 1/4	1 1/4	1	1 1/4	1 1/4	1 1/2	1/2
7000	3/4	1	1	1	1 1/4	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1/4	1 1/2	2
8000	3/4	1	1	1	1 1/4	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/4	1 1/4	1 1/2	1 1/2	2
9000	3/4	1	1	1 1/4	1 1/4	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/4	1 1/4	1 1/2	2	2
10,000	3/4	1	1	1 1/4	1 1/4	1	1 1/4	1 1/4	1 1/2	1 1/2	1 1/4	1 1/2	2	2	2
12,000	1	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	2	1 1/4	1 1/2	2	2	2
14,000	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/2	2	2	1 1/4	2	2	2	2 1/2
16,000	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	2	2	1 1/2	2	2	2 1/2	2 1/2
18,000	1	1 1/4	1 1/4	1 1/2	1 1/2	1 1/4	1 1/2	2	2	2	1 1/2	2	2	2 1/2	2 1/2
20,000	1	1 1/4	1 1/4	1 1/2	1 1/2	1 1/4	1 1/2	2	2	2	1 1/2	2	2 1/2	2 1/2	2 1/2

Plain Steel Tank



Floor Mounted

Ceiling Hung

For example:

Assume a total water content of (V_s) of 3,500 gallons.

Then:

$$0.0051 \times 3{,}500 = 17.8 \text{ gal.}$$

$$\frac{3{,}500 \times 7.8 \times 1.02 \times 10°F \times 60}{10{,}000{,}000} = 1.67 \text{ minutes}$$

$$\frac{17.8}{1.67} = 10.7 \text{ gpm}$$

The rate of flow is the same regardless of the amount of water in the system.

Therefore, the correct size for the connection piping must be large enough to allow a flow of 10.7 gpm through the piping with a negligible pressure drop.

Table 8 in the Reference Supplement (Page Ref-12) lists the correct size for maximum average design temperatures and BTUH outputs of heat generators. The pipe sizes shown are those required to keep pressure drops negligible in the connection piping in the indicated equivalent lengths of piping. Since the 30 foot length of pipe will be carrying expanded water generated at the maximum average design temperature of 270°F, the selection of correct size would be 1 1/2".

B. Individual Connection Pipe Size for Multiple EXTROL Tanks

In the case of multiple EXTROL installations, the pipe size selected from the Table above will be the correct size for the manifold or header to the tanks.

The sizes of the individual tank connection lines from the manifold to each tank can easily be determined by dividing the BTUH output of the heat generator by the number of EXTROL tanks and referring again to Table 8.

For example:

In the sample system, 4 "L"-Series 2000 EXTROL would be required. Dividing the BTUH output by four will give us 2,500,000 BTUH per EXTROL. Referring to Table 8 under the 30-ft. length, we find that a 1" pipe size would be required to connect each EXTROL to the manifold.

C. Vertical Versus Horizontal Installation of EXTROL

Before the advent of the EXTROL concept of a sealed-in, permanently pre-pressurized air cushion, the designer has usually had to install the plain steel tank at a predetermined height above the heat generator. This has been necessary to allow proper air control procedures to be employed so as to prolong the operating period before complete water-logging of the tank occurs.

The plain steel tank then was usually installed in a horizontal position with either floor stands or ceiling suspension. This leads to additional time and material costs for the tank suspension assemblies. (See Chapter Two, Section A.)

However, since the EXTROL can never water-log, it can be placed anywhere in the system. Normally, this would indicate a free-standing floor installation with the EXTROL supported by its own integral floor stand.

There is no further requirement for additional floor or ceiling support assemblies.

With EXTROL, the designer should carefully weigh the economics of horizontal installation, comparing the floor space saved by suspending an EXTROL in a horizontal position against the additional cost of floor or ceiling support assemblies.

In most cases, the minimum floor space required for a free-standing vertical installation of EXTROL will be much more economical than supporting the EXTROL in an elevated horizontal position.

However, the EXTROL may be installed either vertically or horizontally at the designer's discretion. Smaller models (AX-Series) will easily be accommodated by lightweight support racks. Larger EXTROLS ("L"-Series) will be more economically installed vertically, free-standing on their integral floor stands.

D. Earthquake-Proof Installation

If an earthquake-proof installation is required, the EXTROL must be floor mounted vertically, on its integral floor stand. Three mounting lugs constructed of right angle steel and bolts are employed to bolt the EXTROL directly to the equipment room floor. This adaptation provides an EXTROL installation that will withstand seismic resonant frequencies up to 25Hz.

For the correct size of bolts, consult the following table and installation drawings.

Reinforcing Earthquake-Proof Foundation Bolt

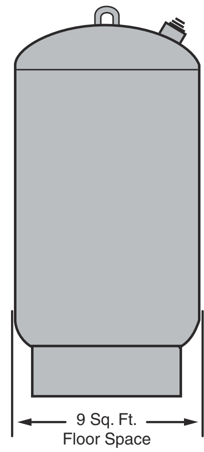
EXTROL diameter in inches	Minimum bolt size in inches
24	1/2
30	5/8
36	3/4
48	1

E. Tank Shut-Off and Drain Valves

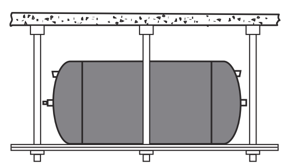
As with any major system component, it is recommended that both shut-off valves and drain valves be installed on each EXTROL line to allow the isolating and draining of the EXTROL. The use of a lock shield gate valve is recommended to eliminate the inadvertent closing of the shut-off valves during operation.

When hydrostatically testing system piping before initial operation, all EXTROL *tank shut-off valves must be closed.*

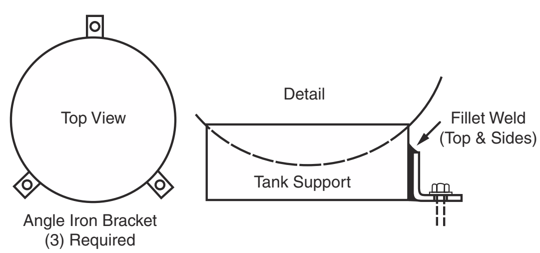
EXTROL Diaphragm-Type Tank



Free - Standing



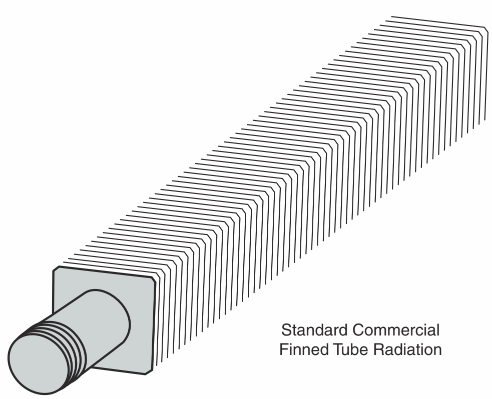
Horizontal Position

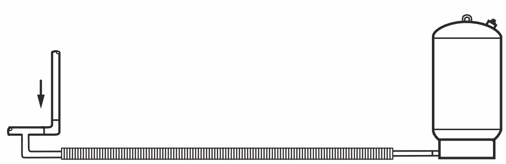


Important! Use of "Heating Leg" Required to Lower Operating Temperatures in Excess of 240°F at EXTROL Diaphragm

⚠ CAUTION

The manufacturers of the diaphragm material presently used in EXTROL expansion tanks restrict maximum operating temperatures at the diaphragm to 240°F. It is recommended that minimum length "heating leg" be used between the point of system connection and the EXTROL if operating temperatures at the diaphragm could be in excess of 240°F.


Standard Commercial Finned Tube Radiation


Finned Tubing as "Heating Leg"

F. Heat Recovery from EXTROL Connection Lines

During each operating cycle, expanded water flows through the connecting piping between the system and expansion tank. As the system is brought up to its maximum temperature, the rate of flow of expanded water increases. This combined with the higher temperature of the water introduced from the system results in a significant amount of BTUs not being available for space heating where needed and which consequently can be considered wasted.

With rising concern over energy usage in large mechanical HVAC systems, the BTUs available in the tank connection line make a good source of recovery energy.

Auxiliary Energy During Each System Cycle

It should be remembered that the heat (BTUs) recovered from the EXTROL connection line will become available when the main system is firing to meet demand.

Therefore, the recovered heat should be applied to areas large enough to absorb it without going beyond critical control limits or, conversely, to areas where the absence of the recovered heat (during system off-cycle) will not cause discomfort or damage.

It should only be used to supplement heat to another system with independent control sources and should not be applied to areas where over-heating could occur.

Simple Heat Recovery with a "Heating Leg"

The simplest and most economical method of heat recovery with EXTROL is to extend a properly sized run of standard commercial finned tube radiation from the point of system connection through an adjacent area where supplemental heat is desired, installing the EXTROL at the end of the run.

Since the EXTROL can be placed anywhere in the system, this "heating leg" can be as long as required to deliver recovered heat where desired.

For Example: A "heating leg" could be used to supply supplemental heat underneath or behind a bank of gas or electrical dryers in a building or laundry room. The recovered heat will heat primary air for the dryers, reducing the amount of required primary fuel.

Calculation of Amount of Commercial Finned Tube to Function as a "Heating Leg":

For any given system, based on BTUH rating of the heat generator and the maximum temperature of the expanded water, Table 9 (Page Ref 12-A) in the Reference Supplement shows:

1. The total amount of BTUH which can be recovered per operating cycle. This is an approximation only and depends on several factors including the length of the "heating leg" or the characteristics of the heat exchanger used.

2. The rate of heat recovery in BTUH which determines the rated capacity of the length of the "heating leg" or a heat exchanger. (Refer to manufacturer's literature.)

With the expansion tank connected to the system return, the maximum temperature of the expanded water could depend on the design temperature drop of the system or could be higher if outlet temperature to the system remains constant and the temperature drop decreases under light load conditions.

Example: In a system including a 5,000,000 BTUH heat generator and a maximum design temperature of 290°F with 40°F temperature drop, there is a large garage area adjacent where temperature control is not critical. How much heat could be recovered by use of expanded water flowing through 2-inch commercial finned tube installed in this space – and how much tube would be required?

It is determined that for this particular system, the maximum temperature at the system connection to the tank will be 250°F From Table 9 (Page Ref 12-A) in the Reference Supplement, we find there would be an estimated 110,000 Btuh per operating cycle which could be recovered if the expanded water flowed through the finned tube “heating leg” with an outlet temperature, at the tank, of 120°F or less.

The length of finned tube used can be selected by using the maximum BTUH carried by the expanded water – in the example, a maximum rate of heat transfer of 335,000 BTUH.

Referring to the manufacturer’s literature, it is indicated that approximately 300 feet should be installed to achieve optimum heat recovery.

A correction factor is shown if higher outlet temperatures from the radiation are desired to decrease the amount of finned tube required without appreciably reducing the heat recovered.

If the “heating leg” were to be used only to reduce the temperature of expanded water at the outlet to 240°F in order to protect the diaphragm in the EXTROL, 23 feet would be sufficient.

Heat Recovery with Heat Exchanger

By using a shell and tube heat exchanger designed for negligible friction loss on the expanded water side in the expansion tank connection line, heat can be recovered for many different types of systems where supplemental heat may be absorbed without adversely affecting temperature control.

Examples:

1. **Space heating using a water-to-water converter.**
2. **Large domestic hot water storage tanks.**
3. **Snow melting systems.**
4. **Large fuel oil storage tanks. In this application, the recovered heat is applied to the fuel oil through an intermediate hot water heating circuit.**

Table 9 – Heat Recovery

NOTE: 1. The approximate MBH available per cycle is shown in parenthesis.
2. The outlet temperature of expanded water from the heat exchanger or heating leg is assumed to be 120°. (see correction factors)
3. System design temperature drop is assumed to be 40°F.

	Maximum rate of heat transfer – MBH						
	Maximum temperature at tank system connection in °F						
System MBH	**220°**	**230°**	**240°**	**250°**	**260°**	**270°**	**280°**
1000	46 (10)	53 (15)	60 (20)	67 (20)	75 (25)	83 (30)	92 (35)
2000	92 (25)	105 (30)	120 (40)	134 (45)	150 (50)	167 (60)	184 (70)
3000	138 (35)	158 (45)	180 (55)	201 (65)	225 (80)	250 (90)	276 (100)
4000	184 (50)	211 (60)	240 (75)	268 (90)	300 (100)	333 (120)	368 (140)
5000	230 (60)	264 (75)	300 (90)	335 (110)	375 (130)	415 (150)	480 (180)
6000	276 (70)	317 (90)	360 (110)	402 (130)	450 (150)	500 (180)	552 (210)
7000	322 (85)	369 (100)	420 (130)	470 (150)	525 (180)	584 (210)	644 (250)
8000	368 (95)	422 (120)	480 (150)	537 (170)	600 (210)	667 (240)	736 (280)
9000	414 (110)	475 (130)	540 (160)	604 (200)	675 (230)	750 (280)	828 (320)
10,000	460 (120)	528 (150)	600 (180)	671 (220)	750 (260)	834 (310)	920 (360)
12,000	552 (150)	633 (180)	720 (220)	805 (260)	900 (310)	1000 (370)	1104 (430)
14,000	644 (170)	739 (210)	840 (260)	939 (310)	1050 (370)	1167 (430)	1288 (500)
16,000	736 (190)	845 (240)	960 (290)	1073 (350)	1200 (420)	1334 (490)	1472 (570)
18,000	828 (220)	950 (270)	1080 (330)	1208 (400)	1350 (470)	1500 (550)	1656 (640)
20,000	920 (240)	1056 (300)	1200 (370)	1342 (440)	1500 (530)	1668 (620)	1840 (720)
140°F	.80 (.95)	.82 (.96)	.83 (.97)	.85 (.98)	.86 (.98)	.87 (.98)	.88 (.98)
160°F	.60 (.85)	.64 (.88)	.67 (.90)	.69 (.92)	.71 (.93)	.73 (.94)	.75 (.95)
180°F	.40 (.66)	.45 (.73)	.50 (.78)	.54 (.81)	.57 (.84)	.60 (.86)	.63 (.88)
200°F	.20 (.38)	.27 (.50)	.33 (.59)	.38 (.66)	.43 (.71)	.47 (.75)	.50 (.79)
220°F	-	.09 (.19)	.17 (.33)	.23 (.44)	.29 (.53)	.33 (.60)	.38 (.65)
240°F	-	-	-	.08 (.17)	.14 (.30)	.20 (.40)	.25 (.48)

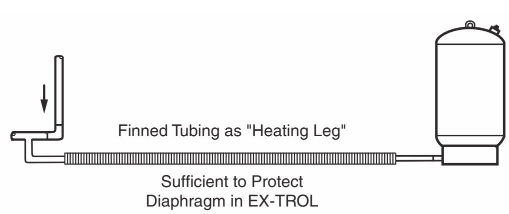


Figure B-5

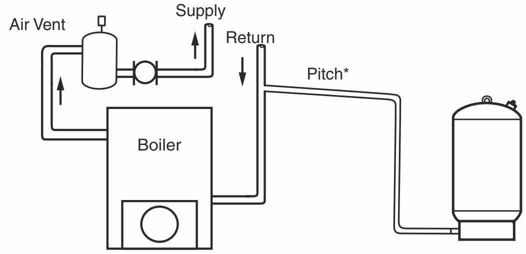


Figure B-6

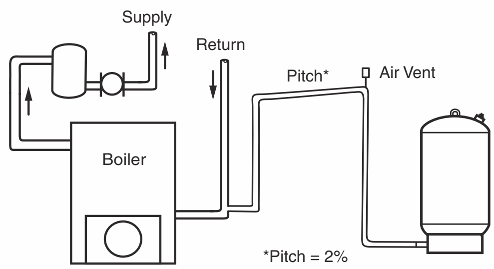


Figure B-7

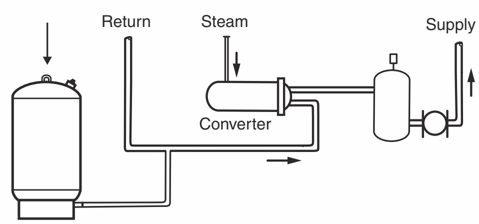
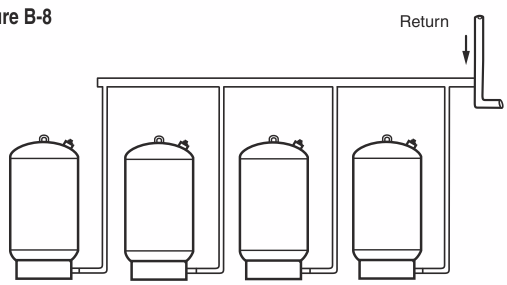


Figure B-8



G. Air Elimination on EXTROL Connection Lines

As in the supply and return lines of any heating system with EXTROL, all air can be and should be eliminated.

Since velocities in the connection piping between the EXTROL expansion tank and the system connection are very low, horizontal runs such as shown in Figure B-5 should be pitched up away from the tank and towards the system connection. Air will flow into the return line where it will be removed by the system's air elimination devices. (see Section C.)

In Figure B-6, the horizontal run should be pitched up towards the tank and a vent installed at the highest point.

H. Typical Specification

Expansion Tanks: Furnish and install as shown on the drawings ___EXTROL Model ____ pre-pressurized diaphragm-type expansion tank as manufactured by AMTROL Inc. It shall be air precharged to ____PSIG. It shall be suitable for a maximum working pressure of ____PSIG and shall be furnished with ASME stamp and certification papers. It shall have a sealed-in elastomer diaphragm or bladder suitable for an operating temperature of 240°F.

(EXTROL to be furnished with saddles for horizontal installation.)

(EXTROL to be furnished with integral ring mount for vertical installation.)

I. Typical Installation

1. EXTROL installed on heating boiler.
2. EXTROL installed with steam-water converter. (Figure B-7).
3. Battery of (4) EXTROLs on heating boiler. (Figure B-8).

J. Changing Air Charge on Job

FTC regulations limit air charge on tanks being carried by commercial carrier to 40 psig. The following procedures should be followed in changing the air charge at the job site:

1. The air charge in the tank should be checked with an air gauge, making sure that the tank is isolated from the system and all water drained from the tank.
2. If the charge is lower than the Minimum Operating Pressure required at the tank (P_f), the charge should be increased, using a source of clean, dry compressed air.

Section C

Definitions, analysis and corrective procedures regarding the problem of "System Air" in closed hot water and chilled water HVAC systems

Chapter One

"System Air" – Its Composition, Changes of State and Traditional Methods of Controlling

Basic Definitions of "System Air" and Its Physical Characteristics

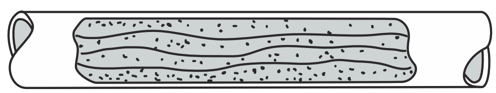
To properly approach the problem of "System Air" and its solution, we must first understand and agree to what it is and what it does in the closed piping system.

Traditionally, the term "System Air" had been widely used to designate the presence of gaseous fluids in the closed piping system.

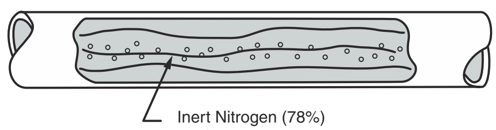
It should be more properly defined as a gas mixture which changes its physical form as well as its chemical composition during system operation and life.

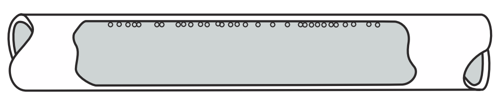
The chemical composition of air is mainly oxygen and nitrogen with a small portion of other gases which vary according to local conditions. For purposes of our discussion, here, we will assume that air is composed of 21% oxygen (including a small mixture of other gases) and 79% inert nitrogen by volume.

The Gas	% of Air
Nitrogen	**.78**
Oxygen	**.21**
(plus other gases)	



Oxygen (21%) Forms Ferrous Oxide or Rust





Free Air Bubbles

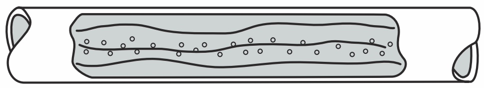
Changes in Chemical Composition

When air is introduced into a closed piping system filled with water, it undergoes a major change in chemical balance. As air is carried throughout the system in a dissolved state (in solution), its oxygen content combines with metallic surfaces to form oxides. In addition to causing corrosion, this portion of the "air" mass is no longer present as "air" in the system. The remaining air mass is the inert gas nitrogen. In other words, there is a reduction of the air mass in a gas state to 78% of its original volume. This change in chemistry begins fairly quickly after "air" is introduced. It should be noted that when in a gaseous state, oxygen will not readily combine with metallic surfaces.

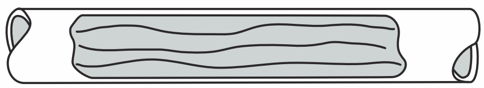
Changes in Physical Form

"System Air" can exist in the water-filled piping in three physical forms:

1. Free Air Bubbles: In this state, free air bubbles locate or collect at the top of vertical and horizontal pipes as well as at the top of system components such as boilers, terminal units and the expansion tank. This occurs when high temperatures and low pressures are present to allow the formation of air bubbles and when there is no system flow or in large diameter components where extremely low velocities allow vertical travel of the free air bubbles.



Entrained Air Bubbles

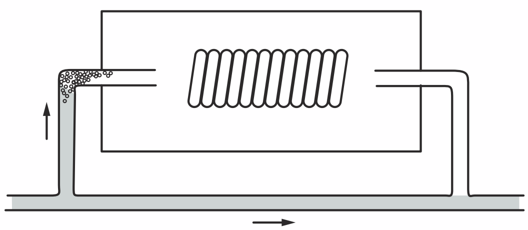


Air in Solution

2. Entrained Air Bubbles: When system flow is present and velocities are in excess of 1.5 feet per second, the free air bubbles are not allowed to rise and collect, but are carried through the piping system (whenever temperature and pressure conditions will permit the formation of free air bubbles entrained in the flowing fluid).

3. Air in Solution: At lower temperatures and higher pressures, air becomes more soluble in water. Under these conditions, the air is absorbed or dissolved in the water and carried in the solution. There are no bubbles present. It is much the same as the change occurring when sugar or salt is dissolved in a fluid. It is in this state that air is highly corrosive.

The Symptoms of "System Air" in Hot Water Heating and Chilled Water Piping Systems



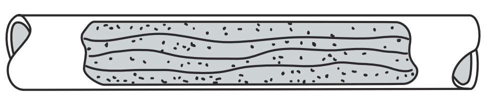
Blockage Caused by "Air Pockets" Can Restrict or Stop Circulation of Water

Inefficient Operation = Increased Maintenance and Energy Consumption

The predominant problem that occurs in piping systems afflicted with "System Air" is inefficient operation. Free gas bubbles entrained in system flow can cause harmful cavitation of pumps. Gas bubbles with system fluids not flowing tend to rise and collect in terminal units and high points in the piping system. Many times, a pocket of "air" will cause a blockage that is not dislodged when flow occurs. This restricts and even stops circulation of hot or chilled water to areas demanding space conditioning. This inefficient operation increases maintenance costs and energy consumption.

Corrosion

The oxygen content of "System Air" when carried in solution combines with all metallic components of the system including the piping. Deposits of rust also reduce efficient operation, increase energy consumption, and will result in increased maintenance and eventual replacement of inoperative components.



Oxygen Forms Ferrous Oxide or Rust

The Sources of "System Air"

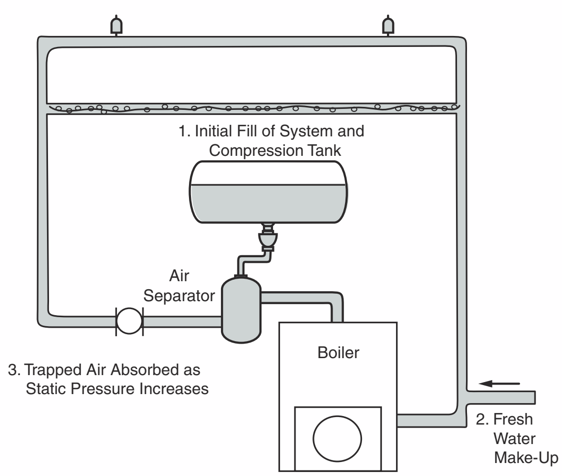
Designers have traditionally held that there are two main sources of "System Air" in the closed hot water heating or chilled water system.

1. Air Absorbed During the Initial Fill of the System

It has been contended that as water fills the piping, air is trapped in the system and becomes absorbed in the system water.

2. Air Carried in Solution in Fresh Make-Up Water

The other major source of "System Air" held by tradition is that make-up water which is added to the system during operation carries with it considerable amounts of air in the solution.



Plain Steel Expansion Tank Also Considered a Source of "System Air."

Designers, in recent years, contend that the air cushion in the plain steel tank also contributes air to the system.

This has been refuted by advocates of the plain steel tank as they claim "air control" procedures properly employed will not only result in the escape of air to the system, but also will allow "System Air" from initial fill and make-up water to be separated, collected and directed to the air cushion in the tank removing it from system piping.

"Air Control" vs. "Air Elimination"

Before a detailed examination of the characteristics of system air can be properly made, we must understand the principles or philosophies behind the two completely opposite practices currently used in combatting "System Air."

Air Control / Plain Steel Tank

The use of the plain steel tank as the pressurizing agent for hot water heating or chilled water systems dictates the necessity of "air control."

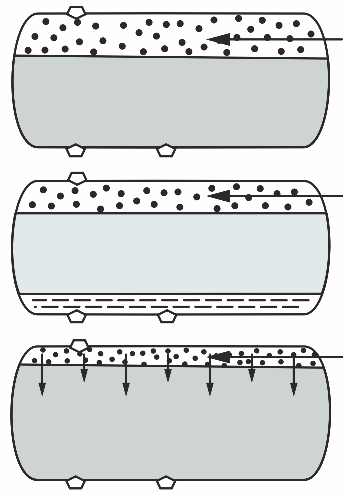
Since it is acknowledged that "air" is absorbed by system water and carried in solution from the tank to other parts of the system, the advocates of the plain steel tank strongly recommend separating and collecting free "air" bubbles at a point convenient to the expansion tank. Then, using special devices, direct the collected air bubbles back to the air cushion of the tank to maintain the proper air volume needed for system pressurization. Complete "air elimination" would remove the effective air volume from the system, hastening the loss of pressurization through waterlogging.

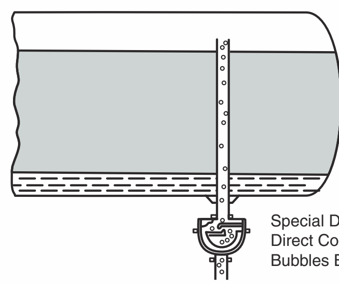
"Air Control" Based on Temperature Increase

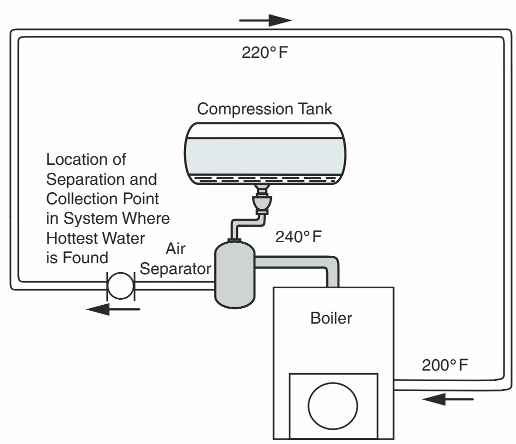
The underlying philosophy in "air control" is that air, carried in solution, is brought out of solution as temperature increases. As the temperature increases from ambient to design, water loses its capability to hold air in solution. The often quoted example of a glass of tap water forming air bubbles as it warms from cold tap temperature to room temperature is an example of this philosophy.

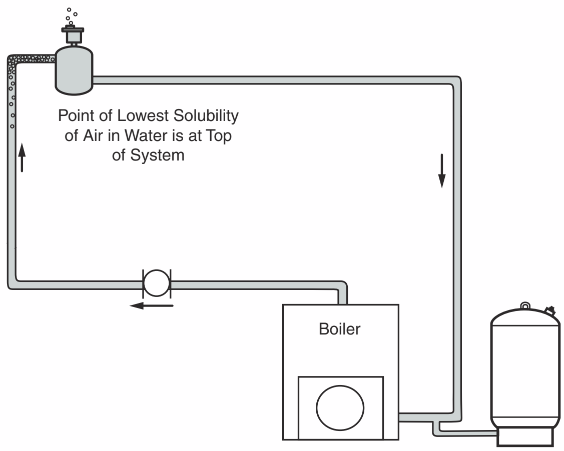
The familiar term, "boiling out the air," is based on this philosophy of air being brought out of solution by temperature increase.

