YORK®

ABSORPTION CHILLERS

APPLICATIONS GUIDE

Supersedes: Nothing

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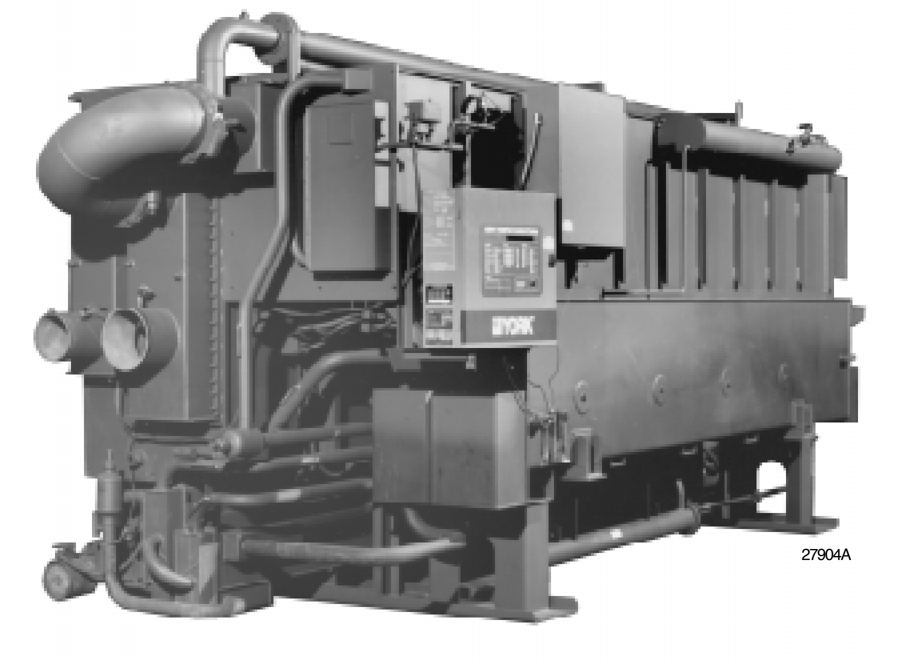

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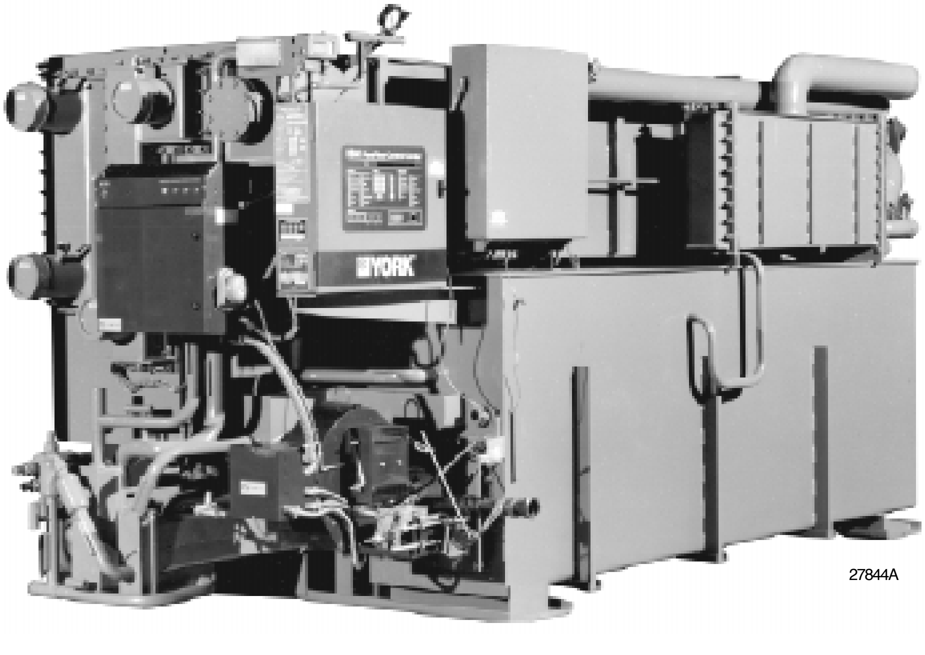

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A BRIEF HISTORY OF ABSORPTION CHILLER TECHNOLOGY

One of the earliest ice-building machines was an absorption machine patented by Ferdinand Carre in 1859; preceding the patent of the first reciprocating compressor by thirty years. Carre's machines produced between 20 and 220 pounds of ice. These ice-machines were sometimes over two stories high and required a great deal of maintenance. Manufacturer's found that these designs were too large and very expensive.