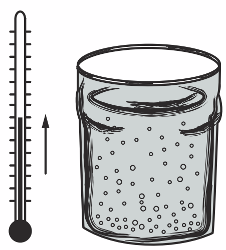
"Air Control" producers, then, dictate locating the separation and collection point in the system where the hottest water is found. This is the supply side, or top, of the boiler. This is also, remember, at the one system point where it is most convenient to direct free air from the separator back to the plain steel expansion tank which is installed directly above the boiler.

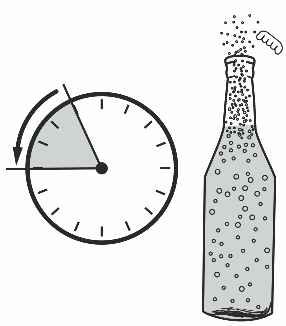












"Air Elimination" Requires a Captive Air Cushion

The basic difference between "air control" and "air elimination" is that with a plain steel tank air cannot be eliminated from the system as this will hasten the waterlogging problem and subsequent loss of vital system pressurization.

However, on the other hand, if the necessary air volume required to maintain system pressurization is sealed-in and not allowed contact with system water during temperature and pressure changes (as in a pre-pressurized diaphragm-type or bladder-type compression tank), then all air in the system piping can be eliminated from the system without adversely affecting pressurization and accommodation of expanded water.

Furthermore, since it is not necessary, or advisable, to separate and collect free "air" at a point convenient to the tank (there is no need to add "air" to the pressure cushion), then the separation and elimination point can be located in the one place where system water has the lowest capability to hold air in the solution. This point is usually *not* at the boiler.

Pressure Decrease is Key to Air Release

A look at the solubility tables 12 and 13 (pages REF-13 and REF-14 in the Reference Supplement) for "System Air" or nitrogen indicates that two changes in system operating conditions will lower the capability of water to hold air in solution. Temperature increase....and pressure decrease.

The traditional glass of warm tap water illustrates the lowered capability of water as it's temperature rises.

If one could also visualize a bottle of soda water, also brought to room temperature, one would see very few bubbles, until the cap was removed. Then the instant pressure decrease would cause a multitude of gas bubbles to instantly come out of solution.

This dramatically illustrates the greater affect pressure decrease has in lowering the capability of water to hold "air" or gas in solution.

In fact, as we shall see in Chapter Two of this section, pressure decrease and increase determine the rate and amount of "air" that comes out of solution…where it comes out…and where it goes back in. While temperature does have an effect…the greater impact of pressure change often counteracts the effect of temperature change and becomes the determining factor for the change from "air" in solution to free "air" bubbles and back.

Section C

Definitions, analysis and corrective procedures regarding the problem of "System Air" in closed hot water and chilled water HVAC systems.

Chapter Two

ANALYSIS OF THE CHARACTERISTICS OF AIR IN AN ENGINEERED HOT WATER HEATING AND CHILLED WATER SYSTEM

Composition of "System Air"

Because much of the material presented in this chapter will pertain to the characteristics of absorbed gas during the first period of operation before oxygen has significantly united with metallic surfaces, we will analyze the characteristics of "air" as being composed of 78% nitrogen and 21% oxygen (including a small amount of other gases).

Table 12 on page REF-13 in the Reference Section shows the maximum amount of air which can be held in solution in system water at varying temperatures and pressures expressed as a decimal equivalent of the percentage of water volume.*

*Measured at standard conditions, 32°F, at atmospheric pressure.

Heating System Conditions and Air Solubility

The family of curves shown in Figure C-1 has been prepared from data in Table 12 and illustrates more clearly than the data itself the relative effect of temperature and pressure on air solubility.

Temperature Increase Releases Air from Solution

For example, at a fixed pressure of 10 psig, we can see that as the temperature increases from 100°F to 200°F (Figure C-1, "A"), the capability of water to hold air in solution changes from 2.4% to 1.0%. In other words, if the water actually contained 2.4% air in solution, 1.42 % of the volume of the water would be released from solution and form air bubbles.

Temperature vs. Pressure Increase

As the temperature increases in a heating system, the pressure also increases. If the pressure increased to 28 psig ("B"), the capability of the water to hold air would remain at 2.4%.

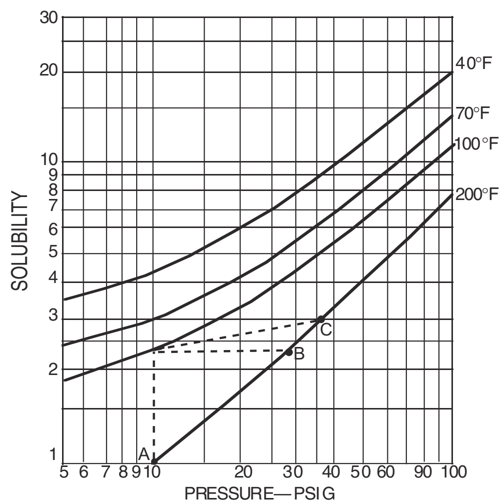
In another system, the pressure would rise to possibly 35 psig ("C"), in which case the capability would rise to 3.0%. If air bubbles were present in the piping, they would be absorbed until the water actually reached its full capability, 3.0%.

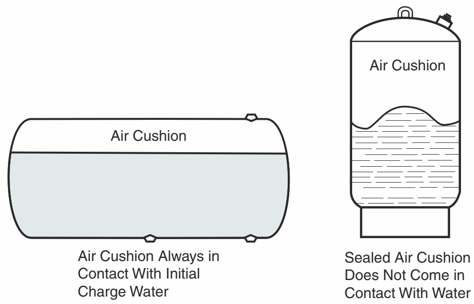
Table 12 – Solubility of Air in Water

Expressed as a fraction of water, volume, at STP

Temp (t) °F	Total Pressure, PSIG												
	0	10	20	30	40	50	60	70	80	90	100	110	120
40	0.0258	0.0435	0.0613	0.0790	0.0967	0.1144	0.1321	0.1499	0.1676	0.1853	0.2030	0.2207	0.2384
50	0.0223	0.0376	0.0529	0.0683	0.0836	0.0989	0.1143	0.1296	0.1449	0.1603	0.1756	0.1909	0.2063
60	0.0197	0.0333	0.0469	0.0605	0.0742	0.0878	0.1014	0.1150	0.1296	0.1423	0.1559	0.1695	0.1831
70	0.0177	0.0300	0.0423	0.0546	0.0669	0.0792	0.0916	0.1039	0.1162	0.1285	0.1408	0.1531	0.1654
80	0.0161	0.0274	0.0387	0.0501	0.0614	0.0727	0.0840	0.0954	0.1067	0.1180	0.1293	0.1407	0.1520
90	0.0147	0.0253	0.0358	0.0464	0.0569	0.0674	0.0750	0.0885	0.0990	0.1090	0.1201	0.1306	0.1412
100	0.0136	0.0235	0.0334	0.0433	0.0532	0.0631	0.0730	0.0829	0.0928	0.1027	0.1126	0.1225	0.1324
110	0.0126	0.0220	0.0314	0.0408	0.0501	0.0595	0.0689	0.0753	0.0877	0.0971	0.1065	0.1158	0.1252
120	0.0117	0.0206	0.0296	0.0385	0.0475	0.0564	0.0654	0.0744	0.0833	0.0923	0.1012	0.1102	0.1191
130	0.0107	0.0193	0.0280	0.0366	0.0452	0.0538	0.0624	0.0710	0.0796	0.0882	0.0968	0.1054	0.1140
140	0.0098	0.0182	0.0265	0.0348	0.0432	0.0515	0.0598	0.0681	0.0765	0.0848	0.0931	0.1015	0.1098
150	0.0089	0.0170	0.0251	0.0332	0.0413	0.0494	0.0574	0.0655	0.0736	0.0817	0.0898	0.0979	0.1060
160	0.0079	0.0158	0.0237	0.0316	0.0395	0.0474	0.0553	0.0632	0.0711	0.0790	0.0869	0.0945	0.1027
170	0.0068	0.0145	0.0223	0.0301	0.0378	0.0456	0.0534	0.0611	0.0689	0.0767	0.0844	0.0922	0.1000
180	0.0055	0.0132	0.0208	0.0285	0.0361	0.0438	0.0514	0.0591	0.0667	0.0744	0.0820	0.0879	0.0973
190	0.0041	0.0116	0.0192	0.0268	0.0344	0.0420	0.0496	0.0571	0.0647	0.0723	0.0799	0.0875	0.0950
200	0.0024	0.0099	0.0175	0.0250	0.0326	0.0401	0.0477	0.0552	0.0628	0.0703	0.0779	0.0854	0.0930
210	0.0004	0.0080	0.0155	0.0230	0.0306	0.0381	0.0457	0.0532	0.0607	0.0683	0.0758	0.0833	0.0909

Fig. C-1



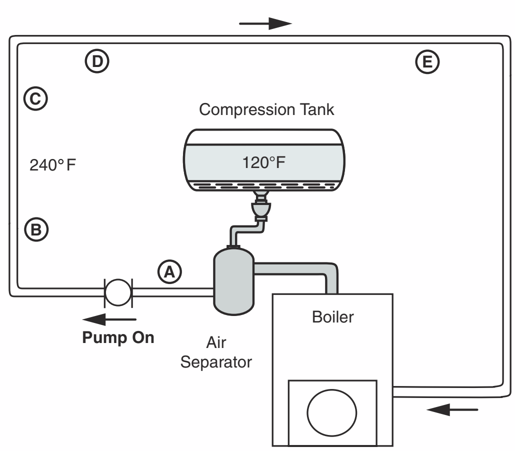


Free air bubbles do exist in one system location

The expansion tank must maintain a trapped air cushion to maintain system pressurization. In the plain steel expansion tank, the initial charge water is in constant direct contact with this cushion of "trapped" air. This water would absorb air from this cushion up to its maximum capability to hold air in solution.

In the diaphragm-type expansion tank, the air cushion is sealed and does not come into contact with system water. No air will be absorbed.

Analysis of the Characteristics of "System Air" with a Plain Steel Expansion Tank Installed



At Maximum Design Temperature

Because the problem of system air can be more critical in larger installations, we will use a typical large closed hot water heating system with the following characteristics for our analysis. For purposes of illustration, we will monitor temperatures and pressures at five specific points in the piping system and at the tank location:

1. Minimum operating pressure (static plus 5 psig adequate positive pressure)

Point A	65 psig
Point B	53 psig
Point C	14 psig
Point D	5 psig
Point E	5 psig
Expansion Tank	65 psig

2. Additive pump effect when pump is operating

Point A	0 psig
Point B	30 psig
Point C	25 psig
Point D	20 psig
Point E	15 psig
Expansion Tank	0 psig

3. Allowable pressure increase occurring when temperature reaches maximum design temperature

At all system points and at tank . . . . .25 psig

4. Maximum design temperature

At all system points . . . . .240°F

In expansion tank . . . . .120°F

NOTE: While temperatures will actually vary from a supply of 240°F to a return of approximately 200°F to 220°F, the effect of this difference on air solubility is negligible. Therefore, for clarity, we will assume 240°F at all system points. The temperature in the tank (120°F), when system points are at maximum design, is based on readings taken in actual operating systems in the field.

5. Amount of air in solution in city main water

2.0% of water volume

NOTE: This figure is based on actual checks of the water departments of five cities.

Determination of State of System Air at Various Points and Various Stages of System Operation

Using the illustrations on the following pages, we will examine closely the effect of pressure and temperature on:

1. The change in state to free air bubbles.
2. The formation of entrained air bubbles.
3. The capability of water to hold air in solution.

Our purpose will be to evaluate where and under what circumstances air in solution changes to air bubbles and vice versa. We can, by this means, understand the problems which arise from air in the system and can effectively correct these problems.

Analysis During Four Stages of System Operation

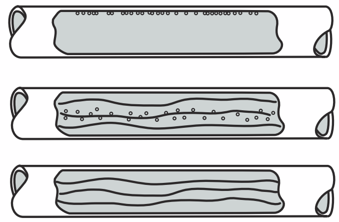
The analysis will be conducted throughout four stages covering the system's operating cycles. We have assumed that, during the analysis, the system will be shut down for energy conservation purposes and that the temperature will return to an ambient of 80°F.

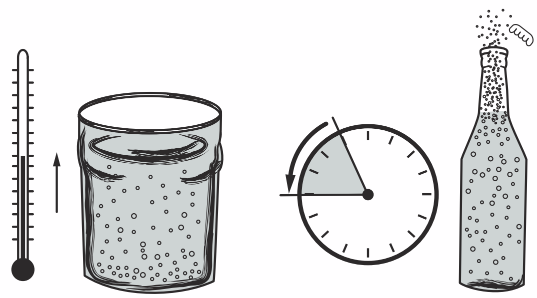
Stage I, Page C2-4. The effect of pressure and temperature on air solubility during the filling of the system and the initial raising of its temperature to design levels.

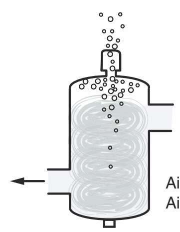
Stage II, Page C2-6. The partial loss of the air cushion in the plain steel expansion tank when the system returns to ambient temperature.

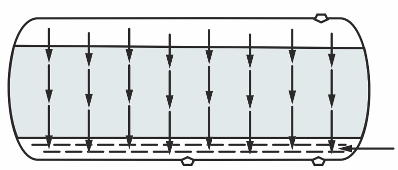
Stage III, Page C2-8. Two important points are stressed here: First – The rise in temperature in the heating boiler causes a certain amount of air in solution to change state and become entrained air bubbles which are efficiently removed by the air separating device. Second – A certain amount of dissolved air remains in solution and cannot be separated from the water.

Stage IV, Page C2-10. During each operating cycle, expanded water, leaving the tank, carries with it further portions of the air cushion in the tank. Finally, the tank is waterlogged.

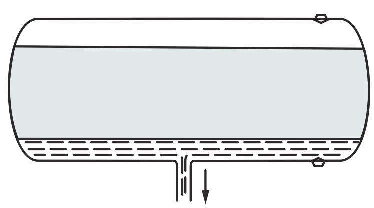








Air Cushion Lost at Ambient Temperature

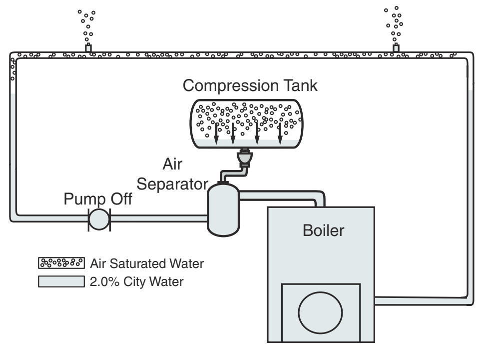


Expanded Water Carries Away Air Cushion

Stage 1

SYSTEM | **AT TANK**

1.



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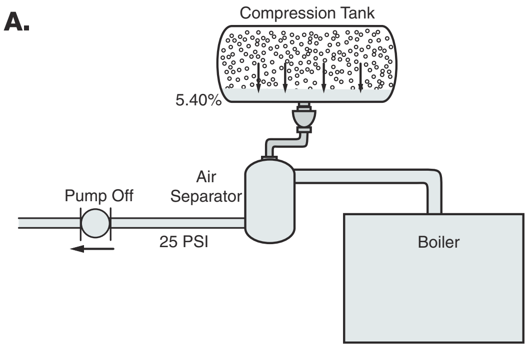
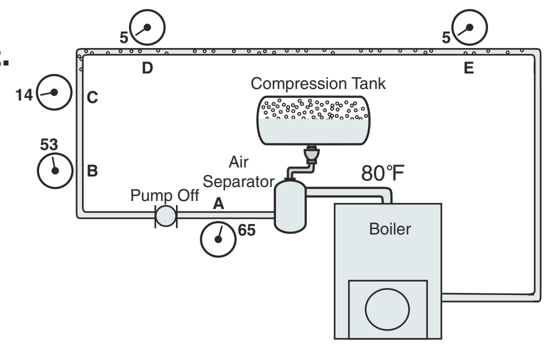


2.



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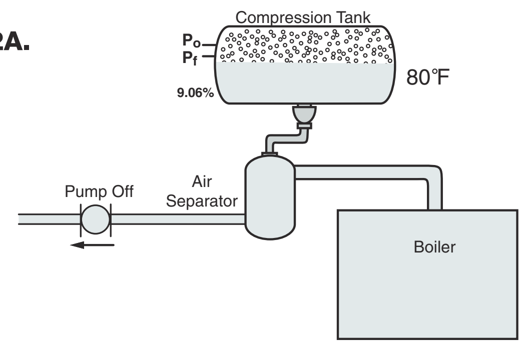

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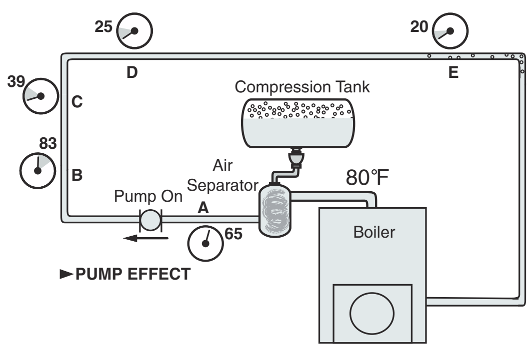

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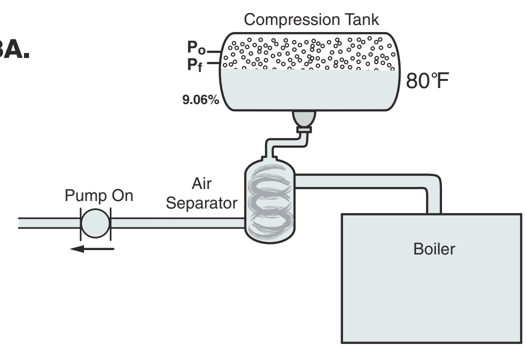

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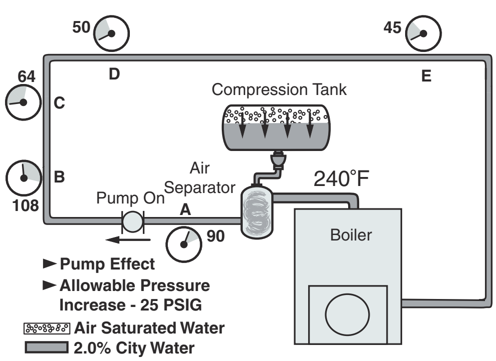

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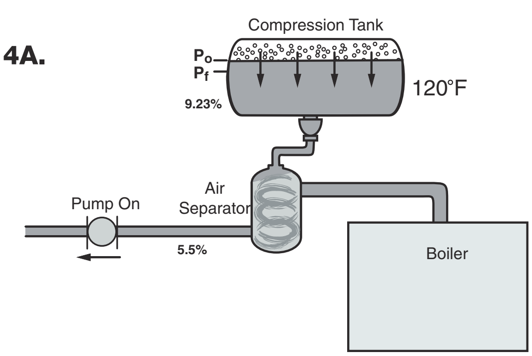


Stage I – The Effect of Pressure and Temperature on Air Solubility During an Initial 0peration

In the illustration opposite (nos. 1 and 1A through 4 and 4A), we will be emphasizing three different pressures exerted at different points in the system.

- The minimum operating pressure with the system at ambient temperature.

  The capability to hold absorbed air in solution is highest at the bottom of the system where the static pressure is highest.

- The additive pump effect with the system at ambient temperature.

  The increased pressure at all points (except adjacent to the tank connection) causes an increase in capability.

- The addition of the allowable pressure increase to the system due to the increase in temperature to the maximum design.

  At the bottom of the system, the temperature increase causes a decrease in percent capability. But at the top, this is offset by the pressure increase — and the net result is an increase in percent capability.

System

1. During the filling operation, as the water level rises in the vertical piping, the water at that point is close to atmospheric pressure and, at 60°F fill temperature, can hold only about 2.0% air in solution – some free air bubbles form – most of which will be removed with the air originally in the piping through manual venting.

2. With the system completely filled and at ambient 80°F temperature, the pressure at the top is 5 psig. We will assume that practically all air originally in the piping plus that released from city water by the decreased pressure at the top has been removed from the system by manual vents. The water at the bottom of the system still holds 2.0%.

3. The increased pressure due to pump operation increases the capability of the water to hold air in solution. Practically all air bubbles go into solution.

4. With the boiler in operation, the system temperature increases to the maximum design temperature. As the pressure rises to the maximum operating pressure, the capability of the water at the top to hold air in solution increases. No entrained free air bubbles exist in the system.

At Tank

1A. As static pressure increases, water enters the expansion tank compressing the air cushion.

At a pressure of 25 psig, the charge water has the capability of holding 5.4% of its volume as air in solution. Since the charge water from the city main has only 2.0% air in solution, 3.4% of its volume will be air absorbed from the air cushion in the tank.

2A. At the minimum operating pressure of 65 psig, water can hold up to 9.06% of its volume as air in solution. Since the charge water in the tank is in direct contact with air, an additional 7.06% of its volume will be absorbed from the air cushion.

3A. At the tank connection to the system, there is no additive pump effect, however, the static pressure is sufficient so that, in the air separator, no air bubbles can form and the 2.0% air in solution remains in solution and, therefore, cannot be separated. There is no change within the tank itself.

4A. At the bottom of the system adjacent to the tank connection, the increase in temperature causes the capability to drop from 9.06% to 5.5%.

But since the amount of air in solution in the piping is only 2.0%, no free air bubbles form, and air in solution cannot be separated.

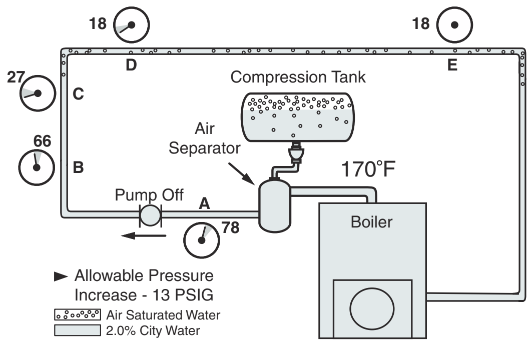
Since the water in the tank itself rises only slightly in temperature and the pressure is now maximum, the capability increases to 9.23% and more of the air cushion is absorbed.

Stage II

SYSTEM

AT TANK

5.



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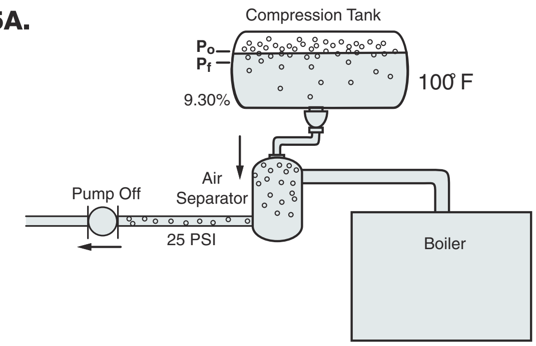

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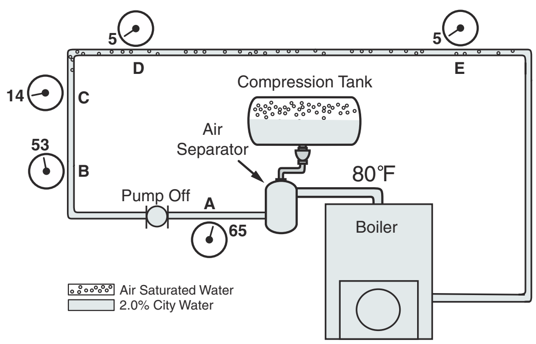

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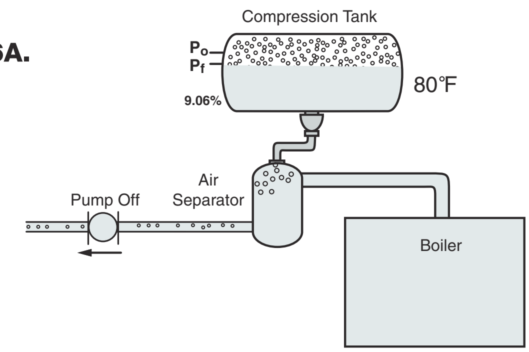

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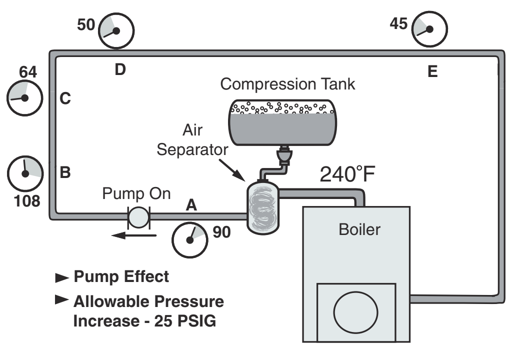

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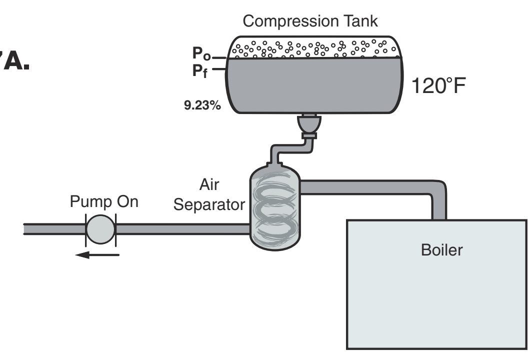

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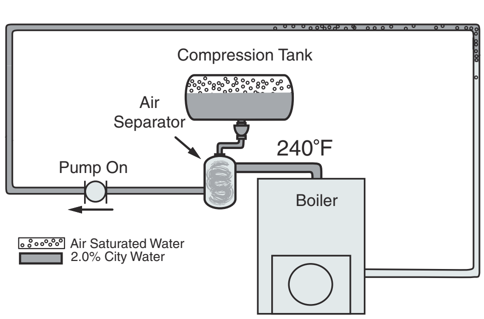

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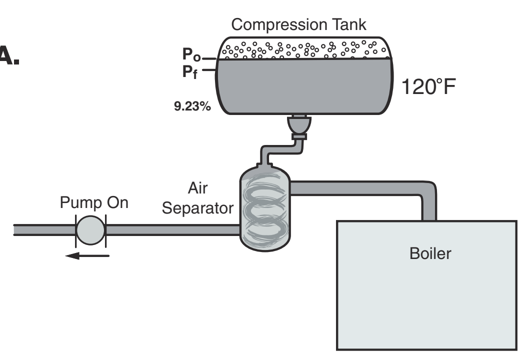


Stage II – Return of System Temperature to Ambient and Expanded Water to System and Return to Maximum Design Temperature.

We have seen, at this point in the operation of the system, that the initial filling of the system has not contributed any significant amount of free air to the system. We have also observed that any free air bubbles formed in the water near its surface during filling could be removed with the initial air in the piping through manual venting at all high points in the piping and components.

In the illustrations opposite (nos. 5 and 5A through 8 and 8A), we will see the formation of a "slug" of water flowing through system piping at ambient temperature containing an amount of air much higher than the 2.0% in other parts of the system.

System

5. With the boiler off, expanded water starts to leave the tank, system pressure decreases.

With the pump off, the additive pump pressure is removed. The drop in pressure causes a drop in the capability of water to hold air in solution. At the top, free air bubbles form as the capability drops below 2.0%.

6. When the system drops to ambient temperature, all expanded water has left the expansion tank and has formed an integral cylinder of water contained in the lowest part of the piping. At the top of the system, the water can hold only 2.40% air in solution.

7. With the boiler in operation and pump on, we will assume that our "slug," still at ambient temperature, has traveled without mixing with system water to the top of the system. As the static pressure decreases, entrained air bubbles form.

8. Entrained air bubbles are carried by the slug as it travels through the piping at the top of the system. When the slug travels down the piping, the static pressure increases and all entrained free air bubbles are reabsorbed.

At Tank

5A. Since the expanded water from the tank is colder than system water and therefore dense, it will flow into the piping at the bottom of the system. Because the water leaving the tank had been in direct contact with the air in the tank, it will hold in solution its full capability – 9.38% absorbed air.

As pressure constantly decreases, some of the absorbed air will form free bubbles in the piping to the air separator.

6A. For the purpose of analyzing the effect of pressure and temperature, we will assume that this cylinder of water at ambient temperature will contain approximately 9.0% air in solution and will not mix with system water as it travels throughout the system – we will refer to it as a "slug" of water.

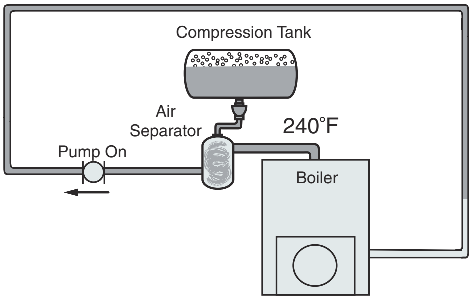
7A. The increase in system temperature has caused a new quantity of expanded water to enter the expansion tank.

8A. Since a portion of the air cushion is now carried in the piping system as absorbed air in solution, the water line in the expansion tank is higher than when the system was first put in operation.

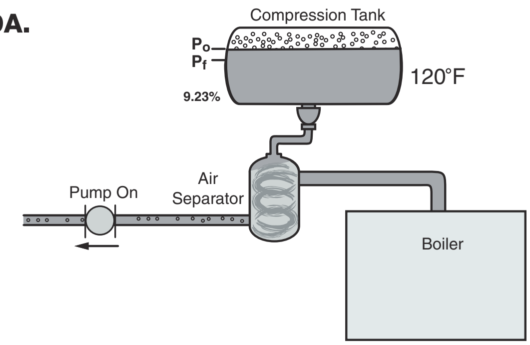
Stage III

SYSTEM | **AT TANK**

9



9A.



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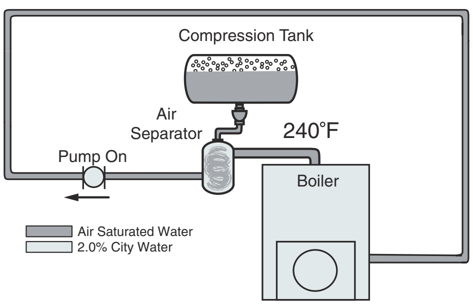
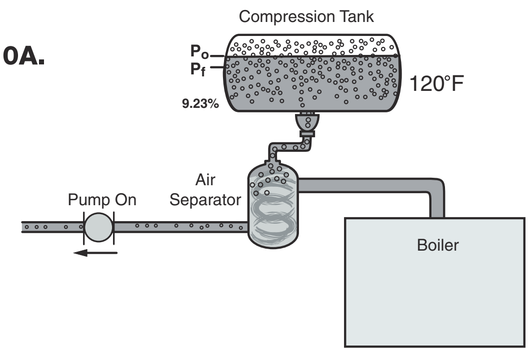


10A.



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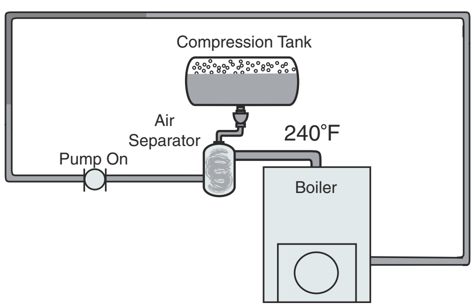
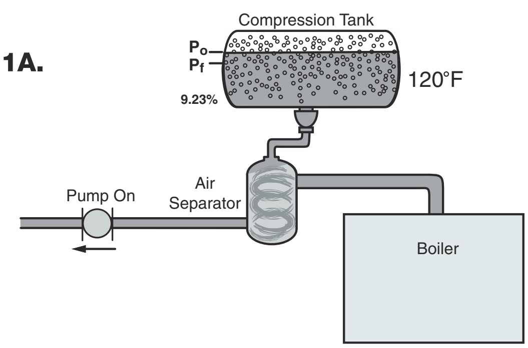


11A.



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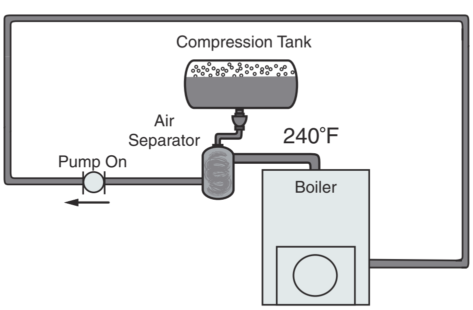
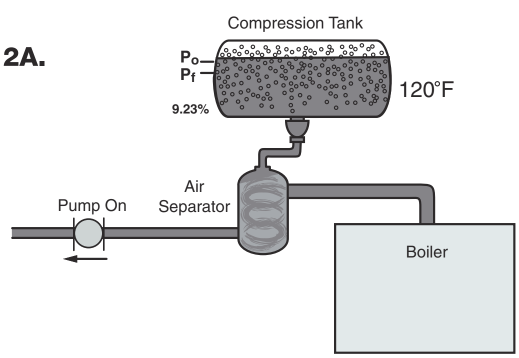


12A.



Stage III – System Temperature Raised to Maximum… “Slug” Goes Through Boiler

The illustrations nos. 9 and 9A through 12 and 12A show that when the slug of water containing a larger amount of air in solution flows through the boiler and the temperature increases, the capability to hold air in solution decreases.

The entrained air bubbles that form have been efficiently removed by the air separator and are returned to the tank where they may be returned to the air cushion. (If the actual amount of air contained by the charge water in the solution happened to be less than its capability – the air bubbles would be absorbed instead of remaining in a free state in the air cushion.)

However, because at the bottom of the system the pressure effect on the capability to hold air in solution is far more significant than temperature, most of the air remains in solution and cannot be separated.

System

9. We will assume that the slug will enter the boiler with approximately 9.0% air in solution and be brought up to temperature without mixing with other water in the boiler.

10. As the slug of water traveled through the boiler, the temperature increased from ambient to the maximum design temperature – the capability to hold air in solution dropped from 9.06% to 5.5%. Over 3.0% of its volume has become entrained air bubbles which are separated from the flowing water by the air separator.

11. As the slug of contaminated water containing approximately 5.5% air in solution reaches the top of the system, over 3.0% of its volume becomes entrained air bubbles.

12. As the slug of water flows downward through the piping and through the boiler, the increase in pressure causes all entrained free air to change phase back into solution…no air can be separated and returned to the tank.

At Tank

9A. The slug of water is still in the boiler. The system water flowing through the piping contains only 2.0% air in solution. This air in solution cannot be separated in the air separator.

10A. In the tank, the capability of the charge water to hold air in solution has increased to 9.23% as the system pressure increased to the maximum.

The free air bubbles released by the air separator are reabsorbed by the charge water before they can reach this air cushion.

11A. Noncontaminated system water containing only 2.0% air in solution flows through the air separator. No air is separated.

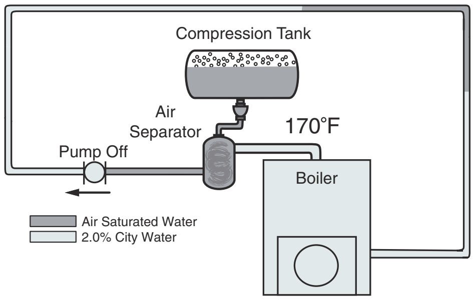
12A. The expanded water which entered the tank during the most recent heating cycles has now absorbed from the air cushion 9.23% of its volume air in solution.

Stage IV

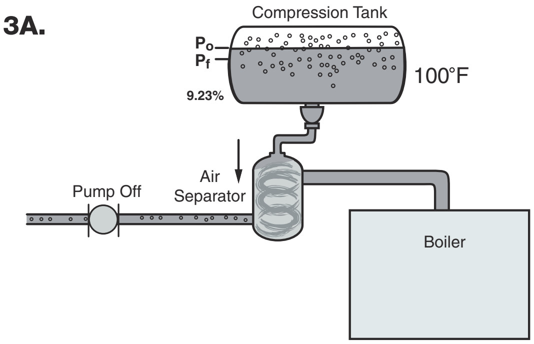
SYSTEM

AT TANK

13.



13A.



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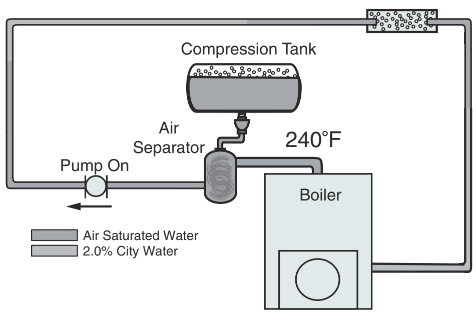

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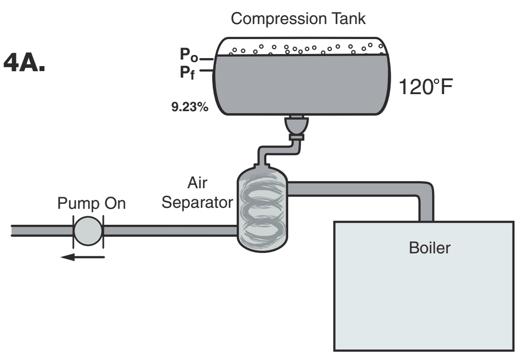

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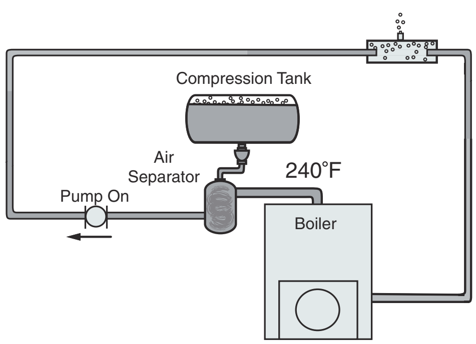
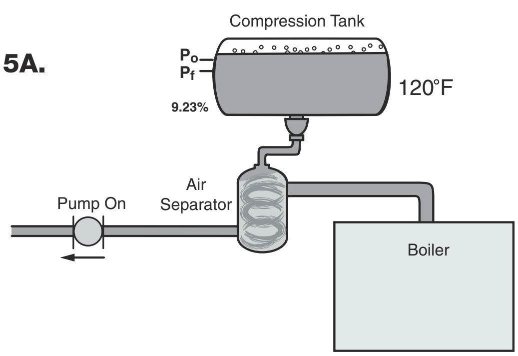


15A.



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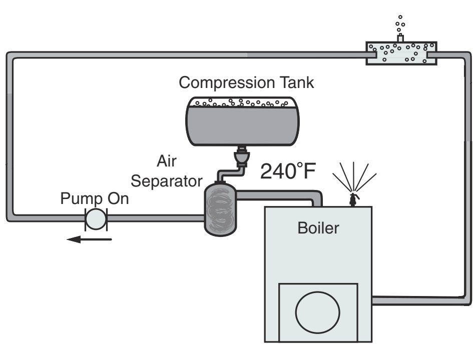
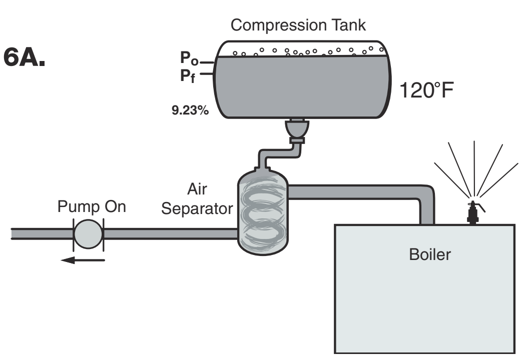


16A.



Stage IV – Loss of Air Cushion on Subsequent Cycles

As more air is removed from the air cushion in the tank, the tank will become waterlogged and must be drained and the air cushion replaced. The previous illustrations (nos. 13 and 13A through 16 and 16A) show this sequence of events.

System

13. At a point in the operating cycle when the pump is off and the system starts to cool, the water at the top of the system can hold only about 2.0% air in solution…3.5% of the volume of slug No. 1 will form free air bubbles.

14. More and more slugs are formed, each carrying entrained free air bubbles at the top – which change to air in solution at the bottom. When the free air bubbles accumulate in terminal units, blockage of flow and inefficient heating can occur.

15. To solve the problem of air accumulation in terminal units, an automatic air vent can be installed. Gradually, the mass of air in the piping system is reduced.

16. To replace the water constantly lost through the relief valve – the automatic fill valve will bring in new makeup water. The water line in the tank constantly rises until finally the tank is waterlogged.

At Tank

13A. When the system cools, expanded water again leaves the tank containing approximately 9.0% air in solution…losing over 3.0% back to the tank on its first trip through the boiler and air separator…carrying 5.5% air, either in the form of bubbles at the top – or air in solution at the bottom. Slug No. 2 has been launched.

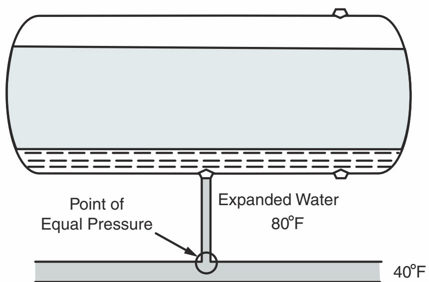
14A. During each cycle, expanded water from the system enters and then leaves the tank – initially containing 2.0% air in solution – eventually containing 5.5% air in solution – during each cycle removing 3.5% of its volume from the air cushion. As the air cushion is depleted, the water line in the tank rises.

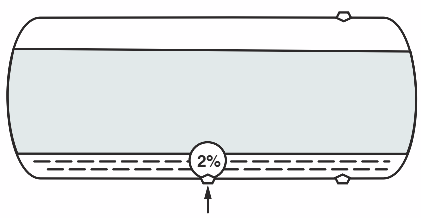
15A. As the mass of air in the piping system is reduced, more of the air cushion in the tank is transferred to the piping system. To compensate for the decrease in the minimum operating pressure (partial vacuum and boiling), an automatic fill valve allows makeup water to enter the system – the water line in the tank rises.

16A. The smaller air cushion in the tank can accommodate the expanded system water only at a pressure higher than the relief valve setting – the valve discharges water from the system.

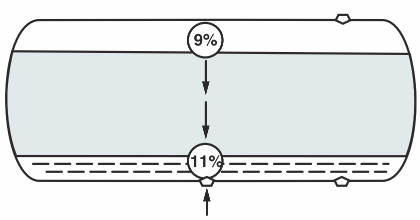
Chilled Water System Conditions and Air Solubility

Minimum Design Temperature . . . . .40°F
Ambient Temperature . . . . .80°F





At Ambient Temperature,
Water Expands into Tank
With 2.0% Air in Solution.



At Designed Temperature
Expanded Water Re-Enters
System With 11% of
Air in Solution.

In the chilled water system, expanded water enters the tank only during system shut-down, during a warm summer weekend, for example, when temperature increases from operating design to ambient temperature. As a result, the designed allowable pressure increase occurs. When the pump and chiller become operative, the system gradually cools and expanded water re-enters the system.

The temperature of the expanded water in the connecting line will be higher than the temperature in the system piping at the point of connection of the expansion tank.

As the pressures are equal at this point, the capability of the system water to hold air in solution will always be higher than the actual air held in solution by the expanded water as it enters the system.

In Chilled Water Systems, No Air Release Occurs at the Separator

In contrast to the heating system, no entrained air bubbles form, air cannot be separated in the air separator and cannot be returned to the air cushion in the tank.

If the example installation previously used for the heating system was a chilled system, operating at a minimum design temperature of 40°F and a high ambient temperature of 80°F during shut-down, expanded water would enter the tanks with 2.0% air in solution leaving with an average 11% and absorbing 9.0% of its volume from the air cushion in the tank.

Plain Steel Tank Sizing for Chilled Water Affected by Air Solubility

Critical sizing of a plain steel tank for chilled water, using an expansion factor based on 40°F to 90°F or 100°F and using Boyle's Law, would result in a relatively small size tank. However, since the resulting small volume of air cushion would be rapidly carried in solution from the tank to other parts of the system, general practice has been to use a much larger size tank with a large amount of air. This effectively prolongs the period of time before the tank must be drained and the air cushion re-established.

In either case, since significantly more air can be held in solution in system water, the entire system will eventually become saturated with highly corrosive air in solution.

ASHRAE Sizing Seemingly Reflects Concern About Air Solubility in Chilled Water Systems

In Chapter 15, ASHRAE Handbook and Product Directory (1976 Systems) the recommendation is made that plain steel expansion tanks for chilled water systems be sized to one half the equivalent hot water size, considering the piping system is operating from 70°F to 200°F.

Analysis Summary

1. An increase in temperature does cause a certain amount of air in solution to become entrained air bubbles which can be efficiently removed by an air separator device.

2. Pressure is more significant than temperature in determining when air will change from a dissolved state to gas bubbles.

3. As expanded water from the heating system left the expansion tank during each cycle, it transferred part of the air cushion to system piping; and when eventually this air accumulated at the top and was vented from the system, waterlogging of the tank occurred.

4. Expanded water leaving the plain steel expansion tank during each cycle of a chilled water system carries with it a part of the air cushion in the tank equal to the maximum capability of that water to hold air in solution.

As accumulated free air bubbles at the top of the system are vented, eventual waterlogging of the tank occurs.

5. The plain steel expansion tank will always contain water "saturated" with dissolved air up to its capability to hold air in solution.

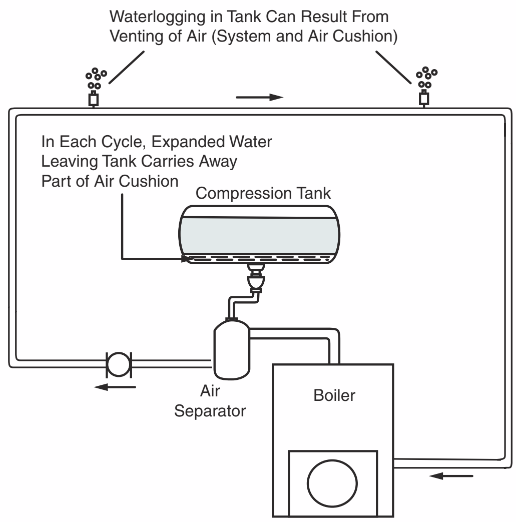
6. In initial filling, the piping system will contain dissolved air equal only to that contained in the city water main. This is usually much less than the water in system piping is capable of holding, so "free air" released will be negligible.

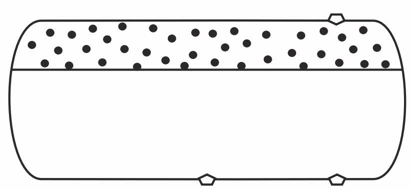
7. The major source of "system air" in a closed hot water heating or chilled water system is the plain steel tank.

8. Air in the system piping is constantly changing from a dissolved state to free and entrained air bubbles throughout the system. Air in a closed heating or chilled water system is far too elusive to be controlled.

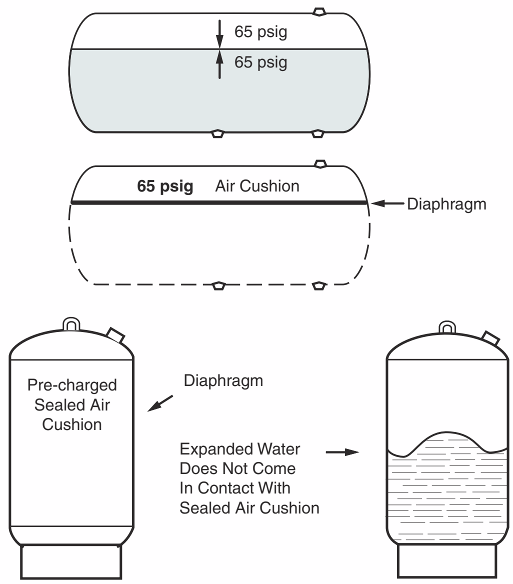
Diaphragm-Type or Bladder-Type Expansion Tanks Do Not Require "Air Control"

The only reason for water in the plain steel expansion tank during initial fill of the system is to compress the air to the minimum operating pressure. The volume of this charge water is determined by the volume of the air cushion required for a given installation (See Chapter 1, Section B.)





Plain Steel Tank – Major Source of "System Air"



If we were to place a diaphragm in the plain steel tank, separating the charge water and the properly sized air cushion and pre-pressurized that volume of air, we would not need the charge water.

We, then, would have a diaphragm-type expansion tank, precharged to the minimum operating pressure and, more importantly, with an air cushion sealed from contact with system water.

The properly sized sealed air cushion in a diaphragm-type tank is not in contact with water in the system and, therefore, has no relation to the "system air" which is the source of so many problems.

We can now drop any pretense of "controlling" system air – an exercise which the solubility tables prove is not practical. We can now consider elimination of system air.

Section C

DEFINITIONS, ANALYSIS AND CORRECTIVE PROCEDURES REGARDING THE PROBLEM OF "SYSTEM AIR" IN CLOSED HOT WATER AND CHILLED WATER HVAC SYSTEMS

Chapter Three

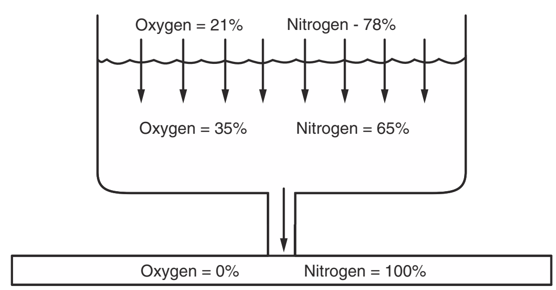
AIR ELIMINATION SOLVES THE PROBLEM OF OXYGEN CORROSION

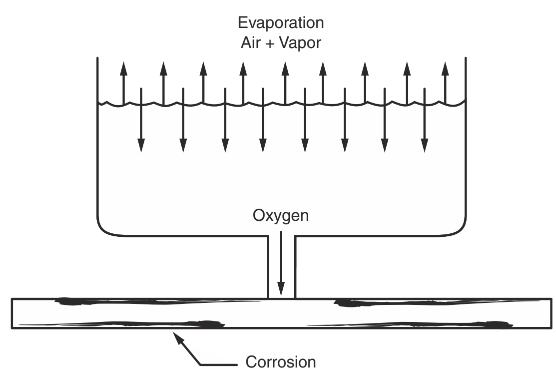
Oxygen Corrosion Potential is Extremely Serious with an Open Tank (Roof Top).

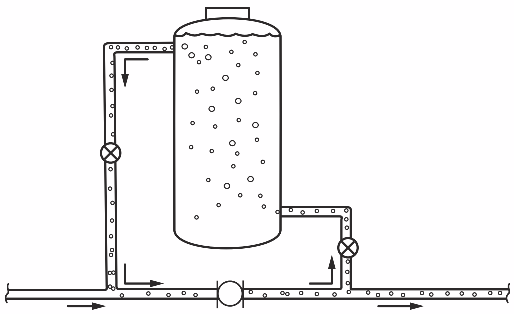
The open tank, or roof top, has several disadvantages over the closed tank (whether plain steel or diaphragm-type). In view of critical energy concern, the constant loss of vapor from the surface of the water (heating system) has become increasingly serious. The water loss through evaporation must be replaced by makeup water carrying more oxygen. In some installations, measures must be taken to prevent freezing – a further energy loss. Constant exposure to the atmosphere results in the introduction and concentration of chemicals, dirt and dust which cause sludge in system piping. Suspended solids cause erosion in piping, heat exchangers and valves. In spite of chemical treatment, deposits of dirt at the bottom of piping cause localized pitting.

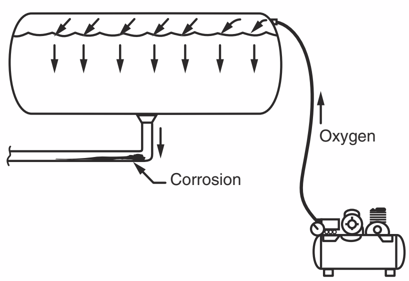
The most serious potential problem is oxygen corrosion. The water in the open tank is at all times in contact with air from the atmosphere. The air in the upper part of the tank is made up of 78% nitrogen and 21% oxygen. The absorbed air, in solution, in the water in the lower portion of the tank, is 65% nitrogen and 35% oxygen. Water has the characteristic of absorbing oxygen more readily than nitrogen. Not only the water at the surface, but also the entire amount of water in the tank must become saturated with oxygen.

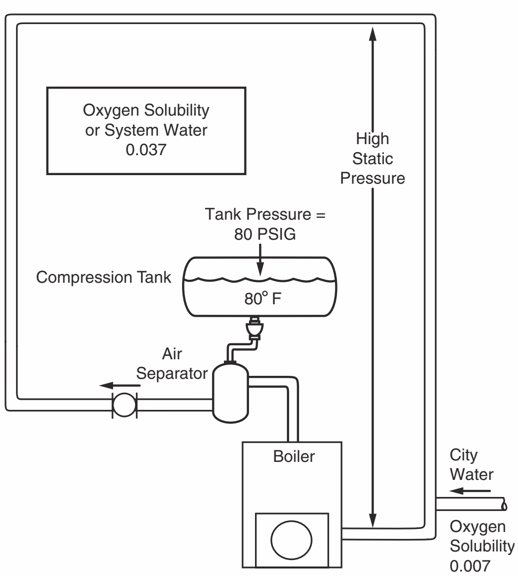
The oxygen in solution in system water is unstable, and unites with the metal components to form rust and corrosion. The amount of oxygen in the water in system piping is therefore always less than the amount that the water in system piping is capable of holding – in a sense, the water is oxygen "starved." In contrast, the water in the open tank, in direct contact with air from the atmosphere, is at, or close to, the saturation point. As long as the water in system piping contains less oxygen than its maximum capability, oxygen will migrate from the water in the tank to water in the piping by diffusion. The process of oxidation guarantees that water in system piping will always contain less than its maximum capability, therefore, the process of oxygen diffusion from the open tank to the system will be constant. This will occur even if there were no movement of water between tank and piping.











The constant movement of expanded water between tank and piping as system temperature fluctuates…expedites the migration of oxygen throughout the system. A drop in temperature results in contraction of the water volume and therefore flow of tank water, saturated with oxygen, into the system. An increase in temperature forces expanded water, always less than saturated with oxygen, into the tank. During each operating cycle, a quantity of oxygen is absorbed from the air and deposited in the steel piping in the form of rust – an oxygen pump.

Chemical treatment to reduce oxygen corrosion is essential in order to minimize maintenance and replacement problems. The anticipated cost of chemical treatment equipment and monitoring devices plus the cost of maintenance and replacement components should be made available to the owner in the evaluation of the life cycle cost for the building.

Oxygen Corrosion Potential with a Plain Steel Tank ("Closed" System) Would Seem to Be Less Than with an Open Tank

The plain steel tank in a "closed" system seemingly has an advantage over the open tank, in that the amount of oxygen available to the open tank is unlimited, whereas the amount of oxygen available to the plain steel tank is the amount in the tank at initial fill. However, the number of times that the tank must be re-charged with air, or in the case of the addition of an air compressor, provides, in essence, an open conduit to the atmosphere…truly an "oxygen pump".

The plain steel tank installed at the bottom of the system, where the static pressure could be quite high and has the capability of holding a high amount of oxygen in solution. For example, in a heating system operating with a tank pressure of 80 psig and a temperature in the tank of 80°F, the oxygen solubility is 0.037, compared to the oxygen solubility in city water of roughly 0.007.

Just as the water flowing in system piping will absorb oxygen from the water in the open tank, it will absorb oxygen from the water in the closed tank. Whether or not there were actually movement of water from the tank or oxygen will migrate through the system, causing corrosion of piping and components.

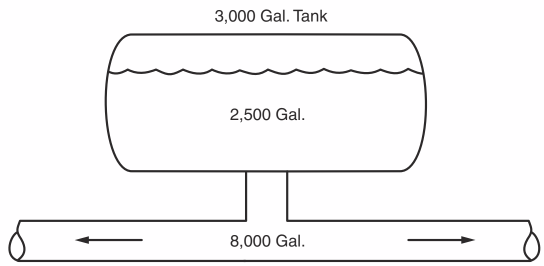
Chilled Water Systems, at Lower Temperature, Have Higher Oxygen Solubility

In a chilled water system, the water flowing through the piping is at a much lower temperature and consequently can carry a much greater amount of oxygen. The solubility of water at 80 psig and 50°F is 0.0514 — the oxygen solubility of city water, 0.007. Makeup water is often cited as a source of corrosion, yet for every gallon of makeup water, only 0.007 gallons of oxygen enter the system. Every gallon of air originally in the plain steel expansion tank contained 0.21 gallons of oxygen — a ration of 1 to 30. For every gallon of air originally in the plain steel tank, 30 gallons of makeup water would be required to have the same corrosive effect. For example, chilled water system with a content of 8,000 gallons, a 3,000 gallon plain steel tank is installed. The 8,000 gallons of water from the city main required to fill the piping system and components contain 56 gallons of oxygen. The 2,500 gallons of water required to charge the tank to 80 psig minimum operating pressure contains 17.5 gallons of oxygen. The air originally in the tank contains 630 gallons of oxygen. Total oxygen of 703.5 gallons. 100,000 gallons of water would be required to have the same corrosive effects, a loss and makeup twelve times the system piping volume. It would appear that a reasonable amount of makeup water is not significant compared to the oxygen added to the system by draining and refilling a plain steel tank or charging with compressed air. The “closed” tank with a compressor is similar to an open tank in that it creates an efficient “oxygen pump.” In reality, the system is no longer “closed” but is “open.”