It was not until the middle of the 20th century that the big three U.S. chiller manufacturers developed and fine tuned the ammonia absorption chillers for use in industrial and commercial applications. These chillers are very similar to the single-stage absorption chillers that are commercially available today that use low pressure steam or waste process heat to generate cooling.

Twenty years after the commercial introduction of the absorption chillers, the U.S. market saw a dramatic increase in the cost of natural gas and petroleum products due to some misguided regulations that resulted in the "Energy Crisis". Coincidentally, interest in gas-fired cooling was increasing in Japan. Japanese manufacturers licensed the absorption chiller designs from the U.S. companies. The Japanese proceeded to improve upon the U.S. design, resulting in the introduction of the two-stage absorption chiller. In the early 90's, the U.S. companies licensed the two-stage designs from the Japanese. The U.S. companies have been working on improving the Japanese designs.

Today, while economics is an ever-present market driver for absorption applications, environmental issues have taken an increasing role in the HVAC&R decision making process. Decision makers are no longer basing their designs on the life of the building, but the effect of their decisions decades, and even centuries, into the future. With tighter regulations from both legislative bodies and building codes and the current actions to deregulate the electric industry, the building designer and operator must carefully balance the advantages and disadvantages of high efficiency and economical applications with environmental issues.

With global interest in absorption technology on the rise, there are a number of questions that must be addressed by the building designers, contractors and engineers when designing HVAC systems that include absorption chillers. It is the goal of this manual to address those questions and bring some of the major issues to the attention of the building designers, contractors and engineers.

This guide is set up to address the applications and questions that are likely to arise with any absorption application. The table of contents, glossary and index will help the designer navigate their way through this manual to find the solutions to many of their questions.

HOW IT WORKS

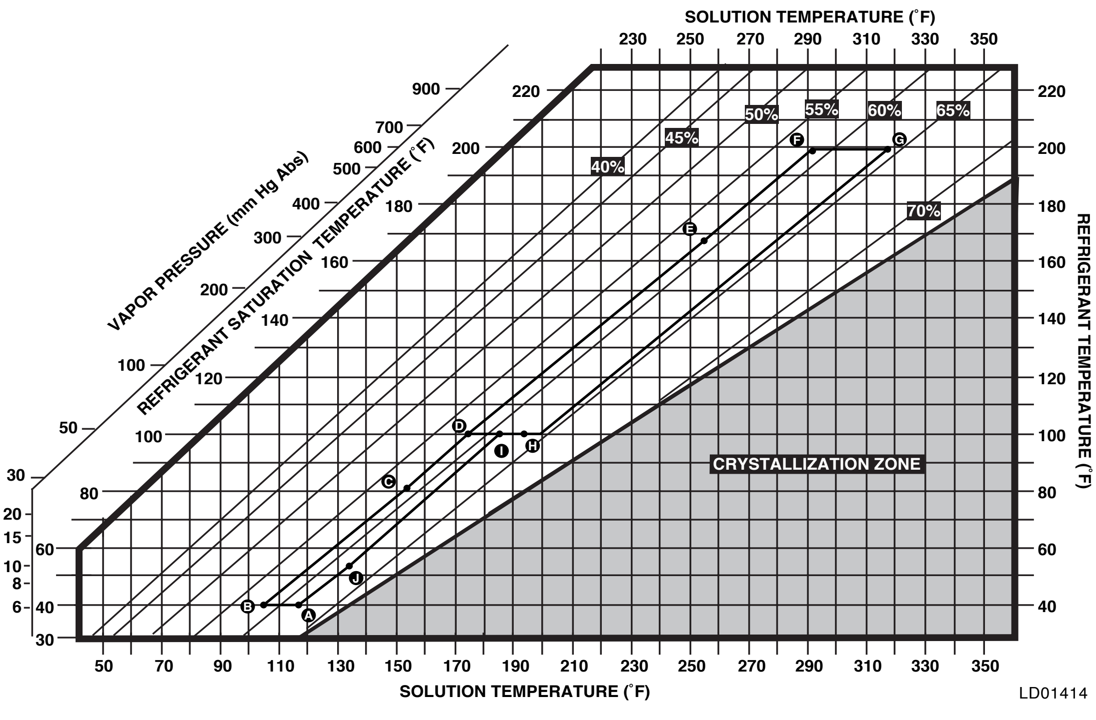
YORK YPC absorption chillers use a parallel-flow design that allows the chiller to operate at lower concentration levels. The two-stage absorption cycle on the following pages depicts the cycle of the YORK YPC two-stage absorption chillers.