For years, engineers have agreed, that in a closed system with a plain steel tank (theoretically), there should be no corrosion problem. Yet, on countless installations, the abrasive action of the black iron oxide, Fe_3O_4, caused failure of mechanical pump seals and valves. Rust deposits formed in piping and components, reducing circulation and causing inefficient operation and wasted energy. In extreme cases, complete blockage of piping has occurred, necessitating reaming of the pipe or replacement.

The Problems Resulting from Chemical Treatment Are Sometimes as Troublesome as Corrosion

Because a “closed” system often becomes an “open” system, chemical treatment has become more common. But the problems resulting from chemical treatment are sometimes as troublesome as the original corrosion due to oxygen. Too small an amount of one chemical could cause pitting. Excessive amounts added intermittently cause problems which could be avoided by constant feeding based on monitored results. Automatic chemical feeders are costly, and many times introduce more oxygen into the system. Standard materials used for pump seals fail when exposed to high concentrations of certain chemicals. Even a low percentage of chromate in system water can cause seal failures. Absorbed in system water, it travels across the mechanical seal face, from the system to the atmosphere. When the water evaporates into the atmosphere, the chromate remains as an abrasive solid, causing seal wear and eventual failure.



8,000 Gal. – System Water

8,000 x 0.21 ÷ 30 = 56 Gal. Oxygen

2,500 Gal. – Tank Charge Water

2,500 x 0.21 ÷ 30 = 17.5 Gal. Oxygen

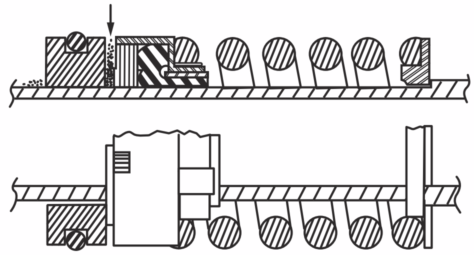
3,000 Gal. – Total Tank Volume

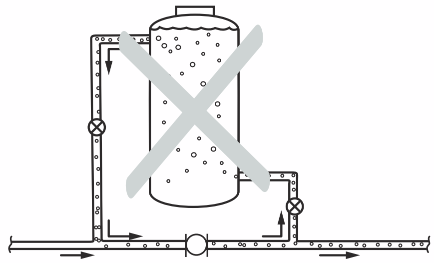
3,000 x 0.21 = 630 Gal. Oxygen

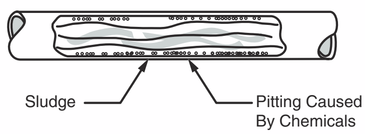
630 + 56 + 17.5 = 703.5 Gal. Total Oxygen

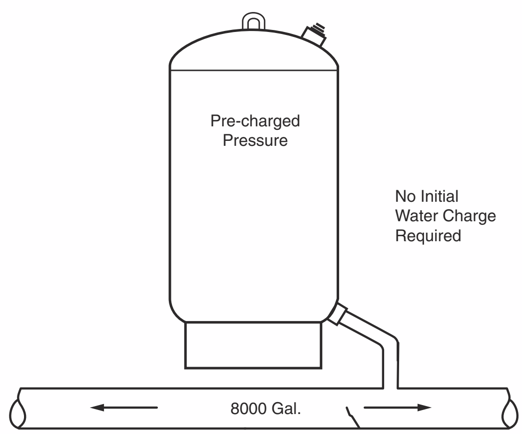
Chemical Imbalance Causes Problems in Chemical Treatment

Pump Seal Failure Caused By Abrasive Solid Chromate









8,000 Gal. - Volume System Water
8,000 x 0.21÷30 = 56 Gal. Oxygen

In some instances, the addition of chemicals causes sludge and frequent boiler blowdown is required. More makeup must be added, carrying more oxygen into the system. Continued dumping of pervasive toxic waste into public sewer systems or streams is a questionable procedure in view of public concern over safety hazards. Hazardous waste is a constant threat to drinking water. Pollution controls and regulations in the local community should always be consulted. In summary, if chemical treatment of system water must be accomplished, the proper equipment must be installed and constantly monitored by trained, responsible maintenance personnel. Observation of many installations reveals that chemical treatment is inconsistent and poorly monitored. The procedures followed by so-called specialists appear rather mystical to many engineers involved in maintenance of HVAC systems – specifications covering chemical treatment vary widely and are many times confusing. The responsibility for control of potential corrosion problems is rarely clearly placed – engineer, contractor, chemical “specialist” and manufacturer, all are involved to varying degrees. If a specification reads that chemical treatment should be “sufficient and proper,” who determines what is “sufficient and proper”? Either overlapping responsibility or lack of responsibility is the result of this confusion. In the final analysis, it is the owner who bears the financial burden for the corrosion problems which occur.

Air Elimination Solves the Problem of Oxygen Corrosion

The diaphragm-type or bladder-type tank offers a better solution to the problem of corrosion. Because the required size air cushion is permanently sealed in, substantially all of the other air in the system can be eliminated. In the example chilled water system mentioned previously, a total of 703.5 gallons of oxygen existed in the system at startup: — 630 gallons in the air originally in the tank; 17.5 gallons in the charge water in the tank; and 56 gallons in the water in system piping and components. The use of a diaphragm tank eliminates the first two, leaving a total of 56 gallons of oxygen, a relatively insignificant amount. By the use of air elimination devices installed at the proper location, and following air elimination procedures at initial system startup (see Chapter Five), most of the 56 gallons of oxygen can be removed.

With reasonable care, the addition of makeup water can be minimized. Therefore, no opportunity exists for any significant changes in the composition of the water in the system. With proper pH control, and except in areas with abnormal water conditions, no chemicals need be added to the water heating and chilled water system.

Section C

Definitions, Analysis and Corrective Procedures Regarding the Problem of "System Air" in Closed Hot Water and Chilled Water HVAC Systems.

Chapter Four

System Air (Nitrogen) and Its Impact on the Design and Operation of the HVAC System

System Air (Nitrogen) and Its Impact on the Design and Operation of the HVAC System

When reference is made to "air," it is essentially a reference to nitrogen, since the oxygen in the expansion tank and in the system water will ultimately be depleted by oxidizing with metal components, particularly with iron to form rust (see Chapter 3), and, thus no longer will be present as a gas.

Fundamental in proper system design is the location of a single stable "point of no pressure change." This location determines the hydraulic stability and controls the operating pressure range to desired limits throughout the operating cycle. Yet, the presence of nitrogen in gaseous form at upper elevations destroys the stability of the "point of no pressure change." It can no longer be used as a reference point, and therefore makes it impossible to predict hydraulic performance.

Gas bubbles in system piping result in blocked terminal units and difficulty in balancing system flow, noise and inefficient pump performance. It is known that free gas does exist in piping and that generally ineffectual attempts are made to "control" it. High system water velocities are maintained in an effort to "purge" bubbles. The reduction of system flow rates to conserve pumping power is seldom practical.

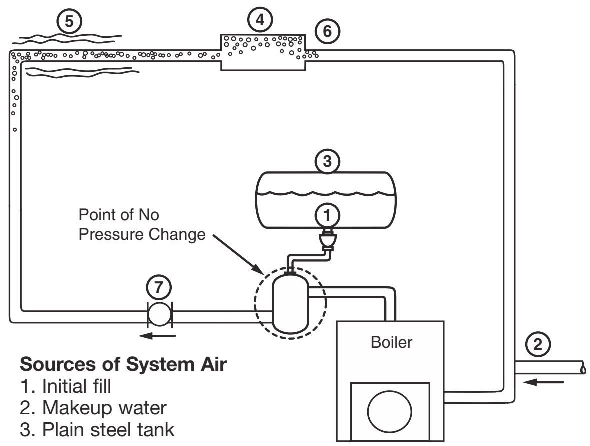
The Presence of Nitrogen as a Gas or Absorbed in System Water Is Determined by Physical Laws

Henry's Law is expressed by the equation: $X = \frac{P}{H}$ where X is the amount of gas in solution, P is the partial pressure of that gas and H is a constant which changes with temperature. For gases oxygen and nitrogen, Henry's Law is expressed:

$$X_O = \frac{P_O}{H_O} \qquad X_N = \frac{P_N}{H_N}$$

The constants H_O and H_N vary with temperature.

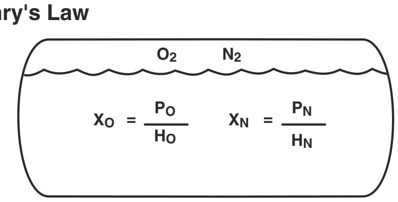
Dalton's Law states that "the total pressure exerted by a gas is equal to the sum of its partial pressures" and is expressed: $P_t = P_O + P_N + P_V$ where the subscripts O, N and V refer to oxygen, nitrogen and water vapor. The water vapor is a function of the temperature of the water, and is independent of the total pressure.



Sources of System Air
1. Initial fill
2. Makeup water
3. Plain steel tank

Problems Caused by System Air
4. Blocked terminals
5. Noise in piping
6. Blockage of system circuit
7. Inefficient pump operation

Henry's Law



Dalton's Law

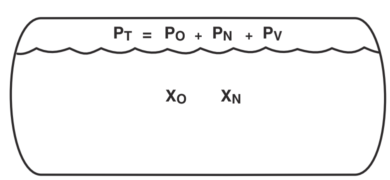


Table 13 – Solubility of nitrogen in water

Expressed as a fraction of water volume, at STP

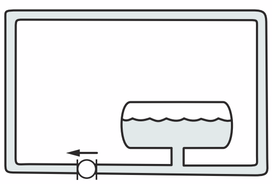
Table 13A Nitrogen – 78% of dry gas volume

Temp (t) °F	Total Pressure, PSIG										
	0	10	20	30	40	50	60	70	80	90	100
40°	0.0166	0.0279	0.0393	0.0506	0.0620	0.0734	0.0847	0.0961	0.1074	0.1188	0.1302
50°	0.0145	0.0245	0.0346	0.0446	0.0546	0.0646	0.0746	0.0846	0.0946	0.1046	0.1146
60°	0.0130	0.0219	0.0303	0.0399	0.0488	0.0578	0.0668	0.0757	0.0847	0.0937	0.1026
70°	0.0117	0.0199	0.0280	0.0362	0.0443	0.0525	0.0607	0.0688	0.0770	0.0851	0.0933
80°	0.0107	0.0182	0.0258	0.0333	0.0405	0.0484	0.0559	0.0634	0.0709	0.0785	0.0860
90°	0.0098	0.0168	0.0238	0.0308	0.0375	0.0445	0.0518	0.0587	0.0657	0.0727	0.0797

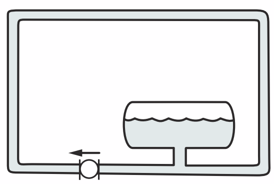
Table 13B Nitrogen – 100% of dry gas volume

Temp (t) °F	Total Pressure, PSIG										
	0	10	20	30	40	50	60	70	80	90	100
40°	0.0210	0.0353	0.0497	0.0641	0.0785	0.0929	0.1072	0.1216	0.1360	0.1504	0.1648
50°	0.0184	0.0311	0.0437	0.0564	0.0691	0.0818	0.0944	0.1071	0.1198	0.1324	0.1451
60°	0.0164	0.0278	0.0391	0.0505	0.0618	0.0732	0.0845	0.0959	0.1072	0.1186	0.1299
70°	0.0148	0.0251	0.0355	0.0458	0.0561	0.0665	0.0768	0.0871	0.0974	0.1078	0.1181
80°	0.0135	0.0231	0.0326	0.0421	0.0517	0.0612	0.0707	0.0803	0.0898	0.0993	0.1089
90°	0.0124	0.0212	0.0301	0.0390	0.0478	0.0567	0.0655	0.0744	0.0832	0.0921	0.1009
100°	0.0114	0.0197	0.0279	0.0362	0.0445	0.0528	0.0611	0.0693	0.0776	0.0859	0.0942
110°	0.0105	0.0183	0.0261	0.0339	0.0417	0.0495	0.0574	0.0652	0.0703	0.0808	0.0886

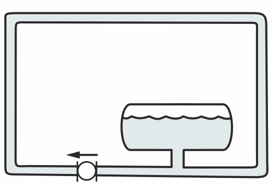
Nitrogen Migration in System



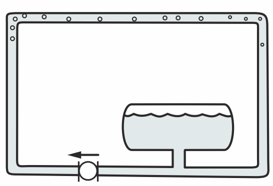
1. Tank and System is Clear.



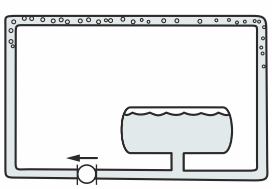
2. Diffusion Begins in Tank.



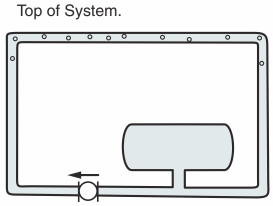
3. As Diffusion Continues Only in Tank, Water Level Rises.



4. Diffusion Continues in Tank and Water Level Rises Higher. System Becomes Slightly Saturated. Small Amount of Bubbles Form at Top of System.



5. System Completely Saturated. Water Level Keeps Rising to Point of Becoming Waterlogged. Large Amount of bubbles Form at Top of System.



6. Tank is Completely Saturated and Water-logged. System is Only Slightly Saturated.

The solubility of nitrogen in water which is in contact with air, 21% oxygen and 78% nitrogen (such as water in the city reservoir), is shown in Table 13A, page REF-14 Reference Section. The solubility of nitrogen in water which is in contact with nitrogen gas, 100% dry gas volume and water vapor (such as in a plain steel expansion tank) is shown in Table 13B, page REF-14 Reference Section.

In an example heating system, the charge water in a plain steel tank is in direct contact with the nitrogen gas cushion, assuming oxygen has been depleted through oxidation. At an operating pressure of 90 psig and an estimated temperature of 100°F, the solubility of nitrogen is 0.0859 (see Table 13B), which means that every 100 gallons of water in the tank can hold up to 8.6 gallons of nitrogen in solution (measured at standard temperature, 32°F, and standard pressure, 14.7 psia).

The amount of nitrogen in the water which initially filled the system probably came from the city reservoir where it was in direct contact with air containing 21% oxygen and 78% nitrogen (see Table 13A, Ref-14). At this condition, and a typical 50°F temperature, every 100 gallons entering the expansion tank, at 0.0145 nitrogen solubility (Table 13A), would absorb up to 7.1 gallons (STP) of nitrogen from the "air" cushion. The actual volume at 90 psig and 100°F is calculated to be 1.13 gallons.

Nitrogen Will Migrate from the "Air Cushion" to All Locations in the System

Henry's Law tells us that as long as system water flowing at the point of tank connection to the system contains less absorbed nitrogen than its maximum solubility at the water temperature and pressure, nitrogen will migrate from the tank to the water in the piping (unless the tank becomes waterlogged first). The use of an air compressor will ensure a fresh supply of nitrogen, in addition to oxygen, and will form an effective "nitrogen pump." According to Henry's Law, this migration occurs solely by diffusion over an extended period of time, even if there were no movement of water between the tank and piping.

In actuality, however, movement does occur and greatly accelerates the diffusion process as temperature changes in the system. If temperature fluctuation is within a narrow range, the movement of expanded water, and migration of nitrogen, into the system occurs less rapidly than when the temperature fluctuation is greater and operating cycles more frequent. Therefore, whether system water is at the maximum solubility level depends upon system temperatures, operating cycle frequency, pressures, system volume, tank volume, the length of time that the system has been in operation and other variables. For the purpose of example, the assumption is made that system water is at the solubility point, nitrogen in water.

Henry's Law also tells us that when system water is saturated with nitrogen at the point of tank connection, any pressure reduction (at constant temperature) will force nitrogen out of solution and cause gas bubbles to form. In a typical installation, system water flows through an air separator which effectively separates a large portion of any entrained nitrogen gas bubbles (but not absorbed nitrogen in solution) and directs them to a plain steel expansion tank. The presence of any bubbles is proof that the system water is at, or close to, the maximum solubility level. A reduction in pressure occurs as the water flows through the exit nozzle of the air separator. Normally, this reduction in pressure would be minimal, but if baffles or a strainer are installed adjacent to the exit nozzle, the pressure drop could be significant and the rate of bubble formation much greater. As system water flows through piping and valves, friction head loss causes a further pressure decrease.

Boyle's Law states that as pressure decreases – volume must increase – the bubbles, which start out small, grow larger.

Henry's Law states that as pressure decreases – the amount of gas coming out of solution increases – the addition of more gas increases the number of bubbles.

Dalton's Law states that as pressure decreases, water vapor must form within the gas bubbles, amplifying their size.

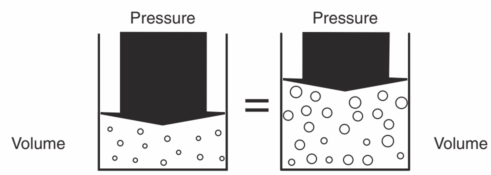
Pump Performance Is Affected by Nitrogen Coming Out of Solution

The pressure decrease between the suction nozzle of the system pump and the eye of the impeller, can cause a drastic expansion of bubble size. In extreme cases pump cavitation and noise can take place. In almost all installations, depending on static pressure and temperature, a decrease in pump head and therefore a loss of efficiency can occur. In extreme cases, pump performance ceases altogether. The phenomenon described above can occur even when the available net positive suction head is more than that required, as specified in the pump manufacturer's rating curves.

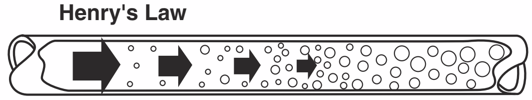
Unpredictable Hydraulic Performance Results from Nitrogen Bubbles in System Piping and Components

The reduction in static pressure in piping and components at higher elevations in the system can cause large amounts of gas bubbles to form, causing blocked terminal units, difficulty in balancing system flow, noise, inefficient pump performance, loss of the "point of no pressure change," and unstable and unpredictable hydraulic performance.

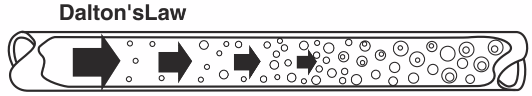
Boyle's Law

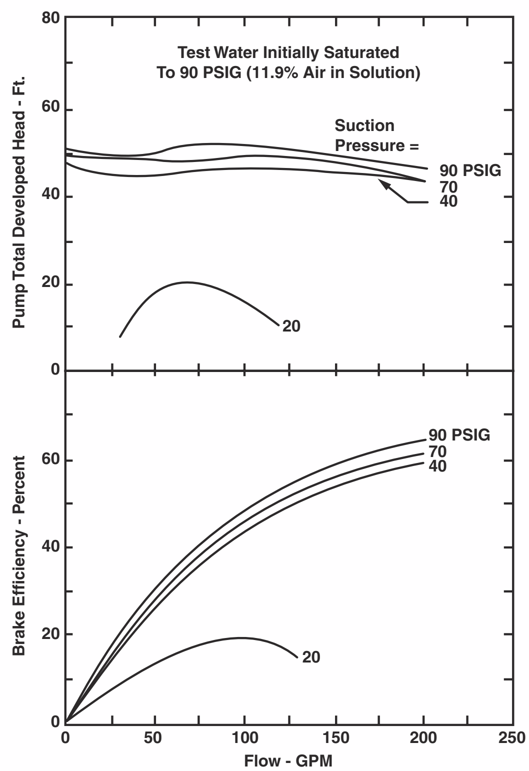


Henry's Law

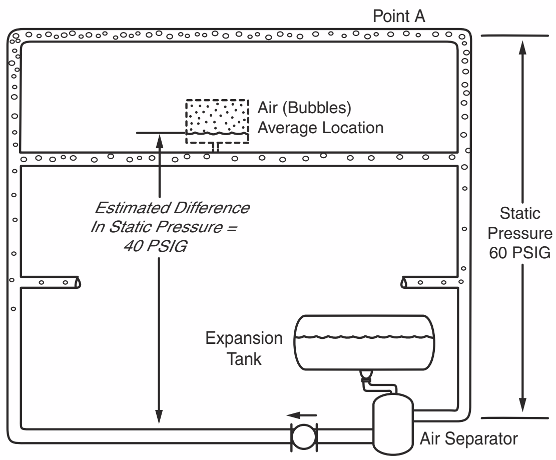


Dalton'sLaw

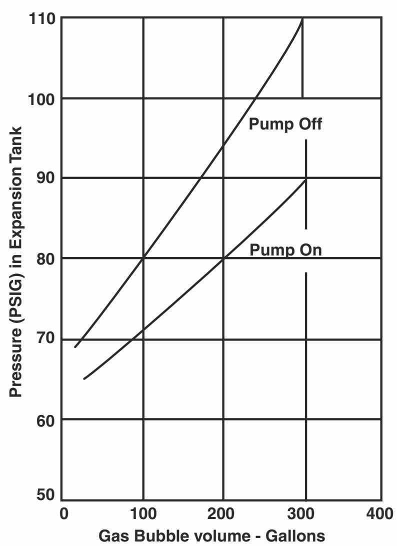




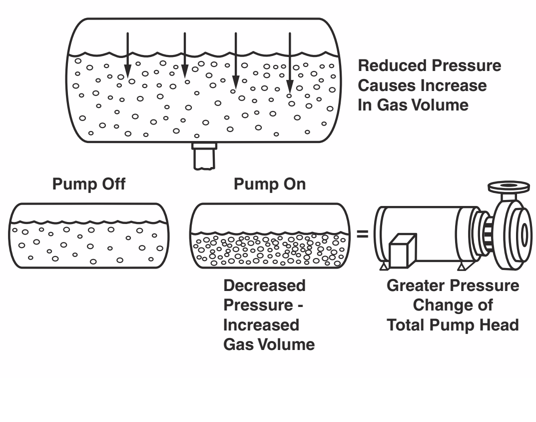
Effect of Air Coming Out of Solution on Pump Performance



Example Heating System



Effect of Bubbles in System on Expansion Tank Pressure



When a portion of the gas cushion originally in the expansion tank, typically at the bottom of the system, migrates to the top of the system, the volume of that gas will increase due to the reduction in static pressure and the formation of water vapor. The water displaced from the top of the system is forced into the expansion tank, compressing the gas remaining there, increasing pressure throughout the system. The increase in system pressure would be partially or wholly offset by the decrease in system pressure due to the loss of oxygen by oxidation and the absorption of nitrogen gas by system water.

At a given temperature, the total volume of the gas in the expansion tank and in the "second air cushion" is constant, and therefore the capability of the combined gas cushions to accommodate increased system volume due to temperature increase remains unchanged. If gas from the "second air cushion" is vented to unblock terminal units, the decrease in the mass of nitrogen in the system will cause system pressure to eventually drop below the minimum operating pressure and an automatic fill valve would allow makeup water to enter the system. The increase in system water would be accompanied by a decrease in the total gas volume at minimum operating pressure to the point that it would be incapable of accommodating expanded water without the maximum operating pressure exceeding the relief valve setting. The tank would be waterlogged.

In an example installation, the effect of a specific, unvented volume of gas at upper elevations in the system (the "second air cushion") on the pressure at the expansion tank can be calculated. From this data, a curve has been developed which shows the gas bubble relationship to pressure, both with the pump operating and not operating.

Disappearance of the "Point of No Pressure Change"

The operation of the pump causes a decrease in pressure at the expansion tank and an increase in pressure at the "second air cushion" at the top of the system. The reduction in pressure at the expansion tank is accompanied by an increase in gas volume as water is withdrawn. The volume of the water displaced from the tank will equal the volume of water displaced to the piping at the top of the system. The larger the volume of the "second air cushion" relative to the volume of the gas remaining in the expansion tank, the greater will be the pressure change relative to the total pump head. If all the gas in the system were located at the top of the system and not in the tank, the decrease in pressure at the pump suction could be approximately half the total pump head. The existence of a "second air cushion" at higher elevations would have destroyed the known point of "no pressure change" due to pump operation and eliminated any possibility of stable hydraulic performance for the system.

Observation of a pressure gauge located at the pump suction in actual installations, with the pump operating and not operating, and comparison with the pump head will indicate not only the presence of "air" at upper elevations, but also the amount, relative to that remaining in the expansion tank.

The Pressure and Entrained Gas Volume at a Specific Critical Location Can Be Estimated

With an understanding of the effect of the total amount of gas bubbles on system pressure, the pressure and the entrained gas bubble volume at a specific critical location can be determined.

In an example system (illustrated), the pressure at the top of a vertical return (Point A) is estimated to be 10 psig. At a temperature of 200°F, the nitrogen solubility is 0.0085 (see page REF-14 Table 13B, Reference Section). The nitrogen solubility of system water flowing through the piping and air separation device adjacent to the plain steel expansion tank at 70 psig and 220°F is 0.0455 (210°F in Table 13B can be used since Henry's constant tends to peak in volume in the neighborhood of 200°F to 210°) The solubility change is 0.037. For every 100 gallons of flowing system water at Point A, 3.7 gallons of nitrogen gas would be released as entrained bubbles. Converted from STP, the actual volume would be 3.0 gallons.

However, at higher temperatures, the effect of vapor pressure on the volume of gas bubbles must also be considered. According to Henry's Law, the amount of nitrogen dissolved in system water at a given temperature is directly proportional to the partial pressure of the gas, and therefore, the decrease in pressure determines the amount of nitrogen released as a gas. The partial pressure of the water vapor (vapor pressure) depends solely upon temperature. From Dalton's Law, the total pressure on the gas bubble is equal to the sum of the partial pressure of the nitrogen and the partial pressure of the water vapor (vapor pressure). When the total pressure is only slightly higher than the vapor pressure, the proportion of vapor to nitrogen in the gas bubbles is very high - the bubble growth accelerates.

In the example system, the effect of vapor pressure is to increase the volume of the gas bubbles from 3.0 gallons to 5.6 gallons. If the temperature were increased to 220°F at Point A, the volume of the gas bubbles would increase to 9.9 gallons. Figure 3 shows graphically the relationship between vapor pressure and gas bubble size.

Experience has proven that the presence of gas bubbles entrained in flowing system water generally requires a minimum velocity of 1 1/2 to 2 feet per second to ensure that the bubbles will be carried through the piping. In the vertical return piping, the velocity must be sufficient to overcome the natural tendency for the bubbles to rise. Figure 4 shows the effect of bubble size on the vertical velocity. With bubble size of 0.5 inches, greater than 2 feet per second, water velocity would be required to overcome vertical velocity of the bubble.

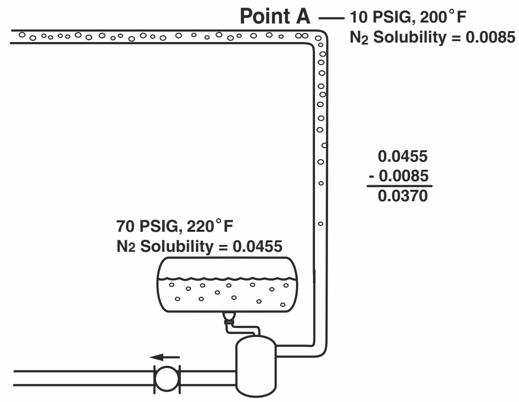


Figure 3.

Effect of Vapor Pressure on the Volume of Gas Bubbles, at 10 PSIG

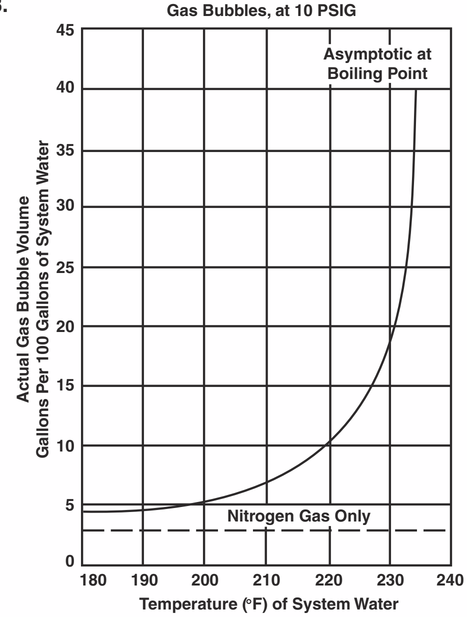
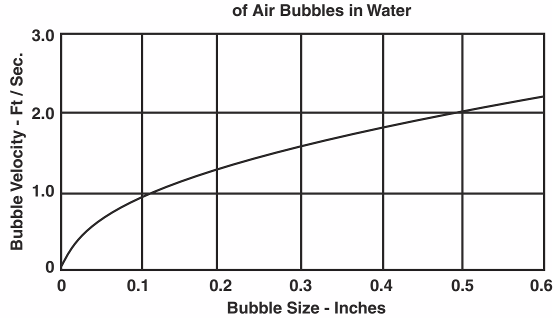


Figure 4.

Effect of Bubble Size on Vertical Velocity of Air Bubbles in Water

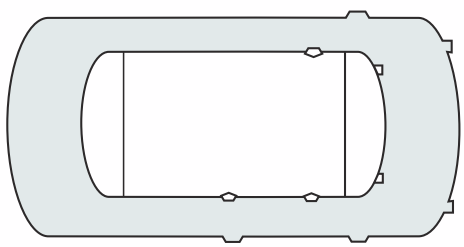


V_{STP} (0.076) = 7.6 Gal. per 100 Gal.

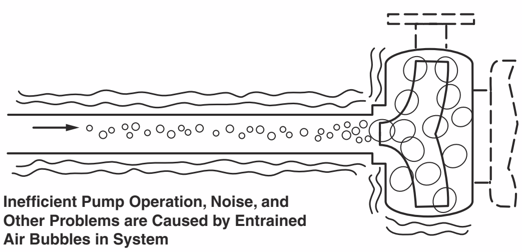
$$V_N = V_{STP}\left(\frac{14.7}{P}\right)\left(\frac{T}{492}\right)$$

$$V_N = 7.6\left(\frac{14.7}{24.7}\right)\left(\frac{510}{492}\right)$$

$$V_N = 4.7 \text{ Gal.}$$



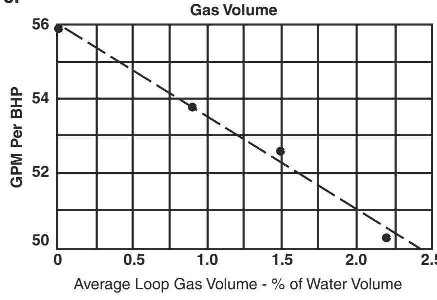
Over-Sizing of Plain Steel Expansion Tank Results in Higher Energy Cost



Inefficient Pump Operation, Noise, and Other Problems are Caused by Entrained Air Bubbles in System

Figure 8.

Overall Test Loop Performance VS Gas Volume



In a Chilled Water System, Colder Water can Absorb a Greater Amount of Gas

Although vapor pressure is not a factor in chilled water systems, the low operating temperature range increases the amount of nitrogen which can be held in solution. Referring to Table 13B page REF-14, Reference Section, will show that at the bottom of the system, at a pressure of 70 psig and a temperature of 50°F, the solubility is 0.107. At the top of the system, at 10 psig pressure, the solubility is 0.031. The difference in solubility, 0.076, results in a calculated entrained nitrogen gas bubble volume of 4.7 gallons for every 100 gallons of flowing system water at the top of the system.

In a chilled water system, the water in the expansion tank, at ambient temperature of 70°F, solubility 0.0871, would hold less nitrogen in solution than system water flowing through the piping at the tank connection. Consequently, although absorbed nitrogen would migrate from the water in the tank to water in the piping, the absorbed nitrogen could not return to the tank.

In addition to the large volume of nitrogen absorbed from the cushion in the expansion tank and carried in an absorbed condition in system piping and components, the large volume of gas bubbles, entrained or in pockets at upper elevations can easily result in the gas cushion in the tank being relatively small. This condition will destroy the tank connection to the system as a reference point for pressures throughout the system with the pump in operation. To compensate for the large amount of gas bubbles in the system, it has been common practice to over-size the plain steel expansion tank by roughly 300% to 500%.

Secondary Pumping and Terminal Unit Performance Are Adversely Affected by the Presence of Gas at Upper Elevations in the System

The presence of a “second air cushion” in the HVAC system has the result that the pressure effect of the main system pump is not wholly additive as desired, thus pressures are generally lower than would be predicted. With lower primary system pressures (accompanied by entrained gas bubble volume, 3%, 4% or 5% of flowing system water volume), inefficient pump operation, wasted pumping energy, noise and, in some cases, cavitation will occur. The presence of entrained gas bubbles in system piping increases pumping cost, as the mass flow rate is less for a given volume pumped.

“Air” pockets trapped in terminal units such as water to air heat exchangers cause inefficient operation with loss of proper heat transfer to occupied spaces and unpredictable balancing. Only by keeping flow rates high and constant, and by maintaining high differential pressures across terminal units, can “air” problems be minimized in the system with a plain steel expansion tank. High constant flow rates and high differential pressures result in wasted pumping power and higher energy cost.

Definitions, Analysis and Evaluation of the Phenomenon of Air in Hot Water and Chilled Water HVAC Systems

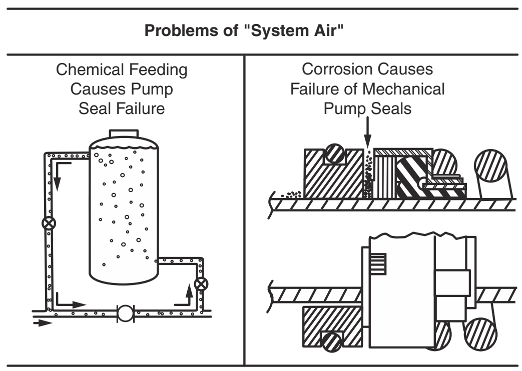
Chapter Five

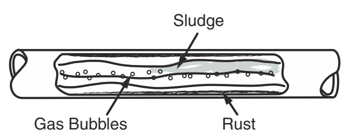
APPLICATION OF THE PRESSURIZATION AND AIR ELIMINATION SYSTEM

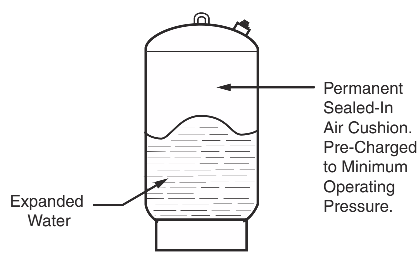
Air Elimination Solves the Problems of "System Air"

The problems caused by "system air" (which are solved by air elimination) are many and varied. Oxygen corrosion causes failure of mechanical pump seals and valves. Rust deposits form in piping and components with reduction in circulation, inefficient pump operation and wasted energy (complete blockage can occur, necessitating replacement or reaming). Addition of chemicals can cause pump seal failure. Formation of sludge requires frequent boiler blowdown, the addition of more makeup water to the system and, in some instances, the dumping of hazardous waste into public sewers or streams.

The formation of gas bubbles causes blocked terminal units, difficulty in balancing system flow, noise and inefficient pump performance. High system water velocities and high differential pressures must be maintained. The presence of gas bubbles in piping and components at upper elevations in the hot water heating or chilled water system destroys the stability of the "point of no pressure change," making it impossible to predict hydraulic performance.







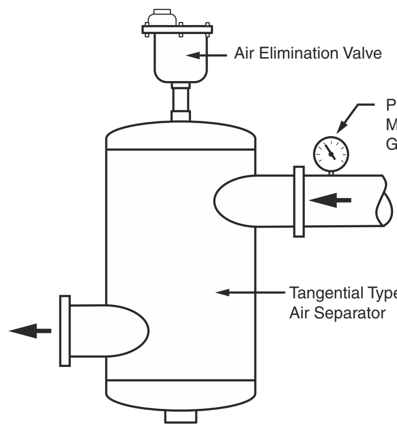
Pre-Pressurized Diaphragm Tank

The Proper Application of the Pressurization and Air Elimination System Will Solve "System Air" Problems

Components:

1. Pressurization Controller:

The pressurization controller is the diaphragm-type or bladder-type expansion tank, precharged to the minimum operating pressure at the location in the system where it is installed. The only air in the system will be the permanent sealed-in air cushion contained in the tank. All other air in the system will be eliminated. The pressurization controller should be located where it can best perform its function, usually at the bottom of the system at the pump suction. As the reference point for pressures throughout the system, it will ensure that any pressure change due to pump operation will be additive, not subtractive. Installation connection at any other point is feasible and occasionally may be desirable, but the pump pressure effects should be considered.



2. Air Separating Elimination Components:

The air separating and elimination component is normally installed at the point of lowest solubility of air in water, typically at a high point in the system. It consists of:

a. A tangential-type air separator which separates entrained air from flowing system water by the creation of a vortex, allowing free air bubbles to rise in the center, the point of lowest velocity, to an air collection chamber.

b. A unique, pilot, gas pressure operated air elimination valve that is guaranteed to open fully at 2-150 psi system pressure when air is present in the valve body. The valve has a high removal rate at low pressure differentials and is capable of eliminating air to the atmosphere as fast as it is separated from system water. The valve is guaranteed to close 100% pressure-tight on systems below 2 psi (including a vacuum), or during the absence of air bubbles at its location. The sealing force is exerted by gas pressure and is far greater than that resulting from float operation.

c. A pressure gauge to monitor the proper minimum operating pressure at the gauge location.

Air Eliminator

Operating pressure range – 2 psig to 150 psig

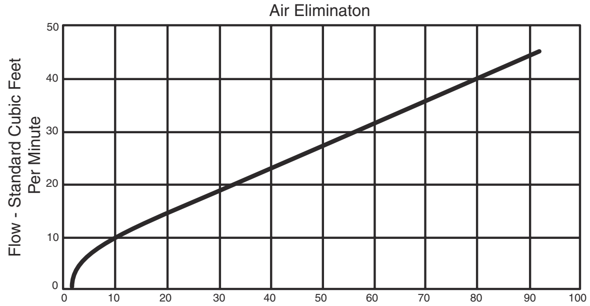
Will not open if negative pressure occur.

Maximum operating temperature - 250°F

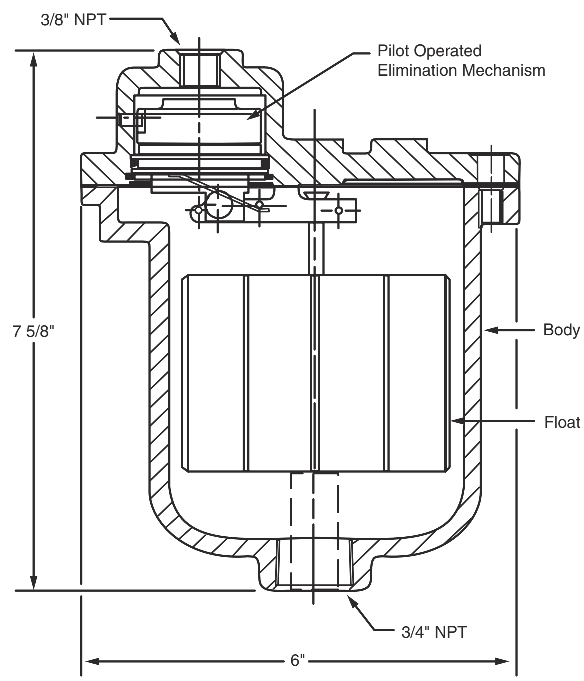
Material: Body and cover – cast iron.
Bolts and nuts – stainless steel.
Pilot mechanism – bronze.

Maximum working pressure — 150 PSIG

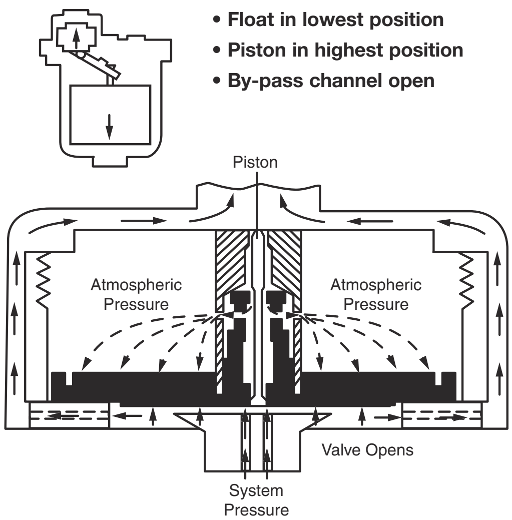
Performance Data Chart
Air Eliminaton

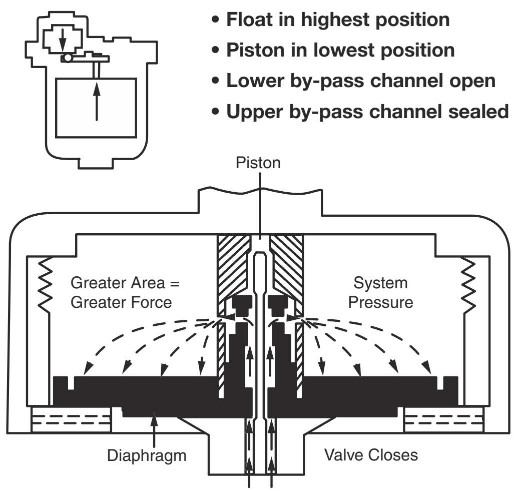


Pressure PSIG (Operating Pressure Range - 2 PSIG to 150 PSIG)



Operation of the Air Elimination Valve





Open Position (Figure C5-1)

When air is accumulated in the body of the valve, the float is in the lowest position, and the piston is in the highest position. The piston allows the upper surface of the diaphragm to be exposed to atmospheric pressure, less than system pressure, 2 psig or greater. The diaphragm is forced upward, off the diaphragm seat, allowing air to escape through radial ports.

Closed Position (Figure C5-2)

When air is exhausted from the body of the valve, the float is in the highest position and the piston is in the lowest position. The piston allows the upper surface of the diaphragm to be exposed to system pressure. Although the pressure is equal on both the upper and lower surfaces, the area of the upper surface is greater than the lower surface. Therefore, the total force is greater and the diaphragm is pressed firmly down against the diaphragm seat, tightly sealing the valve.

The Point of Lowest Solubility of Air in Water

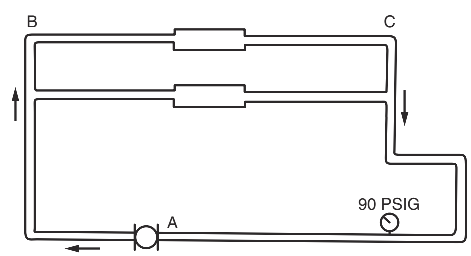
The location in the system where the combined effect of temperature and pressure results in the lowest solubility of air in water is the point where any air would be released as gas bubbles. This is the location for the air separation and elimination components. Table 12 page REF-13 in Reference Supplement Solubility of Air in Water, serves as a convenient tool to evaluate various points.

Example 1.

Multi-story building with relatively high static pressure. At **Point A**, on the suction side of the system pump, the operating pressure is 90 psig.

Point B, at a pressure of 30 psig and a temperature of 210°F, is the point of lowest soluble, air in water and the location for air separation and elimination component.

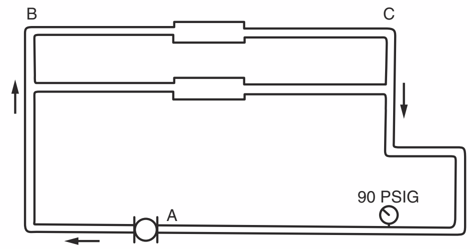
Example 1.



Comparison of Air Solubility

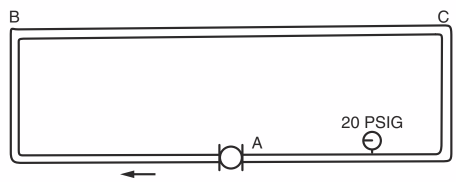
Location	Pressure	Temp.	Solubility
A	90 psig	210°F	.0683
B	30 psig	210°F	.0230
C	20 psig	160°F	.0237

Example 2.



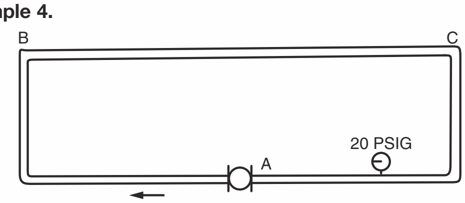
Comparison of Air Solubility			
Location	**Pressure**	**Temp.**	**Solubility**
A	90 psig	210°F	.0683
B	30 psig	210°F	.0230
C	20 psig	170°F	.0223

Example 3.



Comparison of Air Solubility			
Location	**Pressure**	**Temp.**	**Solubility**
A	20 psig	210°F	.0155
B	15 psig	210°F	.0118
C	10 psig	190°F	.0116

Example 4.



Comparison of Air Solubility			
Location	**Pressure**	**Temp.**	**Solubility**
A	20 psig	210°F	.0155
B	25 psig	210°F	.0193
C	15 psig	190°F	.0154

Example 2.

The same building as Example 1, but at Point C, the operating temperature is 170°F.

Point C at a pressure of 20 psig and a temperature of 170°F is the point of lowest solubility.

Example 3.

One-story building with low static pressure. At **Point A**, on the suction side of the system pump, the operating pressure is 20 psig.

Either **Point B** or **Point C** is suitable as the location for the air separation and elimination component. **Point B** would have one advantage in that air bubbles could be removed before system water entered the terminal units at the top of the system.

Example 4.

On the same building, the installation of a higher head pump would cause an increase in pressure and therefore solubility at **Point B** and **Point C.**

Point A or **Point C** is suitable. **Point C** is preferred in that with the pump off, air bubbles which would form adjacent to the point of air separation could be eliminated from the system immediately.

In the four preceding examples, the air separating device was sized for the system pipe size at the point of installation. In many systems, the size of the air separator will be much smaller when installed in piping at the top, and an installation cost saving will be realized. Even though less system water will flow through the separation point in a given period of time, the amount of air to be eliminated from the system will be relatively small with a diaphragm-type tank. Usually, the system will be free of residual air in an hour or so of operation. If the system is very large and velocities are low, several days may be required. A simple test can be made to determine if all residual air has been removed. The presence of air at upper elevations in the system will cause a change in pressure at the pump suction when the pump is operating and when it is not operating (see Chapter 4). If no change occurs, it can be assumed that all air has been eliminated.

With the air separating and elimination components installed at the most crucial location in the system – where system water can hold the least amount of air in solution – residual air will be rapidly eliminated until the system is completely full of water (no air). Only one elimination component is necessary as long as it is in the proper location. More than one may be used if desired…one per major piping zone, for example. If there are many risers in a building, the riser which is most distant from the pump should be selected, since this is where pressure, and thus solubility, is the lowest.

System water acts as a collector and transporter of air in solution. If air bubbles exist somewhere other than at the "point of lowest solubility," they will be absorbed by the water. Water which has released its air at the point of lowest solubility will absorb air everywhere else in the piping, since by definition, it has a higher solubility everywhere else. The air thus absorbed is carried in solution to the air elimination point where it is removed from the system. With the system free of air, it is hydraulically stable, and flow distribution will be reliable at lower flow rates.

With Air Elimination, Initial System Start-Up Procedure Can Be Modified to Reduce Absorbed Air Content in Water Even Further

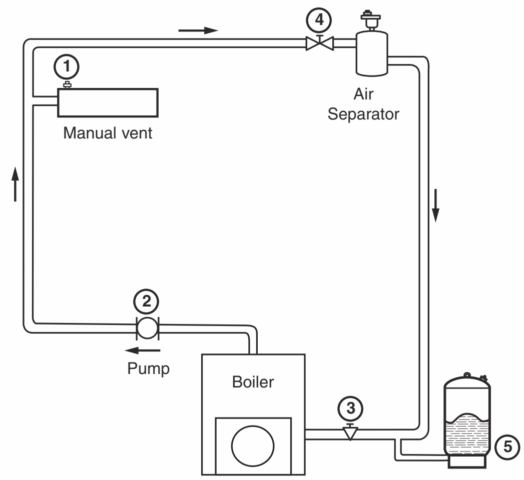
So far, we have recommended eliminating system air through the use of a diaphragm-type tank and by locating the air separating and elimination components at the point of lowest air solubility. This will prevent any air bubbles at any other location in the system. However, if the designer wishes to reduce the amount of air in either the hot water heating system or the chilled water system, the initial system startup procedure can be modified as follows:

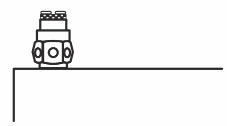
1. After filling, cleaning and flushing the system in accordance with accepted procedures*, refill with clean water, venting at high points and components with manual vents.
2. Circulate the system at ambient temperature, at a minimum velocity of 2 fps, at a pressure slightly below the relief valve setting, to absorb any pockets of air not removed by purging (scavenging the system of gas).
3. Bleed off system water at a location adjacent to the tank connection until the pressure reaches the minimum operating pressure. The decrease in pressure will lower the solubility at the point of air elimination, allowing the air absorbed from any air pockets to be released from the system.
4. On most systems, the pressure can be reduced still further at the point of air elimination by throttling flow at a throttling valve located between the main system pump and the air elimination point. The flow should not be throttled below 2 fps as entrained air bubbles must be carried in the piping to the air elimination point. A 50% reduction in flow will result in a reduction of 75% in pressure, approximately, due to pump operation.

   NOTE: In almost all systems, this procedure will result in the level of solubility being brought down to a point that no air bubbles will form at any period in the operating cycle. The introduction of excessive makeup water would introduce new air to the system. In this situation, the procedure can be repeated. A meter on the fill line from the city main or a flow switch could be used to warn of the addition of excessive makeup water.

5. **(Heating systems only)** With the flow throttled, the system temperature should be gradually increased to the design temperature. Simultaneously, the expanded water should be bled from the system, thus maintaining the system at approximately the minimum operating pressure, during the startup procedure. The additional water which will enter the system later to replace the water bled from the system, will hold a negligible amount of absorbed air.

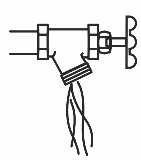
*ASHRAE Handbook and Product Directory, 1980 Systems, Chapter 15.



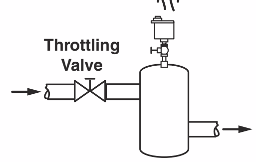


1. Vent Manually at High Points and at Components

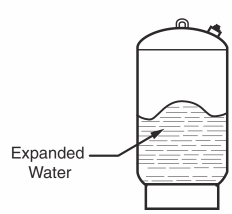
2. Circulate System at Ambient Temperature. Minimum Velocity —2 FPS, Slightly Below Relief Valve Setting



3. Bleed off System Water to Lower Pressure and Solubility



4. Throttle Flow, But Not Below 2 FPS, To Reduce Pressure



5. Bleed Expanded Water to Maintain Minimum Operating Pressure

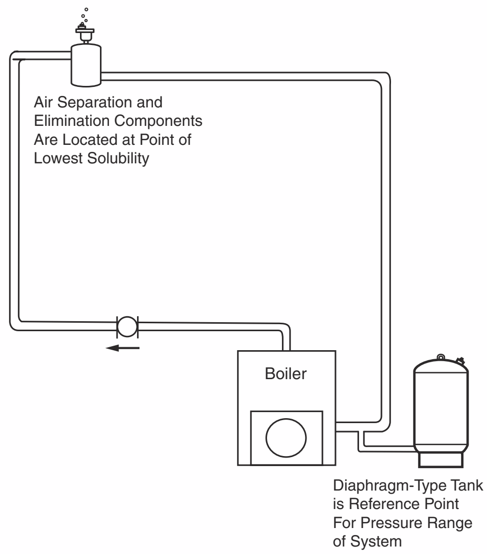
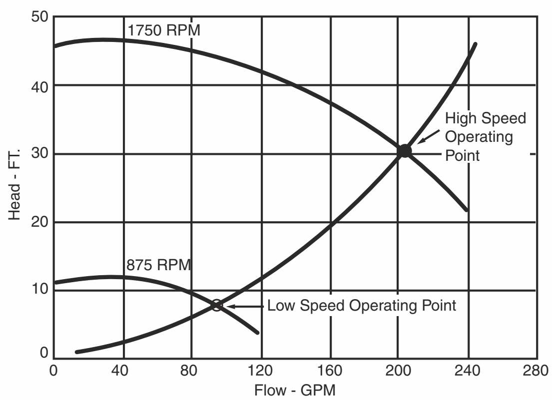


Figure C5-3



With Air Elimination, the Designer Can Design the HVAC System on a Base of Stable Hydraulic Performance

With a properly designed pressurization and air elimination system, all other components in the system can be relied upon to function on a base of stable hydraulic performance. The pressurization controller (diaphragm-type tank), installed at the specific location in the system chosen by the design engineer, will become the reference point for the pressure range throughout the system, at all phases in the operating cycle. The air separating and elimination components installed at the location in the system which is the point of lowest solubility, air in water, will ensure that no gas bubbles will form at another location in the system. Since system water only, not a mixture of air and water, will be flowing through circuits, the balancing of system circuits will be permanent. The lowering of pump efficiency and resulting waste of pumping power because of gas bubble formation will not occur.

The flow through terminal heat exchangers can be designed for seasonal performance without concern for partial air blockage at low flow rates. Reduction in the system water flow rate can be utilized to save significant amounts of pumping energy otherwise wasted with constant full flow. For example, a small pump could be piped in parallel with a large pump. The large pump would operate only during the coldest period in the heating season, possibly 10% of the time. The small pump would be operated during 90% of the heating season.

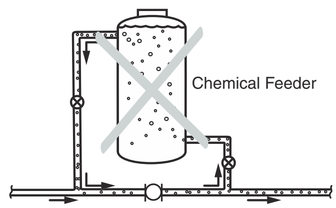
In an example system, a single pump was installed with a two-speed motor, 1,750 rpm and 875 rpm. The flow rate varied directly with speed – at high speed, 200 gpm; at low speed, 100 gpm. The pump head varied as the square of the speed – at high speed, 30 feet; at low speed, 7.5 feet. Figure C5-3 shows the pump curve at high speed and at low speed and the intersection of each with the system curve. In this system, an outdoor temperature sensor allowed the motor to operate at full speed when the temperature dropped below the set point, roughly 10% of the heating season, and at half speed when the temperature was above the set point, roughly 90% of the season. The ratio of the theoretical power required at full flow and that required at half flow is 8 to 1. The actual saving in electric power due to the use of the two-speed motor versus a single-speed motor was 82% (the variance due to a difference in efficiencies).

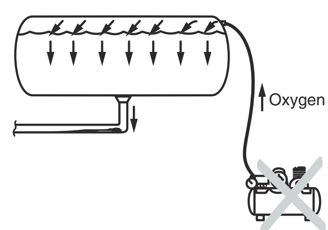
Air elimination permits accurate design, through accurate information, and accurate prediction of hydraulic performance for the life of the system.

With air elimination and a reasonably tight water heating or chilled water system, chemical treatment to prevent oxygen corrosion is not necessary.

In most systems, the principal cause of oxygen corrosion is the plain steel expansion tank, with or without an air compressor, or an open tank.

The installation of the diaphragm-type tank removes this primary source of oxygen. The only oxygen initially in the system will be the relatively small amount contained in the system water at fill plus a certain amount of residual air in components and piping, which will be rapidly removed by the proper air separating and elimination component installed at the correct location in the system. The addition of make-up water can be held to a low level with reasonable care, and any change in the chemical composition of system water should be insignificant. With air elimination and regular inspection and maintenance of mechanical components, the water heating and chilled water system should last the lifetime of the building.





Typical Installations

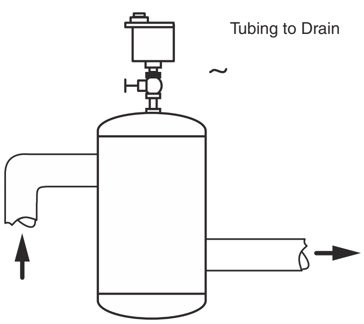


Figure A. The air separator and air elimination valve installed at the top of the supply riser

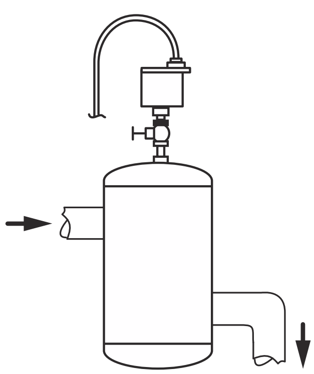


Figure B. The air separator and air elimination valve installed at the top of the vertical return.

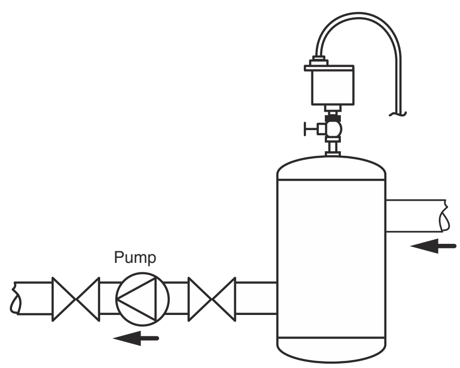


Figure C. The air separator and air elimination valve installed on the suction side of the pump at the bottom of the system.