PRESSURE-TEMPERATURE-CONCENTRATION CHART

The relationship between lithium bromide and water is defined by the pressure-temperature-concentration (PTX) chart. The PTX chart in Fig. 1 shows the solution temperature along the horizontal axis, the refrigerant saturation temperature and pressure along the vertical axis, and solution concentration lines diagonally on the chart. At the bottom right-hand side of the chart is the crystallization area where the concentration of the lithium bromide exceeds the solubility of the solution.

The concentration of the lithium bromide solution can be determined by measuring the solution temperature and refrigerant temperature (or refrigerant pressure as marked on the diagonal axis along the upper left side of the chart).

The plot shown on Fig. 1 below depicts the typical conditions of the lithium bromide solution through a two-stage absorption chiller.



A. 61.6% – Absorption of Refrigerant Vapor in Absorber Begins
B. 59% – Approx 106°F-Solution Leaves the Absorber-To Heat-X
C. 59% – Approx 154°F-Solution Leaves MT Heat-X-Enters LTG
D. 59% – Approx. 175°F-Refrigerant Generation Begins in LTG
E. 59% – Approx 257°F-Solution Leaves HT Heat-X-Enters HTG
F. 59% – Approx 292°F-Refrigerant Generation Begins in HTG
G. 64.5% – Approx 318°F-Solution Leaves HTG
H. 63.4% – Solution from LTG and HTG Mix
I. 61.6% – Solution from LTG and HTG Mix with 59% Eductor Flow
J. 61.6% – Approx 136°F-Solution Enters the Absorber Sprays