Shut off valves should be provided to facilitate cleaning and replacement of the float and pilot assembly if necessary.

Because vapor many times escapes with system air and can condense, good practice indicates that a line should be piped to a drain, sink or container which could be readily checked by maintenance personnel.

Typical Specification Pressurization and Air Elimination System

As shown on the drawings, furnish and install a pressurization and air elimination system to accommodate the expanded water generated by the increase in temperature in a water heating or chilled water system and to control the increase in pressure at all critical components in the system to the maximum allowable for those components.

The pressurization and air elimination system shall ensure that all air in the system shall be eliminated. The only air in the system shall be the permanent sealed-in air cushion contained in the pressurization controller component of the system, a diaphragm-type or bladder-type expansion tank, precharged to the minimum operating pressure at the location indicated on the drawing.

The diaphragm-type or bladder-type expansion tank shall be manufactured by a manufacturer who has supplied substantially the same tanks, which on the date of opening of bids, have been in successful commercial use of operation for not less than five years in projects and units of comparable size. The right is reserved by the owner to require the contractor to submit a list of buildings where they have been in operation, so that such investigation as may be deemed necessary may be made before approval.

The diaphragm-type or bladder-type expansion tank shall be welded steel, constructed, tested and stamped in accordance with Section VIII, Division 1 of the ASME Code for a working pressure ( ) psig and shall be supported by steel legs or a base for a vertical installation or steel saddles for horizontal installations.

All free air originally contained in the system, and all entrained air bubbles carried by system water, shall be eliminated at all points in the piping system where the capability of water to hold air in solution is lowest (the point of lowest solubility). The contractor shall require that the manufacturer, or his or her representative, shall determine the correct location of the air separating and elimination component(s). The air separating and elimination component shall separate entrained air from flowing system water by the creation of a vortex which will allow free air to rise in the center, the point of lowest velocity, to an air elimination valve.

The air separator shall be capable of effectively separating not less than 98% of the entrained air on the first passage of water and not less than 98% of the residual air on each subsequent passage. The contractor shall test and document that no free air exists in the system after air elimination is accomplished. The contractor shall guarantee with proper documents that the piping system is tight and will not require chemical treatment for corrosion due to oxygen for a period of at least five years from the date of documentation.

The pressure drop through the air separator with a flow rate of 6 FPS or less shall be no more than 2 PSIG.

The air separator shall be cast iron or welded steel, constructed, tested and stamped in accordance with Section VIII, Division 1 of the ASME Code for the desired working pressure.

Piping shall be as shown on the drawings.

Air shall be eliminated to the atmosphere as fast as it is separated from system water through a float activated, remote pressure operated air elimination valve installed at the top of the air separator. The air elimination valve shall have a high removal rate at low pressure differentials and shall be fully open for the removal of air at all pressures in the operating range from 2 psi to 150 psi. It shall be tightly sealed against loss of system water and shall prevent entrance of air in negative pressure situations.

The valve shall be constructed of metal and all working parts shall be noncorrosive. Working pressure shall be a minimum of 150 psig.

Net Expansion of Propylene Glycol

20% Propylene Glycol – 80% Water

Final Temp	Initial Temperature									
	20	30	40	50	60	70	80	90	100	110
40°	0.0045	0.0035								
50°	0.0070	0.0060	0.0040	0.0025						
60°	0.0095	0.0085	0.0065	0.0050	0.0025					
70°	0.0115	0.0105	0.0085	0.0070	0.0045	0.0020				
80°	0.0145	0.0135	0.0115	0.0100	0.0075	0.0050	0.0030			
90°	0.0170	0.0160	0.0140	0.0125	0.0100	0.0075	0.0055	0.0025		
100°	0.0195	0.0185	0.0165	0.0150	0.0125	0.0100	0.0080	0.0050	0.0025	
110°	0.0230	0.0220	0.0200	0.0185	0.0160	0.0135	0.0115	0.0085	0.0060	0.0035
120°	0.0265	0.0255	0.0235	0.0220	0.0195	0.0170	0.0150	0.0120	0.0095	0.0070
130°	0.0295	0.0285	0.0265	0.0250	0.0225	0.0200	0.0180	0.0150	0.0125	0.0100
140°	0.0330	0.0320	0.0300	0.0285	0.0260	0.0235	0.0215	0.0185	0.0160	0.0135
150°	0.0360	0.0350	0.0330	0.0315	0.0290	0.0265	0.0245	0.0215	0.0190	0.0165
160°	0.0405	0.0395	0.0375	0.0360	0.0335	0.0310	0.0290	0.0260	0.0235	0.0210
170°	0.0445	0.0435	0.0415	0.0400	0.0375	0.0350	0.0330	0.0300	0.0275	0.0250
180°	0.0485	0.0475	0.0455	0.0440	0.0415	0.0390	0.0370	0.0340	0.0315	0.0290
190°	0.0525	0.0515	0.0495	0.0480	0.0455	0.0430	0.0410	0.0380	0.0355	0.0330
200°	0.0565	0.0555	0.0535	0.0520	0.0495	0.0470	0.0450	0.0420	0.0395	0.0370
210°	0.0610	0.0600	0.0580	0.0565	0.0540	0.0515	0.0495	0.0465	0.0440	0.0415
220°	0.0655	0.0645	0.0625	0.0610	0.0585	0.0560	0.0540	0.0510	0.0485	0.0460
230°	0.0705	0.0695	0.0675	0.0660	0.0635	0.0610	0.0590	0.0560	0.0535	0.0510
240°	0.0755	0.0745	0.0725	0.0710	0.0685	0.0660	0.0640	0.0610	0.0585	0.0560
250°	0.0800	0.0790	0.0770	0.0755	0.0730	0.0705	0.0585	0.0655	0.0630	0.0605

Final Temp	Initial Temperature						
	110	120	130	140	150	160	170
120°	0.0035						
130°	0.0065	0.0030					
140°	0.0100	0.0055	0.0035				
150°	0.0130	0.0095	0.0065	0.0030			
160°	0.0175	0.0140	0.0110	0.0075	0.0045		
170°	0.0215	0.0180	0.0150	0.0115	0.0085	0.0040	
180°	0.0255	0.0220	0.0190	0.0155	0.0125	0.0080	0.0040
190°	0.0295	0.0260	0.0230	0.0195	0.0165	0.0120	0.0080
200°	0.0335	0.0300	0.0270	0.0235	0.0205	0.0160	0.0120
210°	0.0380	0.0345	0.0315	0.0280	0.0250	0.0205	0.0165
220°	0.0425	0.0390	0.0360	0.0325	0.0295	0.0250	0.0210
230°	0.0475	0.0440	0.0410	0.0375	0.0345	0.0300	0.0260
240°	0.0525	0.0490	0.0460	0.0425	0.0395	0.0350	0.0310
250°	0.0570	0.0535	0.0505	0.0470	0.0440	0.0395	0.0355

Net Expansion of Propylene Glycol

30% Propylene Glycol – 70% Water

Final Temp	Initial Temperature									
	10	20	30	40	50	60	70	80	90	100
40°	0.0045	0.0035	0.0015							
50°	0.0070	0.0060	0.0040	0.0025						
60°	0.0095	0.0085	0.0065	0.0050	0.0025					
70°	0.0115	0.0105	0.0085	0.0070	0.0045	0.0020				
80°	0.0145	0.0135	0.0115	0.0100	0.0075	0.0050	0.0030			
90°	0.0170	0.0160	0.0140	0.0125	0.0100	0.0075	0.0055	0.0025		
100°	0.0195	0.0185	0.0165	0.0150	0.0125	0.0100	0.0080	0.0050	0.0025	
110°	0.0230	0.0220	0.0200	0.0185	0.0160	0.0135	0.0115	0.0085	0.0060	0.0035
120°	0.0265	0.0255	0.0235	0.0220	0.0195	0.0170	0.0150	0.0120	0.0095	0.0070
130°	0.0295	0.0285	0.0265	0.0250	0.0225	0.0200	0.0180	0.0150	0.0125	0.0100
140°	0.0330	0.0320	0.0300	0.0285	0.0260	0.0235	0.0215	0.0185	0.0160	0.0135
150°	0.0360	0.0350	0.0330	0.0315	0.0290	0.0265	0.0245	0.0215	0.0190	0.0165
160°	0.0405	0.0395	0.0375	0.0360	0.0335	0.0310	0.0290	0.0260	0.0235	0.0210
170°	0.0445	0.0435	0.0415	0.0400	0.0375	0.0350	0.0330	0.0300	0.0275	0.0250
180°	0.0485	0.0475	0.0455	0.0440	0.0415	0.0390	0.0370	0.0340	0.0315	0.0290
190°	0.0525	0.0515	0.0495	0.0480	0.0455	0.0430	0.0410	0.0380	0.0355	0.0330
200°	0.0565	0.0555	0.0535	0.0520	0.0495	0.0470	0.0450	0.0420	0.0395	0.0370
210°	0.0610	0.0600	0.0580	0.0565	0.0540	0.0515	0.0495	0.0465	0.0440	0.0415
220°	0.0655	0.0645	0.0625	0.0610	0.0585	0.0560	0.0540	0.0510	0.0485	0.0460
230°	0.0705	0.0695	0.0675	0.0660	0.0635	0.0610	0.0590	0.0560	0.0535	0.0510
240°	0.0755	0.0745	0.0725	0.0710	0.0685	0.0660	0.0640	0.0610	0.0585	0.0560
250°	0.0800	0.0790	0.0770	0.0755	0.0730	0.0705	0.0585	0.0655	0.0630	0.0605

Final Temp	Initial Temperature						
	110	120	130	140	150	260	270
120°	0.0035						
130°	0.0065	0.0030					
140°	0.0100	0.0055	0.0035				
150°	0.0130	0.0095	0.0065	0.0030			
160°	0.0175	0.0140	0.0110	0.0075	0.0045		
170°	0.0215	0.0180	0.0150	0.0115	0.0085	0.0040	
180°	0.0255	0.0220	0.0190	0.0155	0.0125	0.0080	0.0040
190°	0.0295	0.0260	0.0230	0.0195	0.0165	0.0120	0.0080
200°	0.0335	0.0300	0.0270	0.0235	0.0205	0.0160	0.0120
210°	0.0380	0.0345	0.0315	0.0280	0.0250	0.0205	0.0165
220°	0.0425	0.0390	0.0360	0.0325	0.0295	0.0250	0.0210
230°	0.0475	0.0440	0.0410	0.0375	0.0345	0.0300	0.0260
240°	0.0525	0.0490	0.0460	0.0425	0.0395	0.0350	0.0310
250°	0.0570	0.0535	0.0505	0.0470	0.0440	0.0395	0.0355

Net Expansion of Propylene Glycol

40% Propylene Glycol – 60% Water

Final Temp	Initial Temperature									
	-10	0	10	20	30	40	50	60	70	80
40°	0.0120	0.0100	0.008	0.0055	0.0030					
50°	0.0145	0.0125	0.0105	0.0080	0.0055	0.0025				
60°	0.0175	0.0155	0.0135	0.011	0.0085	0.0055	0.0030			
70°	0.0210	0.0190	0.0170	0.0145	0.0120	0.0090	0.0065	0.0035		
80°	0.0240	0.0220	0.0200	0.0175	0.0150	0.0120	0.0095	0.0065	0.0030	
90°	0.0275	0.0255	0.0235	0.0210	0.0185	0.0155	0.0130	0.0100	0.0065	0.0035
100°	0.0310	0.0290	0.0270	0.0245	0.0220	0.0190	0.0165	0.0135	0.0100	0.0070
110°	0.0345	0.0325	0.0305	0.0280	0.0255	0.0225	0.0200	0.0170	0.0135	0.0105
120°	0.0385	0.0365	0.0345	0.0320	0.0295	0.0265	0.0240	0.0210	0.0175	0.0145
130°	0.0425	0.0405	0.0385	0.0360	0.0335	0.0305	0.0280	0.0250	0.0215	0.0185
140°	0.0465	0.0445	0.0425	0.0400	0.0375	0.0345	0.0320	0.0290	0.0255	0.0225
150°	0.0510	0.0490	0.0470	0.0445	0.0420	0.0390	0.0365	0.0335	0.0300	0.0270
160°	0.0550	0.0530	0.0510	0.0485	0.0460	0.0430	0.0405	0.0375	0.0340	0.0310
170°	0.0595	0.0575	0.0555	0.0530	0.0505	0.0475	0.0450	0.0420	0.0385	0.0355
180°	0.0640	0.0620	0.0600	0.0575	0.0550	0.0520	0.0495	0.0465	0.0430	0.0400
190°	0.0685	0.0665	0.0645	0.0620	0.0595	0.0565	0.0540	0.0510	0.0475	0.0445
200°	0.0730	0.0710	0.0690	0.0665	0.0640	0.0610	0.0585	0.0555	0.0520	0.0490
210°	0.0775	0.0755	0.0735	0.0710	0.0685	0.0655	0.0630	0.0600	0.0565	0.0535
220°	0.0860	0.0840	0.0820	0.0795	0.0770	0.0740	0.0715	0.0685	0.0650	0.0620
230°	0.0910	0.0890	0.0870	0.0845	0.0820	0.0790	0.0765	0.0735	0.0700	0.0670
240°	0.0955	0.0935	0.0915	0.0890	0.0865	0.0835	0.0810	0.0780	0.0745	0.0715
250°	1.0540	1.0520	1.0500	1.0475	1.0450	1.0420	1.0395	1.0365	1.0330	1.0300

Final Temp	Initial Temperature									
	80	90	100	110	120	130	140	150	160	170
90°	0.0035									
100°	0.0070	0.0035								
110°	0.0105	0.0070	0.0035							
120°	0.0145	0.0110	0.0075	0.0040						
130°	0.0185	0.0150	0.0115	0.0080	0.0040					
140°	0.0225	0.0190	0.0155	0.0120	0.0080	0.0040				
150°	0.0270	0.0235	0.0200	0.0165	0.0125	0.0085	0.0045			
160°	0.0310	0.0275	0.0240	0.0205	0.0165	0.0125	0.0085	0.0040		
170°	0.0355	0.0320	0.0285	0.0250	0.0210	0.0170	0.0130	0.0085	0.0045	
180°	0.0445	0.0410	0.0375	0.0340	0.0300	0.0260	0.0220	0.0175	0.0045	0.0090
190°	0.0490	0.0455	0.0420	0.0385	0.0345	0.0305	0.0265	0.0220	0.0180	0.0135
200°	0.0535	0.0500	0.0465	0.0430	0.0390	0.0350	0.0310	0.0265	0.0225	0.0180
210°	0.0575	0.0540	0.0505	0.0470	0.0430	0.0390	0.0350	0.0305	0.0265	0.0220
220°	0.0620	0.0585	0.0550	0.0515	0.0475	0.0435	0.0395	0.0350	0.0310	0.0265
230°	0.0670	0.0635	0.0600	0.0565	0.0525	0.0485	0.0445	0.0400	0.0360	0.0315
240°	0.0715	0.0680	0.0645	0.0610	0.9585	0.0530	0.0490	0.0445	0.0405	0.0360
250°	1.0300	1.0265	1.0230	1.0195	1.0155	1.0115	1.0075	1.0030	0.9990	0.9945

Net Expansion of Propylene Glycol

50% Propylene Glycol – 50% Water

Final Temp	Initial Temperature									
	-20	-10	0	10	20	30	40	50	60	70
40°	0.0170	0.0150	0.0125	0.0095	0.0065	0.0030				
50°	0.0205	0.0185	0.0160	0.0130	0.0100	0.0065	0.0035			
60°	0.0240	0.0220	0.0195	0.0165	0.0135	0.0100	0.0070	0.0035		
70°	0.0275	0.0255	0.0230	0.0200	0.0170	0.0135	0.0105	0.0070	0.0035	
80°	0.0315	0.0295	0.0270	0.0240	0.0210	0.0175	0.0145	0.0110	0.0075	0.0040
90°	0.0355	0.0335	0.0310	0.0280	0.0250	0.0215	0.0185	0.0150	0.0115	0.0080
100°	0.0390	0.0370	0.0345	0.0315	0.0285	0.0250	0.0220	0.0185	0.0150	0.0115
110°	0.0430	0.0410	0.0385	0.0355	0.0325	0.0290	0.0260	0.0225	0.0190	0.0155
120°	0.0470	0.0450	0.0425	0.0395	0.0365	0.0330	0.0300	0.0265	0.0230	0.0195
130°	0.0515	0.0495	0.0470	0.0440	0.0410	0.0375	0.0345	0.0310	0.0275	0.0240
140°	0.0570	0.0550	0.0525	0.0495	0.0465	0.0430	0.0400	0.0365	0.0330	0.0295
150°	0.0600	0.0580	0.0555	0.0525	0.0495	0.0460	0.0430	0.0395	0.0360	0.0325
160°	0.0645	0.0625	0.0600	0.0570	0.0540	0.0505	0.0475	0.0440	0.0405	0.0370
170°	0.0690	0.0670	0.0645	0.0615	0.0585	0.0550	0.0520	0.0485	0.0450	0.0415
180°	0.0735	0.0715	0.0690	0.0660	0.0630	0.0595	0.0565	0.0530	0.0495	0.0460
190°	0.0780	0.0760	0.0735	0.0705	0.0675	0.0640	0.0610	0.0575	0.0540	0.0505
200°	0.0825	0.0805	0.0780	0.0750	0.0720	0.0685	0.0655	0.0620	0.0585	0.0550
210°	0.0870	0.0850	0.0825	0.0795	0.0765	0.0730	0.0700	0.0665	0.0630	0.0595
220°	0.0920	0.0900	0.0875	0.0845	0.0815	0.0780	0.0750	0.0715	0.0680	0.0645
230°	0.0955	0.0945	0.0920	0.0890	0.0860	0.0825	0.0795	0.0760	0.0725	0.0690
240°	0.1010	0.0990	0.0965	0.0935	0.0905	0.0870	0.0840	0.0805	0.0770	0.0735
250°	0.1070	0.1050	0.1025	0.0995	0.0965	0.0930	0.0900	0.0865	0.0830	0.0795

Final Temp	Initial Temperature									
	80	90	100	110	120	130	140	150	160	170
90°	0.0040									
100°	0.0075	0.0035								
110°	0.0115	0.0075	0.0040							
120°	0.0155	0.0115	0.0080	0.0040						
130°	0.0200	0.0160	0.0125	0.0085	0.0045					
140°	0.0255	0.0215	0.0180	0.0140	0.0100	0.0055				
150°	0.0285	0.0245	0.0210	0.0170	0.0130	0.0085	0.0030			
160°	0.0330	0.0290	0.0255	0.0215	0.0175	0.0130	0.0075	0.0045		
170°	0.0375	0.0335	0.0300	0.0260	0.0220	0.0175	0.0120	0.0090	0.0045	
180°	0.0420	0.0380	0.0345	0.0305	0.0265	0.0220	0.0165	0.0135	0.0045	0.0045
190°	0.0465	0.0425	0.0390	0.0350	0.0310	0.0265	0.0210	0.0180	0.0135	0.0090
200°	0.0510	0.0470	0.0435	0.0395	0.0355	0.0310	0.0255	0.0225	0.0180	0.0135
210°	0.0555	0.0515	0.0480	0.0440	0.0400	0.0355	0.0300	0.0270	0.0225	0.0180
220°	0.0605	0.0565	0.0530	0.0490	0.0450	0.0405	0.0350	0.0320	0.0275	0.0230
230°	0.0650	0.0610	0.0575	0.0535	0.0495	0.0450	0.0395	0.0365	0.0320	0.0275
240°	0.0695	0.0655	0.0620	0.0580	0.9660	0.0495	0.0440	0.0410	0.0365	0.0320
250°	0.0755	0.0715	0.0680	0.0640	0.0600	0.0555	0.0500	0.0470	0.0425	0.0380

Table 15 – Method for Estimating System Volume (Vs)

The following equation is based on the two largest dimensions (length, width, height) of the building and the gallon per lineal foot of the largest diameter main (in inches) of the building. An approximation is made considering the length of the main in a reverse return piping system where the thermal source (boiler/chiller) is located furthest from the last terminal (load) before returning to the source. This distance (supply and return) is doubled to approximate the volume represented in all additional circuit piping, boiler or chiller, fan coils, valves and fittings.

Actual system volumes in systems that were estimated by this method have been checked in a number of buildings. In reverse-return systems, the estimates were over-sized by 10% to 15%. In other systems checked, over-sizing ranged to 25%.

$Vs = (Dim_1 + Dim_2) \times 4 \times$ gal./lin. ft.

Where:

$(Dim_1 + Dim_2) =$ The sum of the two largest dimensions of the building (in feet).

4 = Is a constant

Gal./lin. ft. = The gallons per lineal feet of the largest pipe, or main (in inches). Select from Table 2 or estimate as follows:

$$\frac{(Dia)^2}{24}$$

Example:

Width 120'
Length 408'
Height 200'
Main shown as 6" pipe

$$Vs = (\text{length} + \text{height}) \times 4 \times \frac{(Dia")^2}{24}$$

$$Vs = (408' + 200') \times 4 \times \frac{(36)}{24}$$

$$Vs = 608 \times 4 \times (1.5)$$

$$Vs = 3{,}648 \text{ Gal.}$$

Simplified equation derived from above:

$$Vs = \frac{(Dim_1 + Dim_2) \times (Dia")^2}{6}$$

Example:

$$Vs = \frac{(408 + 200) \times 36}{6}$$

Vs=3,648 Gal.

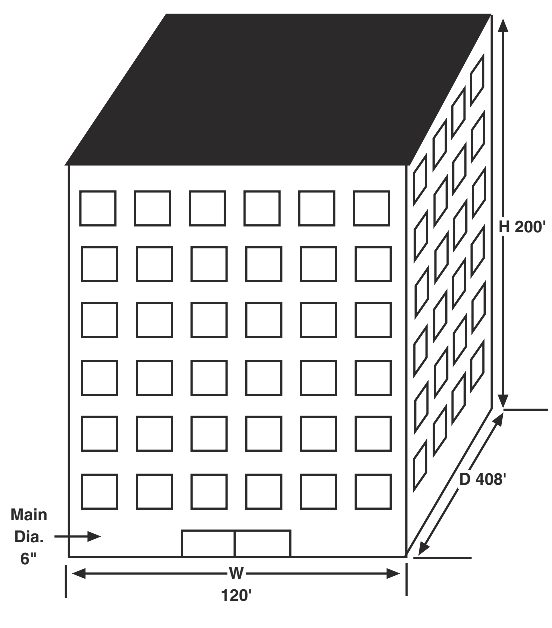


Table 1 –

Water Content…
Unit Heaters, Fan Coil
Units and Convectors
(BTUH to Gallons Conversion Factors)

	Gal./10,000 BTUH	
	At 200°F	**At 180°F**
Convectors	0.64	-
Unit Heaters	-	0.2
Fan Coil Units	-	0.2

Table 2 –

Volume of Water in Pipe Tubing

Gallons per lineal Foot

Nominal Pipe Size in Inches	Steel Pipe	Copper Tube
	Gal/Foot	Gal/Foot
1/2	0.016	0.012
3/4	0.028	0.025
1	0.045	0.043
1 1/4	0.078	0.065
1 1/2	0.105	0.092
2	0.172	0.161
2 1/2	0.250	0.250
3	0.385	0.357
4	0.667	0.625
5	1.000	1.000
6	1.500	1.400
8	2.630	2.430
10	4.200	3.780
12	5.900	5.400

Table 3 –

Water Content… Heat Exchangers

Shell Dia. (Nominal pipe size in inches)	Gal./Foot of Shell Length	
	In Shell	In Tubes
4	0.4	0.2
6	1.0	0.5
8	1.8	0.9
10	2.4	1.2
12	4.0	2.2
14	5.0	2.6
16	6.5	3.5
18	8.0	4.5
20	10.0	5.5
24	15.0	7.5

Table 4 –

Adequate Positive Pressures to Prevent Flashing
Based on tables in "Thermodynamic Properties of Steam" by Keenan & Keyes, published by John Wiley & Sons, Inc.

The following table includes a safety factor of 15°F.

Max. Operating Temperature in °F.	Adequate Positive Pressure Must Be At Least:
200°	2 psig *
205°	3 psig *
210°	5 psig
215°	7 psig
220°	9 psig
225°	11 psig
230°	13 psig
235°	16 psig
240°	19 psig
245°	21 psig
250°	25 psig
255°	28 psig
260°	32 psig
265°	35 psig
270°	40 psig
275°	44 psig
280°	49 psig
285°	53 psig
290°	58 psig
295°	64 psig
300°	70 psig

Recommended Adequate Positive Pressure to ensure **Positive Air Venting at the top of the system should be **at least 5 psig**. When Air Elimination as defined in **Section D** is used, **2 psig** is recommended.*

Table 5 – Net Expansion of Water

Factors for calculating Net Expansion of Water

Gross expansion minus system piping expansion. (Based on expansion of steel.)

Final Temp °F	Initial Temperature °F												
	40	45	50	55	60	65	70	75	80	85	90	95	100
50°	.00006	.00008											
55°	.00025	.00027	.00019										
60°	.00055	.00057	.00049	.00030									
65°	.00093	.00095	.00087	.00068	.00038								
70°	.00149	.00151	.00143	.00124	.00094	.00056							
75°	.00194	.00196	.00188	.00169	.00139	.00101	.00045						
80°	.00260	.00262	.00254	.00235	.00205	.00167	.00111	.00066					
85°	.00326	.00328	.00320	.00301	.00271	.00233	.00177	.00132	.00066				
90°	.00405	.00407	.00399	.00380	.00350	.00312	.00256	.00211	.00145	.00079			
95°	.00485	.00487	.00479	.00460	.00430	.00392	.00336	.00291	.00225	.00159	.00080		
100°	.00575	.00577	.00569	.00550	.00520	.00485	.00426	.00381	.00315	.00249	.00170	.00090	
105°	.00671	.00673	.00665	.00646	.00616	.00578	.00522	.00477	.00411	.00345	.00266	.00186	.00096
110°	.00771	.00773	.00765	.00746	.00716	.00678	.00622	.00577	.00511	.00445	.00366	.00286	.00196
115°	.00879	.00881	.00873	.00854	.00824	.00786	.00730	.00685	.00619	.00553	.00474	.00394	.00304
120°	.01004	.01006	.00998	.00979	.00949	.00911	.00855	.00810	.00744	.00678	.00599	.00519	.00429
125°	.01111	.01113	.01105	.01086	.01056	.01018	.00962	.00917	.00851	.00785	.00706	.00625	.00536
130°	.01236	.01238	.01230	.01211	.01181	.01143	.01087	.01042	.00976	.00910	.00831	.00751	.00661
135°	.01368	.01370	.01362	.01342	.01313	.01275	.01219	.01174	.01108	.01042	.00963	.00883	.00793
140°	.01501	.01503	.01495	.01476	.01446	.01408	.01352	.01307	.01241	.01175	.01096	.01016	.00926
145°	.01643	.01645	.01637	.01618	.01588	.01550	.01494	.01449	.01383	.01317	.01238	.01158	.01068
150°	.01787	.01787	.01779	.01760	.01730	.01692	.01636	.01591	.01525	.01459	.01330	.01300	.01210
155°	.01937	.01939	.01931	.01912	.01882	.01844	.01788	.01743	.01677	.01611	.01532	.01452	.01362
160°	.02092	.02094	.02086	.02067	.02037	.01999	.01943	.01877	.01811	.01732	.01652	.01572	.01482
165°	.02252	.02254	.02246	.02227	.02197	.02159	.02103	.02058	.01992	.01926	.01847	.01767	.01677
170°	.02418	.02420	.02412	.02393	.02363	.02325	.02269	.02224	.02158	.02092	.02013	.01933	.01843
175°	.02588	.02590	.02582	.02563	.02533	.02495	.02439	.02394	.02328	.02262	.02183	.02103	.02013
180°	.02763	.02765	.02757	.02738	.02708	.02670	.02614	.02569	.02503	.02437	.02358	.02278	.02188
185°	.02941	.02943	.02935	.02916	.02886	.02848	.02792	.02747	.02681	.02615	.02536	.02456	.02366
190°	.03127	.03129	.03121	.03102	.03072	.03034	.02978	.02933	.02867	.02801	.02722	.02642	.02552
195°	.03314	.03316	.03308	.03289	.03259	.03221	.03165	.03120	.03065	.02988	.02909	.02829	.02739
200°	.03510	.03512	.03504	.03485	.03455	.03417	.03361	.03316	.03250	.03184	.03105	.03025	.02935
205°	.03707	.03709	.03701	.03682	.03652	.03614	.03558	.03513	.03447	.03381	.03302	.03222	.03135
210°	.03911	.03913	.03905	.03885	.03856	.03818	.03762	.03717	.03651	.03585	.03506	.03426	.03336
215°	.04120	.04122	.04114	.04095	.04065	.04027	.03971	.03926	.03860	.03794	.03715	.03635	.03545
220°	.04335	.04337	.04329	.04310	.04280	.04242	.04186	.04141	.04075	.04009	.03930	.03850	.03760
225°	.04549	.04551	.04543	.04524	.04494	.04456	.04400	.04355	.04289	.04223	.04144	.04064	.03974
230°	.04762	.04764	.04756	.04737	.04707	.04669	.04613	.04568	.04502	.04436	.04357	.04277	.04187
235°	.04991	.04993	.04985	.04966	.04936	.04898	.04842	.04797	.04731	.04665	.04586	.04506	.04416
240°	.05220	.05222	.05214	.05195	.05165	.05127	.05071	.05026	.04960	.04894	.04815	.04735	.04645
245°	.05449	.05451	.05443	.05424	.05394	.05356	.05300	.05255	.05189	.05123	.05044	.04964	.04874
250°	.05684	.05686	.05678	.05659	.05629	.05591	.05535	.05490	.05424	.05358	.05279	.05199	.05109
255°	.05929	.05931	.05923	.05904	.05874	.05836	.05780	.05735	.05669	.05603	.05524	.05444	.05354
260°	.06174	.06176	.06168	.06149	.06119	.06081	.06025	.05980	.05914	.05848	.05769	.05689	.05599
265°	.06420	.06422	.06414	.06395	.06365	.06327	.06271	.06226	.06160	.06094	.06015	.05935	.05845
270°	.06685	.06687	.06679	.06660	.06630	.06592	.06536	.06491	.06425	.06359	.06280	.06200	.06110
275°	.06956	.06958	.06950	.06931	.06901	.06863	.06807	.06762	.06696	.06630	.06551	.06471	.06381
280°	.07226	.07228	.07220	.07201	.07171	.07133	.07077	.07032	.06966	.06900	.06821	.06741	.06651
285°	.07500	.07502	.07494	.07475	.07445	.07407	.07351	.07306	.07240	.07174	.07095	.07015	.06925
290°	.07790	.07792	.07784	.07765	.07735	.07697	.07641	.07596	.07530	.07464	.07385	.07305	.07215
295°	.08080	.08082	.08074	.08055	.08025	.07987	.07931	.07886	.07820	.07754	.07675	.07595	.07505
300°	.08370	.08372	.08364	.08345	.08315	.08277	.08221	.08176	.08110	.08044	.07965	.07885	.07795

Continued

Table 5 – Net Expansion of Water

Factors for calculating Net Expansion of Water

Gross expansion minus system piping expansion. (Based on expansion of steel.)

Final Temp °F	Initial Temperature °F												
	105	110	115	120	125	130	135	140	145	150	155	160	165°
110°	.00100												
115°	.00208	.00108											
120°	.00333	.00233	.00125										
125°	.00440	.00340	.00232	.00107									
130°	.00565	.00465	.00357	.00232	.00125								
135°	.00697	.00597	.00489	.00364	.00257	.00132							
140°	.00830	.00730	.00622	.00497	.00390	.00265	.00133						
145°	.00972	.00872	.00764	.00639	.00532	.00407	.00275	.00142					
150°	.01114	.01014	.00906	.00783	.00676	.00551	.00419	.00286	.00144				
155°	.01266	.01166	.01058	.00933	.00826	.00701	.00569	.00436	.00294	.00150			
160°	.01386	.01286	.01178	.01088	.00981	.00856	.00724	.00591	.00449	.00305	.00155		
165°	.01581	.01481	.01373	.01248	.01141	.01016	.00884	.00751	.00609	.00465	.00315	.00160	
170°	.01747	.01647	.01539	.01414	.01307	.01182	.01050	.00917	.00775	.00631	.00481	.00326	.00166
175°	.01917	.01817	.01709	.01584	.01477	.01352	.01220	.01087	.00945	.00801	.00651	.00496	.00336
180°	.02092	.01992	.01884	.01759	.01652	.01527	.01395	.01262	.01120	.00976	.00826	.00671	.00511
185°	.02270	.02170	.02062	.01937	.01830	.01705	.01573	.01440	.01298	.01154	.01004	.00849	.00689
190°	.02456	.02356	.02248	.02123	.02016	.01891	.01759	.01626	.01484	.01340	.01190	.01035	.00875
195°	.02643	.02543	.02435	.02310	.02203	.02078	.01946	.01813	.01671	.01527	.01377	.01222	.01062
200°	.02839	.02739	.02631	.02506	.02398	.02274	.02142	.02009	.01867	.01723	.01573	.01418	.01258
205°	.03036	.02936	.02828	.02703	.02596	.02471	.02339	.02206	.02064	.01920	.01770	.01615	.01455
210°	.03240	.03140	.03032	.02907	.02800	.02675	.02543	.02410	.02268	.02124	.01974	.01819	.01659
215°	.03449	.03349	.03241	.03116	.03009	.02884	.02752	.02619	.02477	.02333	.02183	.02028	.01868
220°	.03664	.03564	.03456	.03331	.03224	.03099	.02967	.02834	.02692	.02548	.02398	.02243	.02083
225°	.03878	.03778	.03670	.03545	.03438	.03313	.03181	.03048	.02906	.02762	.02612	.02457	.02297
230°	.04091	.03991	.03883	.03758	.03651	.03526	.03394	.03261	.03119	.02975	.02825	.02670	.02510
235°	.04320	.04220	.04112	.03987	.03880	.03755	.03623	.03491	.03348	.03204	.03054	.02899	.02739
240°	.04549	.04449	.04341	.04216	.04109	.03984	.03852	.03719	.03577	.03433	.03283	.03128	.02968
245°	.04778	.04678	.04570	.04445	.04338	.04213	.04081	.03948	.03806	.03662	.03512	.03357	.03197
250°	.05013	.04913	.04805	.04680	.04573	.04448	.04316	.04183	.04041	.03897	.03747	.03592	.03432
255°	.05258	.05158	.05050	.04925	.04818	.04693	.04561	.04428	.04286	.04142	.03992	.03837	.03677
260°	.05503	.05403	.05295	.05170	.05063	.04938	.04806	.04673	.04531	.04387	.04237	.04082	.03922
265°	.05749	.05649	.05541	.05416	.05309	.05184	.05052	.04919	.04777	.04633	.04483	.04328	.04168
270°	.06014	.05914	.05806	.05681	.05574	.05449	.05317	.05184	.05042	.04898	.04748	.04593	.04433
275°	.06285	.06185	.06077	.05952	.05845	.05720	.05588	.05455	.05313	.05169	.05019	.04864	.04704
280°	.06555	.06455	.06347	.06222	.06115	.05990	.05858	.05725	.05583	.05439	.05289	.05134	.04974
285°	.06829	.06729	.06621	.06496	.06383	.06264	.06132	.05999	.05857	.05713	.05563	.05408	.05248
290°	.07119	.07019	.06911	.06786	.06679	.06554	.06422	.06289	.06147	.06003	.05853	.05698	.05538
295°	.07409	.07309	.07201	.07076	.06969	.06844	.06712	.06579	.06437	.06293	.06143	.05988	.05828
300°	.07699	.07599	.07491	.07366	.07259	.07134	.07002	.06869	.06727	.06583	.06433	.06278	.06118

Continued

Table 5 – Net Expansion of Water

Factors for Calculating Net Expansion of Water

Gross expansion minus system piping expansion. (Based on expansion of steel.)

Final Temp	Initial Temperature °F												
°F	**170**	**175**	**180**	**185**	**190**	**195**	**200**	**205**	**210**	**215**	**220**	**225**	**230**
175°	.00170												
180°	.00345	.00175											
185°	.00523	.00353	.00178										
190°	.00709	.00539	.00364	.00186									
195°	.00896	.00726	.00551	.00373	.00187								
200°	.01092	.00922	.00747	.00569	.00383	.00196							
205°	.01289	.01119	.00944	.00766	.00580	.00393	.00197						
210°	.01493	.01323	.01148	.00970	.00784	.00597	.00401	.00204					
215°	.01702	.01532	.01357	.01179	.00993	.00806	.00610	.00413	.00209				
220°	.01917	.01747	.01572	.01394	.01208	.01021	.00825	.00628	.00424	.00215			
225°	.02131	.01961	.01786	.01608	.01422	.01235	.01038	.00842	.00638	.00429	.00214		
230°	.02344	.02174	.01999	.01821	.01635	.01448	.01252	.01055	.00851	.00642	.00427	.00213	
235°	.02573	.02403	.02228	.02050	.01864	.01677	.01481	.01284	.01080	.00871	.00656	.00442	.00229
240°	.02802	.02632	.02457	.02279	.02093	.01906	.01710	.01513	.01309	.01100	.00885	.00671	.00458
245°	.03031	.02861	.02686	.02508	.02322	.02135	.01939	.01742	.01538	.01329	.00215	.00900	.00687
250°	.03266	.03096	.02921	.02743	.02557	.02370	.02174	.01977	.01773	.01564	.01349	.01135	.00922
255°	.03511	.03341	.03166	.02988	.02802	.02615	.02419	.02222	.02018	.01809	.01594	.01380	.01167
260°	.03756	.03586	.03411	.03233	.03047	.02860	.02664	.02467	.02263	.02054	.01839	.01625	.01412
265°	.04002	.03832	.03657	.03479	.03293	.03106	.02910	.02713	.02509	.02300	.02085	.01871	.01658
270°	.04267	.04097	.03922	.03744	.03558	.03371	.03175	.02978	.02774	.02565	.02350	.02136	.01923
275°	.04538	.04368	.04193	.04015	.03829	.03642	.03446	.03249	.03045	.02836	.02621	.02407	.02194
280°	.04808	.04638	.04463	.04285	.04099	.03912	.03716	.03519	.03315	.03106	.02891	.02677	.02464
285°	.05082	.04912	.04737	.04559	.04373	.04186	.03990	.03793	.03589	.03380	.03165	.02951	.02738
290°	.05372	.05202	.05027	.04849	.04663	.04476	.04280	.04083	.03879	.03670	.03455	.03241	.03028
295°	.05662	.05492	.05317	.05139	.04953	.04766	.04570	.04373	.04169	.03960	.03745	.03531	.03318
300°	.05952	.05782	.05607	.05429	.05243	.05056	.04860	.04663	.04459	.04250	.04035	.03821	.03608

Final Temp	Initial Temperature °F													
°F	**235**	**240**	**245**	**250**	**255**	**260**	**265**	**270**	**275**	**280**	**285**	**290**	**295**	**300**
240°	.00229													
245°	.00458	.00229												
250°	.00693	.00464	.00235											
255°	.00938	.00709	.00480	.00245										
260°	.01183	.00954	.00725	.00490	.00245									
265°	.01429	.01200	.00971	.00736	.00491	.00246								
270°	.01694	.01465	.01236	.01001	.00756	.00511	.00265							
275°	.01965	.01736	.01507	.01272	.01027	.00782	.00536	.00271						
280°	.02235	.02006	.01777	.01542	.01297	.01052	.00806	.00541	.00270					
285°	.02509	.02280	.02051	.01816	.01571	.01326	.01080	.00815	.00544	.00274				
290°	.02799	.02570	.02341	.02106	.01861	.01616	.01370	.01105	.00834	.00564	.00290			
295°	.03089	.02860	.02631	.02396	.02151	.01906	.01660	.01395	.01124	.00854	.00580	.00290		
300°	.03379	.03150	.02921	.02686	.02441	.02196	.01950	.01685	.01414	.01144	.00870	.00580	.00290	

Table 6 – Acceptance factors $\left(1-\left(\frac{P_f + 14.7}{P_o + 14.7}\right)\right)$

(Use gauge pressure)

Table values are absolute pressure.

P_o Maximum Operating Pressure PSIG	P_f-Minimum operating pressure at tank (PSIG)											
	5	10	12	15	20	25	30	35	40	45	50	55
10	0.202											
12	0.262	0.075										
15	0.337	0.168	0.101									
20	0.432	0.288	0.231	0.144								
25	0.504	0.378	0.328	0.252	0.126							
27	0.527	0.408	0.360	0.288	0.168							
30	0.560	0.447	0.403	0.336	0.224	0.112						
35	0.604	0.503	0.463	0.403	0.302	0.202	0.101					
40	0.640	0.548	0.512	0.457	0.366	0.274	0.183	0.091				
45	0.670	0.586	0.553	0.503	0.419	0.335	0.251	0.168	0.084			
50	0.696	0.618	0.587	0.541	0.464	0.386	0.309	0.232	0.155	0.078		
55	0.717	0.646	0.617	0.574	0.502	0.430	0.359	0.287	0.215	0.144	0.072	
60	0.736	0.669	0.643	0.602	0.536	0.469	0.402	0.335	0.268	0.201	0.134	0.067
65	0.753	0.690	0.665	0.627	0.565	0.502	0.439	0.376	0.314	0.251	0.188	0.125
70	0.767	0.708	0.685	0.649	0.590	0.531	0.472	0.413	0.354	0.295	0.236	0.177
75	0.780	0.725	0.702	0.669	0.613	0.558	0.502	0.446	0.390	0.333	0.279	0.223
80	0.792	0.739	0.718	0.686	0.634	0.581	0.528	0.475	0.422	0.370	0.317	0.264
85	0.802	0.752	0.732	0.702	0.652	0.602	0.552	0.502	0.451	0.401	0.351	0.301
90	0.812	0.764	0.745	0.716	0.669	0.621	0.573	0.525	0.478	0.430	0.382	0.335
95	0.820	0.775	0.757	0.729	0.684	0.638	0.593	0.547	0.501	0.456	0.410	0.365
100	0.828	0.785	0.767	0.741	0.698	0.654	0.610	0.567	0.523	0.479	0.436	0.392
105	0.835	0.794	0.777	0.752	0.710	0.668	0.626	0.585	0.543	0.501	0.459	0.418
110	0.842	0.802	0.786	0.762	0.723	0.682	0.642	0.601	0.561	0.521	0.481	0.441
115	0.848	0.810	0.794	0.771	0.734	0.694	0.655	0.617	0.578	0.540	0.501	0.463
120	0.854	0.817	0.802	0.780	0.742	0.705	0.668	0.631	0.594	0.557	0.520	0.483
125	0.859	0.823	0.809	0.787	0.752	0.716	0.680	0.644	0.608	0.573	0.537	0.501
130	0.864	0.829	0.815	0.795	0.760	0.726	0.691	0.657	0.622	0.586	0.553	0.519
135	0.868	0.835	0.822	0.802	0.768	0.735	0.701	0.668	0.635	0.601	0.563	0.534
140	0.873	0.840	0.827	0.808	0.776	0.743	0.711	0.679	0.647	0.614	0.582	0.550
145	0.877	0.845	0.833	0.814	0.783	0.751	0.720	0.689	0.658	0.626	0.595	0.564
150	0.880	0.850	0.838	0.820	0.789	0.759	0.729	0.699	0.668	0.638	0.608	0.577
155	0.884	0.854	0.843	0.825	0.795	0.766	0.736	0.707	0.677	0.648	0.618	0.589
160	0.887	0.859	0.847	0.830	0.801	0.773	0.744	0.716	0.687	0.658	0.630	0.601
165	0.890	0.863	0.851	0.835	0.807	0.779	0.751	0.724	0.696	0.668	0.640	0.612
170	0.893	0.866	0.855	0.839	0.812	0.785	0.758	0.731	0.704	0.677	0.649	0.622
175	0.896	0.870	0.859	0.843	0.817	0.791	0.764	0.738	0.711	0.685	0.659	0.632
180	0.899	0.873	0.863	0.847	0.822	0.796	0.770	0.745	0.719	0.693	0.668	0.642
185	0.901	0.876	0.866	0.851	0.826	0.801	0.776	0.751	0.726	0.701	0.676	0.651
190	0.904	0.879	0.870	0.855	0.831	0.806	0.782	0.757	0.733	0.709	0.684	0.660
195	0.906	0.882	0.873	0.858	0.835	0.811	0.787	0.763	0.739	0.716	0.692	0.668
200	0.908	0.885	0.876	0.862	0.838	0.815	0.792	0.768	0.745	0.722	0.699	0.675
205	0.910	0.888	0.878	0.865	0.842	0.819	0.796	0.774	0.751	0.728	0.705	0.682
210	0.912	0.890	0.881	0.868	0.845	0.823	0.801	0.779	0.756	0.734	0.712	0.689
215	0.914	0.892	0.884	0.871	0.849	0.827	0.805	0.783	0.762	0.740	0.718	0.696
220	0.916	0.895	0.886	0.873	0.852	0.831	0.810	0.788	0.767	0.746	0.724	0.703
225	0.918	0.897	0.889	0.876	0.855	0.834	0.813	0.792	0.772	0.751	0.730	0.709
230	0.919	0.899	0.891	0.879	0.858	0.838	0.817	0.797	0.777	0.756	0.736	0.715
235	0.921	0.901	0.893	0.881	0.861	0.841	0.821	0.801	0.780	0.760	0.740	0.720
240	0.923	0.903	0.895	0.883	0.864	0.884	0.825	0.805	0.785	0.766	0.746	0.727
245	0.924	0.905	0.897	0.886	0.866	0.847	0.828	0.808	0.789	0.770	0.751	0.731
250	0.926	0.907	0.899	0.888	0.869	0.850	0.831	0.812	0.793	0.774	0.755	0.737

Continued

Table 6 – Acceptance Factors $\left(1-\left(\dfrac{P_f + 14.7}{P_o + 14.7}\right)\right)$

(Use gauge pressure)
Table values are absolute pressure.

P_o Maximum Operating Pressure PSIG	P_f-Minimum operating pressure at tank (PSIG)											
	60	**65**	**70**	**75**	**80**	**85**	**90**	**95**	**100**	**105**	**110**	**115**
60												
65	0.062											
70	0.118	0.059										
75	0.167	0.117	0.056									
80	0.211	0.158	0.106	0.053								
85	0.251	0.201	0.151	0.101	0.050							
90	0.287	0.239	0.191	0.143	0.096	0.048						
95	0.319	0.273	0.228	0.182	0.137	0.091	0.045					
100	0.347	0.305	0.261	0.218	0.174	0.131	0.087	0.043				
105	0.376	0.334	0.292	0.250	0.208	0.167	0.125	0.083	0.041			
110	0.401	0.361	0.321	0.281	0.241	0.200	0.160	0.120	0.080	0.040		
115	0.424	0.386	0.347	0.309	0.270	0.232	0.193	0.155	0.116	0.007	0.039	
120	0.446	0.408	0.371	0.334	0.297	0.260	0.223	0.186	0.149	0.111	0.074	0.037
125	0.465	0.429	0.394	0.358	0.322	0.286	0.250	0.215	0.179	0.143	0.107	0.071
130	0.484	0.450	0.415	0.381	0.346	0.312	0.277	0.243	0.208	0.173	0.138	0.104
135	0.501	0.468	0.439	0.401	0.367	0.334	0.301	0.267	0.234	0.200	0.167	0.134
140	0.517	0.485	0.453	0.420	0.388	0.356	0.324	0.291	0.259	0.226	0.194	0.162
145	0.532	0.501	0.470	0.438	0.407	0.376	0.344	0.313	0.282	0.250	0.219	0.188
150	0.547	0.517	0.486	0.456	0.426	0.396	0.365	0.335	0.305	0.273	0.243	0.213
155	0.559	0.530	0.500	0.471	0.441	0.412	0.382	0.353	0.323	0.295	0.265	0.236
160	0.573	0.544	0.515	0.487	0.458	0.430	0.401	0.372	0.344	0.315	0.286	0.258
165	0.585	0.557	0.529	0.501	0.473	0.446	0.418	0.390	0.362	0.334	0.306	0.278
170	0.595	0.568	0.541	0.514	0.487	0.460	0.433	0.460	0.378	0.352	0.325	0.298
175	0.606	0.579	0.553	0.527	0.500	0.474	0.447	0.421	0.395	0.369	0.343	0.316
180	0.616	0.590	0.565	0.539	0.513	0.488	0.462	0.436	0.411	0.385	0.360	0.334
185	0.626	0.601	0.576	0.551	0.526	0.501	0.476	0.451	0.426	0.401	0.376	0.351
190	0.635	0.611	0.587	0.562	0.538	0.513	0.489	0.465	0.440	0.415	0.391	0.366
195	0.644	0.620	0.597	0.573	0.549	0.525	0.501	0.478	0.454	0.429	0.405	0.381
200	0.652	0.629	0.605	0.582	0.559	0.535	0.512	0.489	0.466	0.443	0.419	0.396
205	0.660	0.637	0.614	0.591	0.568	0.546	0.523	0.450	0.477	0.455	0.432	0.410
210	0.667	0.645	0.622	0.600	0.578	0.556	0.533	0.510	0.489	0.467	0.445	0.423
215	0.674	0.653	0.631	0.609	0.587	0.565	0.544	0.522	0.500	0.479	0.457	0.435
220	0.682	0.660	0.639	0.618	0.597	0.575	0.554	0.533	0.511	0.490	0.469	0.447
225	0.688	0.667	0.646	0.625	0.604	0.583	0.563	0.542	0.521	0.501	0.478	0.459
230	0.695	0.675	0.654	0.634	0.613	0.593	0.573	0.552	0.532	0.511	0.490	0.470
235	0.700	0.680	0.660	0.640	0.620	0.600	0.579	0.559	0.539	0.521	0.501	0.481
240	0.707	0.687	0.668	0.648	0.629	0.609	0.589	0.570	0.550	0.530	0.510	0.491
245	0.712	0.693	0.673	0.654	0.635	0.615	0.596	0.577	0.558	0.539	0.520	0.501
250	0.718	0.699	0.680	0.661	0.642	0.623	0.604	0.585	0.566	0.548	0.529	0.510

Table 8 – Minimum pipe size from tank to system (in inches)

	Equivalent length up to 10'					Equivalent length 11' to 30'					Equivalent length 31' to 100'				
	Max. average design temp. °F					Max. average design temp. °F					Max. average design temp. °F				
MBH	**100°**	**150°**	**200°**	**250°**	**300°**	**100°**	**150°**	**200°**	**250°**	**300°**	**100°**	**150°**	**200°**	**250°**	**300°**
1000	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	3/4	3/4	1/2	3/4	3/4	3/4	1
2000	1/2	1/2	1/2	1/2	3/4	1/2	3/4	3/4	3/4	1	3/4	3/4	1	1	1 1/4
3000	1/2	1/2	3/4	3/4	3/4	3/4	3/4	1	1	1	3/4	1	1	1 1/4	1 1/4
4000	1/2	3/4	3/4	3/4	1	3/4	1	1	1	1 1/4	1	1	1 1/4	1 1/4	1 1/4
5000	1/2	3/4	3/4	1	1	3/4	1	1	1 1/4	1 1/4	1	1 1/4	1 1/4	1 1/4	1 1/2
6000	1/2	3/4	1	1	1	3/4	1	1 1/4	1 1/4	1 1/4	1	1 1/4	1 1/4	1 1/2	1 1/2
7000	3/4	1	1	1	1 1/4	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/2	2
8000	3/4	1	1	1	1 1/4	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/4	1 1/4	1 1/2	1 1/2	2
9000	3/4	1	1	1 1/4	1 1/4	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/4	1 1/4	1 1/2	2	2
10,000	3/4	1	1	1 1/4	1 1/4	1	1 1/4	1 1/4	1 1/2	1 1/2	1 1/4	1 1/2	2	2	2
12,000	1	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	2	1 1/4	1 1/2	2	2	2
14,000	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/4	1 1/4	1 1/2	2	2	1 1/4	2	2	2	2 1/2
16,000	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/4	1 1/2	1 1/2	2	2	1 1/2	2	2	2 1/2	2 1/2
18,000	1	1 1/4	1 1/4	1 1/2	1 1/2	1 1/4	1 1/2	2	2	2	1 1/2	2	2	2 1/2	2 1/2
20,000	1	1 1/4	1 1/4	1 1/2	1 1/2	1 1/4	1 1/2	2	2	2	1 1/2	2	2 1/2	2 1/2	2 1/2

Table 9 – Heat Recovery

NOTE: 1. The approximate MBH available per cycle is shown in parenthesis.
2. The outlet temperature of expanded water from the heat exchanger or heating leg is assumed to be 120°. (see correction factors)
3. System design temperature drop is assumed to be 40°F.

	Maximum rate of heat transfer – MBH						
	Maximum temperature at tank system connection in °F						
System MBH	220°	230°	240°	250°	260°	270°	280°
1000	46 (10)	53 (15)	60 (20)	67 (20)	75 (25)	83 (30)	92 (35)
2000	92 (25)	105 (30)	120 (40)	134 (45)	150 (50)	167 (60)	184 (70)
3000	138 (35)	158 (45)	180 (55)	201 (65)	225 (80)	250 (90)	276 (100)
4000	184 (50)	211 (60)	240 (75)	268 (90)	300 (100)	333 (120)	368 (140)
5000	230 (60)	264 (75)	300 (90)	335 (110)	375 (130)	415 (150)	480 (180)
6000	276 (70)	317 (90)	360 (110)	402 (130)	450 (150)	500 (180)	552 (210)
7000	322 (85)	369 (100)	420 (130)	470 (150)	525 (180)	584 (210)	644 (250)
8000	368 (95)	422 (120)	480 (150)	537 (170)	600 (210)	667 (240)	736 (280)
9000	414 (110)	475 (130)	540 (160)	604 (200)	675 (230)	750 (280)	828 (320)
10,000	460 (120)	528 (150)	600 (180)	671 (220)	750 (260)	834 (310)	920 (360)
12,000	552 (150)	633 (180)	720 (220)	805 (260)	900 (310)	1000 (370)	1104 (430)
14,000	644 (170)	739 (210)	840 (260)	939 (310)	1050 (370)	1167 (430)	1288 (500)
16,000	736 (190)	845 (240)	960 (290)	1073 (350)	1200 (420)	1334 (490)	1472 (570)
18,000	828 (220)	950 (270)	1080 (330)	1208 (400)	1350 (470)	1500 (550)	1656 (640)
20,000	920 (240)	1056 (300)	1200 (370)	1342 (440)	1500 (530)	1668 (620)	1840 (720)
CORRECTION FACTORS							
140°F	.80 (.95)	.82 (.96)	.83 (.97)	.85 (.98)	.86 (.98)	.87 (.98)	.88 (.98)
160°F	.60 (.85)	.64 (.88)	.67 (.90)	.69 (.92)	.71 (.93)	.73 (.94)	.75 (.95)
180°F	.40 (.66)	.45 (.73)	.50 (.78)	.54 (.81)	.57 (.84)	.60 (.86)	.63 (.88)
200°F	.20 (.38)	.27 (.50)	.33 (.59)	.38 (.66)	.43 (.71)	.47 (.75)	.50 (.79)
220°F	-	.09 (.19)	.17 (.33)	.23 (.44)	.29 (.53)	.33 (.60)	.38 (.65)
240°F	-	-	-	.08 (.17)	.14 (.30)	.20 (.40)	.25 (.48)

Table 12 – Solubility of Air in Water

Expressed as a fraction of water, volume, at STP

Temp (t) °F	Total Pressure, PSIG												
	0	10	20	30	40	50	60	70	80	90	100	110	120
40°	0.0258	0.0435	0.0613	0.0790	0.0967	0.1144	0.1321	0.1499	0.1676	0.1853	0.2030	0.2207	0.2384
50°	0.0223	0.0376	0.0529	0.0683	0.0836	0.0989	0.1143	0.1296	0.1449	0.1603	0.1756	0.1909	0.2063
60°	0.0197	0.0333	0.0469	0.0605	0.0742	0.0878	0.1014	0.1150	0.1296	0.1423	0.1559	0.1695	0.1831
70°	0.0177	0.0300	0.0423	0.0546	0.0669	0.0792	0.0916	0.1039	0.1162	0.1285	0.1408	0.1531	0.1654
80°	0.0161	0.0274	0.0387	0.0501	0.0614	0.0727	0.0840	0.0954	0.1067	0.1180	0.1293	0.1407	0.1520
90°	0.0147	0.0253	0.0358	0.0464	0.0569	0.0674	0.0750	0.0885	0.0990	0.1090	0.1201	0.1306	0.1412
100°	0.0136	0.0235	0.0334	0.0433	0.0532	0.0631	0.0730	0.0829	0.0928	0.1027	0.1126	0.1225	0.1324
110°	0.0126	0.0220	0.0314	0.0408	0.0501	0.0595	0.0689	0.0753	0.0877	0.0971	0.1065	0.1158	0.1252
120°	0.0117	0.0206	0.0296	0.0385	0.0475	0.0564	0.0654	0.0744	0.0833	0.0923	0.1012	0.1102	0.1191
130°	0.0107	0.0193	0.0280	0.0366	0.0452	0.0538	0.0624	0.0710	0.0796	0.0882	0.0968	0.1054	0.1140
140°	0.0098	0.0182	0.0265	0.0348	0.0432	0.0515	0.0598	0.0681	0.0765	0.0848	0.0931	0.1015	0.1098
150°	0.0089	0.0170	0.0251	0.0332	0.0413	0.0494	0.0574	0.0655	0.0736	0.0817	0.0898	0.0979	0.1060
160°	0.0079	0.0158	0.0237	0.0316	0.0395	0.0474	0.0553	0.0632	0.0711	0.0790	0.0869	0.0945	0.1027
170°	0.0068	0.0145	0.0223	0.0301	0.0378	0.0456	0.0534	0.0611	0.0689	0.0767	0.0844	0.0922	0.1000
180°	0.0055	0.0132	0.0208	0.0285	0.0361	0.0438	0.0514	0.0591	0.0667	0.0744	0.0820	0.0879	0.0973
190°	0.0041	0.0116	0.0192	0.0268	0.0344	0.0420	0.0496	0.0571	0.0647	0.0723	0.0799	0.0875	0.0950
200°	0.0024	0.0099	0.0175	0.0250	0.0326	0.0401	0.0477	0.0552	0.0628	0.0703	0.0779	0.0854	0.0930
210°	0.0004	0.0080	0.0155	0.0230	0.0306	0.0381	0.0457	0.0532	0.0607	0.0683	0.0758	0.0833	0.0909

Based on derivation by Professor Ferdinand Votta, Jr.,
Department of Chemical Engineering, University of Rhode Island

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Table 13 – Solubility of nitrogen in water

Expressed as a fraction of water volume, at STP

Table 13A Nitrogen – 79% of dry gas volume

Temp (t) °F	Total Pressure, PSIG												
	0	10	20	30	40	50	60	70	80	90	100	110	120
40°	0.0166	0.0279	0.0393	0.0506	0.0620	0.0734	0.0847	0.0961	0.1074	0.1188	0.1302	0.1415	0.1529
50°	0.0145	0.0245	0.0346	0.0446	0.0546	0.0646	0.0746	0.0846	0.0946	0.1046	0.1146	0.1246	0.1347
60°	0.0130	0.0219	0.0303	0.0399	0.0488	0.0578	0.0668	0.0757	0.0847	0.0937	0.1026	0.1116	0.1206
70°	0.0117	0.0199	0.0280	0.0362	0.0443	0.0525	0.0607	0.0688	0.0770	0.0851	0.0933	0.1015	0.1096
80°	0.0107	0.0182	0.0258	0.0333	0.0405	0.0484	0.0559	0.0634	0.0709	0.0785	0.0860	0.0935	0.1011
90°	0.0098	0.0168	0.0238	0.0308	0.0375	0.0445	0.0518	0.0587	0.0657	0.0727	0.0797	0.0867	0.0937
100°	0.0090	0.0155	0.0221	0.0286	0.0352	0.0417	0.0482	0.0548	0.0613	0.0679	0.0744	0.0809	0.0875
110°	0.0083	0.0145	0.0206	0.0268	0.0330	0.0391	0.0453	0.0515	0.0577	0.0638	0.0700	0.0761	0.0823
120°	0.0077	0.0136	0.0195	0.0254	0.0313	0.0372	0.0431	0.0490	0.0548	0.0607	0.0666	0.0725	0.0784
130°	0.0071	0.0125	0.0185	0.0241	0.0298	0.0355	0.0412	0.0469	0.0525	0.0582	0.0639	0.0696	0.0753
140°	0.0065	0.0120	0.0174	0.0229	0.0284	0.0339	0.0394	0.0449	0.0505	0.0558	0.0613	0.0668	0.0723
150°	0.0059	0.0112	0.0165	0.0219	0.0272	0.0325	0.0379	0.0432	0.0485	0.0539	0.0592	0.0645	0.0699
160°	0.0052	0.0104	0.0156	0.0209	0.0261	0.0313	0.0365	0.0418	0.0470	0.0522	0.0574	0.0627	0.0679
170°	0.0045	0.0097	0.0149	0.0200	0.0252	0.0304	0.0356	0.0408	0.0459	0.0511	0.0563	0.0615	0.0667
180°	0.0037	0.0089	0.0140	0.0192	0.0243	0.0295	0.0346	0.0398	0.0449	0.0501	0.0552	0.0604	0.0655
190°	0.0028	0.0079	0.0130	0.0182	0.0233	0.0284	0.0336	0.0387	0.0438	0.0490	0.0541	0.0592	0.0644
200°	0.0016	0.0067	0.0119	0.0170	0.0221	0.0272	0.0323	0.0374	0.0425	0.0477	0.0528	0.0579	0.0630
210°	0.0003	0.0054	0.0105	0.0156	0.0207	0.0285	0.0309	0.0360	0.0410	0.0461	0.0512	0.0563	0.0614

Table 13B Nitrogen – 100% of dry gas volume

Temp (t) °F	Total Pressure, PSIG												
	0	10	20	30	40	50	60	70	80	90	100	110	120
40°	0.0210	0.0353	0.0497	0.0641	0.0785	0.0929	0.1072	0.1216	0.1360	0.1504	0.1648	0.1791	0.1935
50°	0.0184	0.0311	0.0437	0.0564	0.0691	0.0818	0.0944	0.1071	0.1198	0.1324	0.1451	0.1578	0.1704
60°	0.0164	0.0278	0.0391	0.0505	0.0618	0.0732	0.0845	0.0959	0.1072	0.1186	0.1299	0.1413	0.1526
70°	0.0148	0.0251	0.0355	0.0458	0.0561	0.0665	0.0768	0.0871	0.0974	0.1078	0.1181	0.1284	0.1388
80°	0.0135	0.0231	0.0326	0.0421	0.0517	0.0612	0.0707	0.0803	0.0898	0.0993	0.1089	0.1184	0.1279
90°	0.0124	0.0212	0.0301	0.0390	0.0478	0.0567	0.0655	0.0744	0.0832	0.0921	0.1009	0.1098	0.1186
100°	0.0114	0.0197	0.0279	0.0362	0.0445	0.0528	0.0611	0.0693	0.0776	0.0859	0.0942	0.1025	0.1107
110°	0.0105	0.0183	0.0261	0.0339	0.0417	0.0495	0.0574	0.0652	0.0703	0.0808	0.0886	0.0964	0.1042
120°	0.0097	0.0171	0.0245	0.0319	0.0393	0.0468	0.0542	0.0616	0.0690	0.0764	0.0839	0.0913	0.0987
130°	0.0090	0.0162	0.0234	0.0305	0.0377	0.0449	0.0521	0.0593	0.0665	0.0737	0.0809	0.0881	0.0953
140°	0.0082	0.0151	0.0221	0.0290	0.0360	0.0429	0.0498	0.0568	0.0637	0.0707	0.0776	0.0846	0.0915
150°	0.0074	0.0142	0.0209	0.0277	0.0344	0.0412	0.0479	0.0547	0.0614	0.0682	0.0749	0.0817	0.0884
160°	0.0066	0.0132	0.0198	0.0264	0.0330	0.0396	0.0463	0.0529	0.0595	0.0661	0.0727	0.0793	0.0859
170°	0.0057	0.0123	0.0188	0.0254	0.0319	0.0385	0.0450	0.0516	0.0582	0.0647	0.0713	0.0778	0.0844
180°	0.0047	0.0112	0.0177	0.0243	0.0308	0.0373	0.0438	0.0503	0.0569	0.0634	0.0699	0.0764	0.0830
190°	0.0035	0.0100	0.0165	0.0230	0.0295	0.0360	0.0425	0.0490	0.0555	0.0620	0.0685	0.0750	0.0815
200°	0.0021	0.0085	0.0150	0.0215	0.0280	0.0344	0.0409	0.0474	0.0537	0.0603	0.0668	0.0733	0.0798
210°	0.0004	0.0068	0.0133	0.0197	0.0262	0.0326	0.0391	0.0455	0.0520	0.0584	0.0649	0.0713	0.0777

Based on derivation by Professor Ferdinand Votta, Jr.,
Department of Chemical Engineering, University of Rhode Island

Table 14 – Solubility of Oxygen in Water

Expressed as a fraction of water volume, at STP

Oxygen – 21% of Dry Gas Volume

Temp (t) °F	Total Pressure, PSIG												
	0	10	20	30	40	50	60	70	80	90	100	110	120
40°	0.0090	0.0151	0.0213	0.0274	0.0336	0.0397	0.0459	0.0520	0.0582	0.0644	0.0705	0.0767	0.0828
50°	0.0079	0.0133	0.0188	0.0242	0.0296	0.0351	0.0405	0.0459	0.0514	0.0568	0.0622	0.0677	0.0731
60°	0.0070	0.0118	0.0166	0.0214	0.0263	0.0311	0.0359	0.0408	0.0456	0.0504	0.0552	0.0601	0.0649
70°	0.0062	0.0106	0.0149	0.0192	0.0236	0.0279	0.0322	0.0366	0.0409	0.0453	0.0496	0.0539	0.0583
80°	0.0056	0.0095	0.0134	0.0174	0.0213	0.0252	0.0291	0.0331	0.0370	0.0409	0.0449	0.0488	0.0527
90°	0.0051	0.0087	0.0123	0.0159	0.0195	0.0231	0.0267	0.0303	0.0339	0.0376	0.0412	0.0448	0.0484
100°	0.0046	0.0080	0.0114	0.0148	0.0182	0.0215	0.0249	0.0283	0.0317	0.0350	0.0384	0.0418	0.0452
110°	0.0043	0.0074	0.0106	0.0138	0.0170	0.0202	0.0234	0.0265	0.0297	0.0329	0.0361	0.0393	0.0424
120°	0.0039	0.0069	0.0100	0.0130	0.0160	0.0190	0.0220	0.0250	0.0281	0.0311	0.0341	0.0371	0.0401
130°	0.0036	0.0065	0.0094	0.0123	0.0152	0.0180	0.0209	0.0238	0.0267	0.0296	0.0325	0.0354	0.0383
140°	0.0033	0.0061	0.0088	0.0116	0.0144	0.0172	0.0200	0.0227	0.0255	0.0283	0.0311	0.0339	0.0367
150°	0.0030	0.0056	0.0083	0.0110	0.0137	0.0164	0.0191	0.0218	0.0244	0.0271	0.0298	0.0325	0.0352
160°	0.0026	0.0052	0.0078	0.0104	0.0130	0.0156	0.0183	0.0209	0.0235	0.0261	0.0287	0.0313	0.0339
170°	0.0022	0.0048	0.0073	0.0099	0.0124	0.0150	0.0175	0.0200	0.0226	0.0251	0.0277	0.0302	0.0328
180°	0.0018	0.0043	0.0068	0.0093	0.0118	0.0143	0.0168	0.0193	0.0218	0.0243	0.0268	0.0293	0.0318
190°	0.0013	0.0038	0.0063	0.0087	0.0112	0.0137	0.0161	0.0186	0.0211	0.0235	0.0260	0.0285	0.0309
200°	0.0008	0.0032	0.0057	0.0081	0.0106	0.0130	0.0154	0.0179	0.0203	0.0228	0.0252	0.0277	0.0301
210°	0.0001	0.0026	0.0050	0.0074	0.0099	0.0123	0.0144	0.0172	0.0196	0.0220	0.0245	0.0269	0.0293

Based on derivation by Professor Ferdinand Votta, Jr.,
Department of Chemical Engineering, University of Rhode Island.

Table 15 – Net expansion of ethylene glycol mixture *

(By Volume)

50% Ethylene Glycol – 50% Water

Final Temp(t) °F	Initial Temperature (T_f) °F									
	-20°	-10°	0°	10°	20°	30°	40°	50°	60°	70°
50°	0.0145	0.0126	0.0106	0.0090	0.0066	0.0046	0.0026			
60°	0.0171	0.0151	0.0132	0.0116	0.0091	0.0071	0.0052	0.0026		
70°	0.0200	0.0181	0.0161	0.0145	0.0120	0.0100	0.0081	0.0054	0.0029	
80°	0.0226	0.0207	0.0187	0.0171	0.0146	0.0126	0.0107	0.0080	0.0054	0.0026
90°	0.0255	0.0236	0.0216	0.0200	0.0175	0.0155	0.0135	0.0109	0.0083	0.0054
100°	0.0290	0.0270	0.0250	0.0234	0.0209	0.0189	0.0169	0.0142	0.0117	0.0088
110°	0.0320	0.0301	0.0281	0.0265	0.0239	0.0219	0.0199	0.0173	0.0147	0.0118
120°	0.0358	0.0338	0.0318	0.0302	0.0277	0.0256	0.0236	0.0209	0.0183	0.0154
130°	0.0385	0.0365	0.0346	0.0329	0.0304	0.0283	0.0263	0.0236	0.0210	0.0181
140°	0.0422	0.0406	0.0383	0.0366	0.0341	0.0320	0.0300	0.0273	0.0247	0.0217
150°	0.0461	0.0441	0.0421	0.0404	0.0379	0.0358	0.0338	0.0311	0.0284	0.0255
160°	0.0499	0.0479	0.0459	0.0442	0.0417	0.0396	0.0376	0.0349	0.0322	0.0293
170°	0.0535	0.0515	0.0495	0.0478	0.0452	0.0431	0.0411	0.0384	0.0357	0.0327
180°	0.0580	0.0560	0.0540	0.0523	0.0497	0.0476	0.0456	0.0428	0.0402	0.0372
190°	0.0619	0.0599	0.0579	0.0562	0.0536	0.0515	0.0494	0.0467	0.0440	0.0410
200°	0.0659	0.0639	0.0619	0.0602	0.0576	0.0555	0.0534	0.0506	0.0479	0.0449
210°	0.0711	0.0690	0.0670	0.0652	0.0626	0.0605	0.0584	0.0556	0.0529	0.0499
220°	0.0751	0.0730	0.0709	0.0692	0.0666	0.0645	0.0624	0.0596	0.0569	0.0538
230°	0.0795	0.0774	0.0754	0.0736	0.0710	0.0688	0.0668	0.0639	0.0612	0.0582
240°	0.0844	0.0823	0.0802	0.0785	0.0758	0.0737	0.0716	0.0688	0.0660	0.0630
250°	0.0896	0.0876	0.0855	0.0837	0.0810	0.0789	0.0768	0.0739	0.0712	0.0681

Final Temp(t) °F	Initial Temperature (T_f) °F									
	80°	90°	100°	110°	120°	130°	140°	150°	160°	170°
80°										
90°	0.0028									
100°	0.0062	0.0033								
110°	0.0092	0.0063	0.0030							
120°	0.0128	0.0100	0.0066	0.0036						
130°	0.0155	0.0126	0.0092	0.0063	0.0026					
140°	0.0191	0.0162	0.0129	0.0099	0.0062	0.0036				
150°	0.0228	0.0200	0.0166	0.0135	0.0099	0.0072	0.0036			
160°	0.0266	0.0237	0.0203	0.0173	0.0136	0.0109	0.0073	0.0037		
170°	0.0301	0.0272	0.0237	0.0207	0.0170	0.0143	0.0107	0.0070	0.0034	
180°	0.0345	0.0316	0.0281	0.0251	0.0214	0.0187	0.0151	0.0114	0.0077	0.0043
190°	0.0383	0.0354	0.0319	0.0288	0.0251	0.0224	0.0188	0.0151	0.0113	0.0080
200°	0.0422	0.0393	0.0358	0.0327	0.0290	0.0263	0.0226	0.0189	0.0152	0.0118
210°	0.0472	0.0442	0.0407	0.0377	0.0339	0.0312	0.0275	0.0238	0.0200	0.0166
220°	0.0511	0.0481	0.0446	0.0415	0.0378	0.0350	0.0313	0.0276	0.0238	0.0204
230°	0.0554	0.0524	0.0489	0.0458	0.0420	0.0393	0.0356	0.0318	0.0280	0.0245
240°	0.0602	0.0572	0.0537	0.0505	0.0467	0.0440	0.0402	0.0364	0.0326	0.0292
250°	0.0653	0.0623	0.0587	0.0556	0.0518	0.0490	0.0452	0.0414	0.0376	0.0341

**Based on derivation by Professor Ferdinand Votta, Jr., Department of Chemical Engineering, University of Rhode Island.*

Table 15 – Net expansion of ethylene glycol mixture *

(By Volume)

40% Ethylene Glycol – 60% Water

Final Temp(t) °F	Initial Temperature (T_f) °F									
	-10°	0°	10°	20°	30°	40°	50°	60°	70°	80°
50°	0.0104	0.0088	0.0072	0.0056	0.0042	0.0026				
60°	0.0124	0.0107	0.0092	0.0076	0.0062	0.0045	0.0026			
70°	0.0151	0.0134	0.0118	0.0102	0.0088	0.0072	0.0049	0.0026		
80°	0.0171	0.0154	0.0138	0.0122	0.0108	0.0092	0.0069	0.0046	0.0020	
90°	0.0198	0.0181	0.0165	0.0149	0.0135	0.0118	0.0096	0.0073	0.0046	0.0026
100°	0.0227	0.0211	0.0194	0.0178	0.0165	0.0148	0.0125	0.0102	0.0075	0.0055
110°	0.0262	0.0245	0.0229	0.0213	0.0199	0.0182	0.0159	0.0136	0.0109	0.0089
120°	0.0292	0.0275	0.0259	0.0243	0.0229	0.0212	0.0189	0.0166	0.0139	0.0119
130°	0.0319	0.0302	0.0286	0.0270	0.0256	0.0239	0.0216	0.0192	0.0166	0.0146
140°	0.0358	0.0340	0.0324	0.0308	0.0294	0.0277	0.0254	0.0230	0.0203	0.0183
150°	0.0389	0.0371	0.0355	0.0339	0.0325	0.0307	0.0284	0.0261	0.0234	0.0214
160°	0.0427	0.0410	0.0394	0.0377	0.0363	0.0346	0.0323	0.0299	0.0272	0.0251
170°	0.0467	0.0450	0.0433	0.0417	0.0402	0.0385	0.0362	0.0338	0.0311	0.0290
180°	0.0506	0.0489	0.0472	0.0455	0.0441	0.0424	0.0400	0.0376	0.0349	0.0328
190°	0.0553	0.0536	0.0519	0.0502	0.0488	0.0470	0.0447	0.0423	0.0395	0.0375
200°	0.0585	0.0567	0.0551	0.0534	0.0519	0.0502	0.0478	0.0454	0.0427	0.0406
210°	0.0636	0.0619	0.0602	0.0585	0.0571	0.0553	0.0529	0.0505	0.0477	0.0456
220°	0.0677	0.0659	0.0642	0.0625	0.0610	0.0593	0.0569	0.0544	0.0517	0.0496
230°	0.0726	0.0708	0.0691	0.0674	0.0659	0.0641	0.0617	0.0593	0.0565	0.0544
240°	0.0770	0.0752	0.0735	0.0718	0.0703	0.0685	0.0661	0.0636	0.0608	0.0587
250°	0.0823	0.0805	0.0788	0.0771	0.0756	0.0738	0.0714	0.0689	0.0660	0.0639

Final Temp(t) °F	Initial Temperature (T_f) °F								
	90°	100°	110°	120°	130°	140°	150°	160°	170°
90°									
100°	0.0029								
110°	0.0062	0.0033							
120°	0.0092	0.0063	0.0030						
130°	0.0119	0.0090	0.0056	0.0026					
140°	0.0156	0.0127	0.0093	0.0063	0.0037				
150°	0.0186	0.0157	0.0123	0.0093	0.0067	0.0030			
160°	0.0224	0.0195	0.0161	0.0131	0.0104	0.0067	0.0037		
170°	0.0263	0.0233	0.0199	0.0169	0.0142	0.0105	0.0075	0.0038	
180°	0.0301	0.0271	0.0237	0.0207	0.0180	0.0142	0.0112	0.0075	0.0037
190°	0.0347	0.0317	0.0283	0.0253	0.0226	0.0188	0.0158	0.0120	0.0082
200°	0.0378	0.0348	0.0314	0.0283	0.0256	0.0218	0.0188	0.0150	0.0112
210°	0.0429	0.0398	0.0364	0.0333	0.0306	0.0268	0.0237	0.0199	0.0161
220°	0.0468	0.0438	0.0403	0.0372	0.0345	0.0306	0.0276	0.0238	0.0199
230°	0.0516	0.0485	0.0450	0.0420	0.0392	0.0354	0.0323	0.0285	0.0246
240°	0.0559	0.0529	0.0493	0.0462	0.0435	0.0396	0.0365	0.0327	0.0288
250°	0.0611	0.0580	0.0545	0.0514	0.0486	0.0447	0.0416	0.0377	0.0338

**Based on derivation by Professor Ferdinand Votta, Jr., Department of Chemical Engineering, University of Rhode Island.*

Table 15 – Net Expansion of Ethylene Glycol Mixture *

(By Volume)

30% Ethylene Glycol – 70%Water

Final Temp(t) °F	Initial Temperature (T_f) °F									
	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
50°	0.0053	0.0047	0.0029	0.0020						
60°	0.0070	0.0064	0.0046	0.0037	0.0017					
70°	0.0089	0.0083	0.0066	0.0056	0.0036	0.0019				
80°	0.0114	0.0107	0.0090	0.0080	0.0060	0.0043	0.0024			
90°	0.0133	0.0127	0.0109	0.0100	0.0080	0.0063	0.0043	0.0019		
100°	0.0161	0.0155	0.0137	0.0127	0.0107	0.0090	0.0071	0.0047	0.0027	
110°	0.0191	0.0185	0.0167	0.0157	0.0137	0.0120	0.0101	0.0076	0.0057	0.0029
120°	0.0219	0.0212	0.0195	0.0185	0.0165	0.0147	0.0128	0.0104	0.0084	0.0056
130°	0.0247	0.0240	0.0223	0.0213	0.0193	0.0175	0.0156	0.0131	0.0112	0.0084
140°	0.0278	0.0271	0.0253	0.0243	0.0223	0.0206	0.0186	0.0162	0.0142	0.0114
150°	0.0313	0.0307	0.0289	0.0279	0.0258	0.0241	0.0221	0.0197	0.0177	0.0149
160°	0.0344	0.0338	0.0320	0.0310	0.0289	0.0272	0.0252	0.0228	0.0208	0.0180
170°	0.0383	0.0377	0.0359	0.0349	0.0328	0.0311	0.0291	0.0266	0.0246	0.0218
180°	0.0422	0.0416	0.0397	0.0387	0.0367	0.0349	0.0329	0.0304	0.0284	0.0256
190°	0.0465	0.0458	0.0440	0.0430	0.0409	0.0392	0.0372	0.0347	0.0326	0.0298
200°	0.0502	0.0496	0.0477	0.0467	0.0446	0.0428	0.0408	0.0383	0.0363	0.0335
210°	0.0545	0.0539	0.0520	0.0510	0.0489	0.0471	0.0451	0.0426	0.0405	0.0377
220°	0.0586	0.0579	0.0561	0.0551	0.0530	0.0512	0.0491	0.0466	0.0446	0.0417
230°	0.0638	0.0631	0.0612	0.0602	0.0581	0.0563	0.0542	0.0517	0.0496	0.0468
240°	0.0680	0.0673	0.0654	0.0644	0.0623	0.0604	0.0584	0.0558	0.0538	0.0509
250°	0.0733	0.0726	0.0707	0.0697	0.0675	0.0657	0.0636	0.0611	0.0590	0.0561

Final Temp(t) °F	Initial Temperature (T_f) °F						
	110°	120°	130°	140°	150°	160°	170°
110°							
120°	0.0027						
130°	0.0054	0.0027					
140°	0.0085	0.0058	0.0030				
150°	0.0119	0.0092	0.0064	0.0034			
160°	0.0150	0.0123	0.0095	0.0065	0.0030		
170°	0.0188	0.0161	0.0133	0.0103	0.0068	0.0038	
180°	0.0226	0.0198	0.0170	0.0140	0.0105	0.0075	0.0037
190°	0.0268	0.0240	0.0212	0.0181	0.0147	0.0116	0.0078
200°	0.0304	0.0276	0.0248	0.0217	0.0182	0.0152	0.0114
210°	0.0346	0.0318	0.0290	0.0259	0.0224	0.0193	0.0155
220°	0.0386	0.0358	0.0330	0.0299	0.0263	0.0232	0.0194
230°	0.0437	0.0409	0.0380	0.0349	0.0313	0.0282	0.0243
240°	0.0478	0.0450	0.0421	0.0390	0.0354	0.0322	0.0284
250°	0.0530	0.0501	0.0472	0.0441	0.0405	0.0373	0.0334

**Based on derivation by Professor Ferdinand Votta, Jr., Department of Chemical Engineering, University of Rhode Island.*

Table 15 – Net expansion of ethylene glycol mixture *

(By Volume)

20% Ethylene Glycol – 80% Water

Final Temp(t) °F	Initial Temperature (T_f) °F									
	20°	30°	40°	50°	60°	70°	80°	90°	100°	110°
50°	0.0035	0.0024	0.0017							
60°	0.0044	0.0034	0.0026	0.0010						
70°	0.0062	0.0051	0.0044	0.0027	0.0017					
80°	0.0081	0.0071	0.0064	0.0047	0.0037	0.0020				
90°	0.0099	0.0089	0.0081	0.0064	0.0054	0.0037	0.0017			
100°	0.0118	0.0108	0.0100	0.0083	0.0074	0.0056	0.0036	0.0019		
110°	0.0147	0.0136	0.0128	0.0111	0.0102	0.0084	0.0064	0.0047	0.0028	
120°	0.0174	0.0164	0.0156	0.0139	0.0129	0.0112	0.0092	0.0074	0.0055	0.0027
130°	0.0202	0.0192	0.0184	0.0167	0.0157	0.0140	0.0119	0.0102	0.0083	0.0055
140°	0.0230	0.0220	0.0212	0.0195	0.0185	0.0167	0.0147	0.0129	0.0110	0.0082
150°	0.0261	0.0250	0.0243	0.0225	0.0216	0.0198	0.0178	0.0160	0.0141	0.0113
160°	0.0297	0.0286	0.0279	0.0261	0.0251	0.0234	0.0213	0.0196	0.0176	0.0148
170°	0.0329	0.0318	0.0310	0.0293	0.0283	0.0265	0.0245	0.0227	0.0207	0.0179
180°	0.0369	0.0358	0.0350	0.0332	0.0323	0.0305	0.0284	0.0266	0.0247	0.0218
190°	0.0409	0.0398	0.0390	0.0372	0.0362	0.0344	0.0324	0.0306	0.0286	0.0258
200°	0.0449	0.0438	0.0430	0.0412	0.0402	0.0384	0.0364	0.0346	0.0326	0.0297
210°	0.0489	0.0478	0.0470	0.0452	0.0442	0.0424	0.0403	0.0385	0.0365	0.0337
220°	0.0530	0.0519	0.0511	0.0493	0.0483	0.0465	0.0444	0.0426	0.0406	0.0377
230°	0.0575	0.0564	0.0556	0.0538	0.0528	0.0509	0.0489	0.0470	0.0450	0.0421
240°	0.0624	0.0613	0.0605	0.0587	0.0577	0.0558	0.0537	0.0519	0.0499	0.0469
250°	0.0669	0.0658	0.0650	0.0632	0.0621	0.0603	0.0582	0.0563	0.0543	0.0513

Final Temp(t) °F	Initial Temperature (T_f) °F					
	120°	130°	140°	150°	160°	170°
120°						
130°	0.0028					
140°	0.0055	0.0027				
150°	0.0085	0.0057	0.0030			
160°	0.0121	0.0093	0.0065	0.0035		
170°	0.0151	0.0123	0.0096	0.0066	0.0030	
180°	0.0191	0.0162	0.0135	0.0104	0.0069	0.0039
190°	0.0230	0.0202	0.0174	0.0143	0.0108	0.0077
200°	0.0269	0.0241	0.0213	0.0182	0.0147	0.0116
210°	0.0309	0.0280	0.0252	0.0221	0.0186	0.0155
220°	0.0349	0.0320	0.0292	0.0261	0.0225	0.0194
230°	0.0393	0.0364	0.0336	0.0305	0.0269	0.0237
240°	0.0441	0.0412	0.0384	0.0352	0.0316	0.0285
250°	0.0485	0.0456	0.0427	0.0396	0.0360	0.0328

**Based on derivation by Professor Ferdinand Votta, Jr., Department of Chemical Engineering, University of Rhode Island.*

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