FIG. 1 – PRESSURE – TEMPERATURE – CONCENTRATION (PTX) CHART OF AQUEOUS LIBR SOLUTION

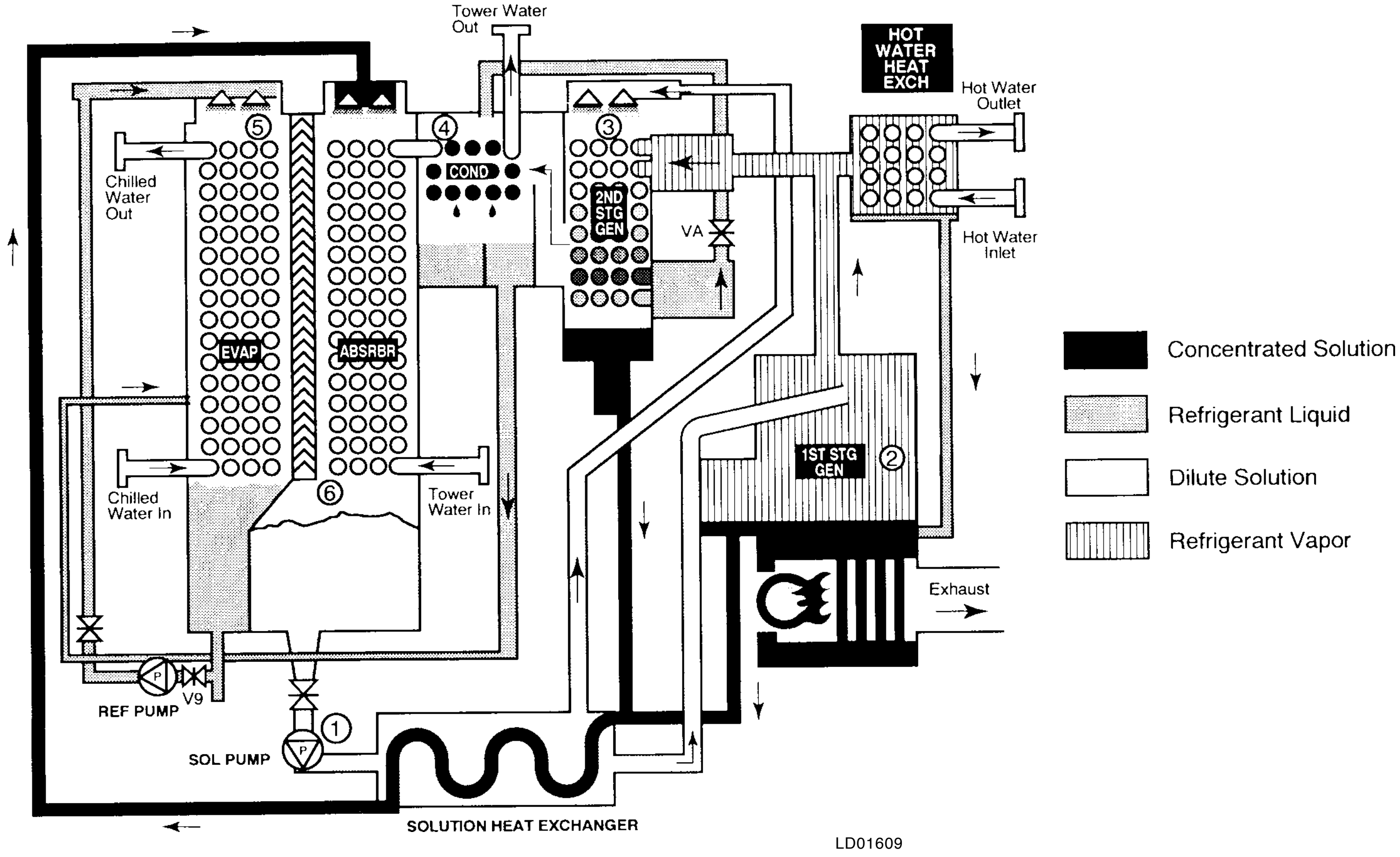


FIG. 2 – COOLING CYCLE

The two-stage absorption refrigeration cycle uses water as the refrigerant and lithium bromide as the absorbent. It is the strong affinity these two substances have for each other that makes the cycle work. The entire process occurs in hermetic vessels in an almost complete vacuum.

Fig. 2 above indicates the complete chilling cycle. The six steps are detailed below, with corresponding numbers in the diagram to show where each step takes place. YPC's two-stage absorption chilling cycle is continuous; however, for the sake of clarity and simplicity, it is divided into six steps.

1. Solution Pump/Heat Exchangers

A dilute solution of lithium bromide and water descends from the Absorber to the Solution Pump. This flow of dilute solution is split into two streams and pumped through heat exchangers to the First-Stage Generator and to the Second-Stage Generator.

YORK's two-stage chillers split the solution flow to eliminate the possibility of crystallization (solidification) by allowing the unit to operate at much lower solution concentration and temperatures than series flow systems.

Note: There are two heat exchangers, but only one is shown for illustrative purposes.

2. First-Stage Generator

A heat source heats dilute lithium bromide coming from the Solution Pump/Heat Exchangers. This produces hot refrigerant vapor which is sent to the Second-Stage Generator, leaving a concentrated solution that is returned to the Heat Exchangers.

3. Second-Stage Generator

The heat source for the production of refrigerant vapor in the Second-Stage Generator is the hot refrigerant vapor produced by the First-Stage Generator.

This is the heart of YORK's remarkably efficient two-stage absorption effect. The refrigerant vapor produced in the First-Stage Generator is increased by 40% – at no additional expense of fuel. The result is much higher efficiency than in conventional systems.

This additional refrigerant vapor is produced when dilute solution from the Heat Exchanger is heated by refrigerant vapor from the First-Stage Generator. The additional concentrated solution that results is returned to the Heat Exchanger. The refrigerant vapor from the First-Stage Generator condenses into liquid giving up its heat, and continues to the Condenser.

4. Condenser

Refrigerant from two sources – (1) liquid resulting from the condensing of vapor produced in the First-Stage Generator and (2) vapor produced by the Second-Stage Generator – enters the Condenser. As the liquid refrigerant enters the low pressure of the condenser it flashes to vapor. The two sources of refrigerant vapor combine and condense to liquid as they are cooled by the condenser water. The liquid then flows down to the Evaporator.

5. Evaporator

Refrigerant liquid from the Condenser passes through a metering valve and flows down to the Refrigerant Pump, where it is pumped up to the top of the Evaporator. Here the liquid is sprayed out as a fine mist over the Evaporator tubes. Due to the extreme vacuum (6 mm Hg) in the Evaporator, some of the refrigerant liquid vaporizes, creating the refrigerant effect. (This vacuum is created by hydroscopic action – the strong affinity lithium bromide has for water – in the Absorber directly below.)

The refrigerant effect cools the returning system chilled water in the Evaporator tubes. The refrigerant liquid/vapor picks up the heat of the returning chilled water, cooling it from 54°F to 44°F (12.2°C to 6.7°C). The chilled water is then supplied back to the system.

6. Absorber

As the refrigerant liquid/vapor descends to the Absorber from the Evaporator, a concentrated solution coming from the Heat Exchanger is sprayed out into the flow of descending refrigerant. The hydroscopic action between lithium bromide and water-and the related changes in concentration and temperature-result in the creation of an extreme vacuum in the Evaporator directly above. The dissolving of the lithium bromide in water gives off heat, which is removed by condenser water entering from the cooling tower at 85°F (29.4°C) and leaving for the Condenser at 92°F (33.3°C). The resultant dilute lithium bromide solution collects in the bottom of the Absorber, where it flows down to the Solution Pump.

The chilling cycle is now completed and begins again at Step 1.

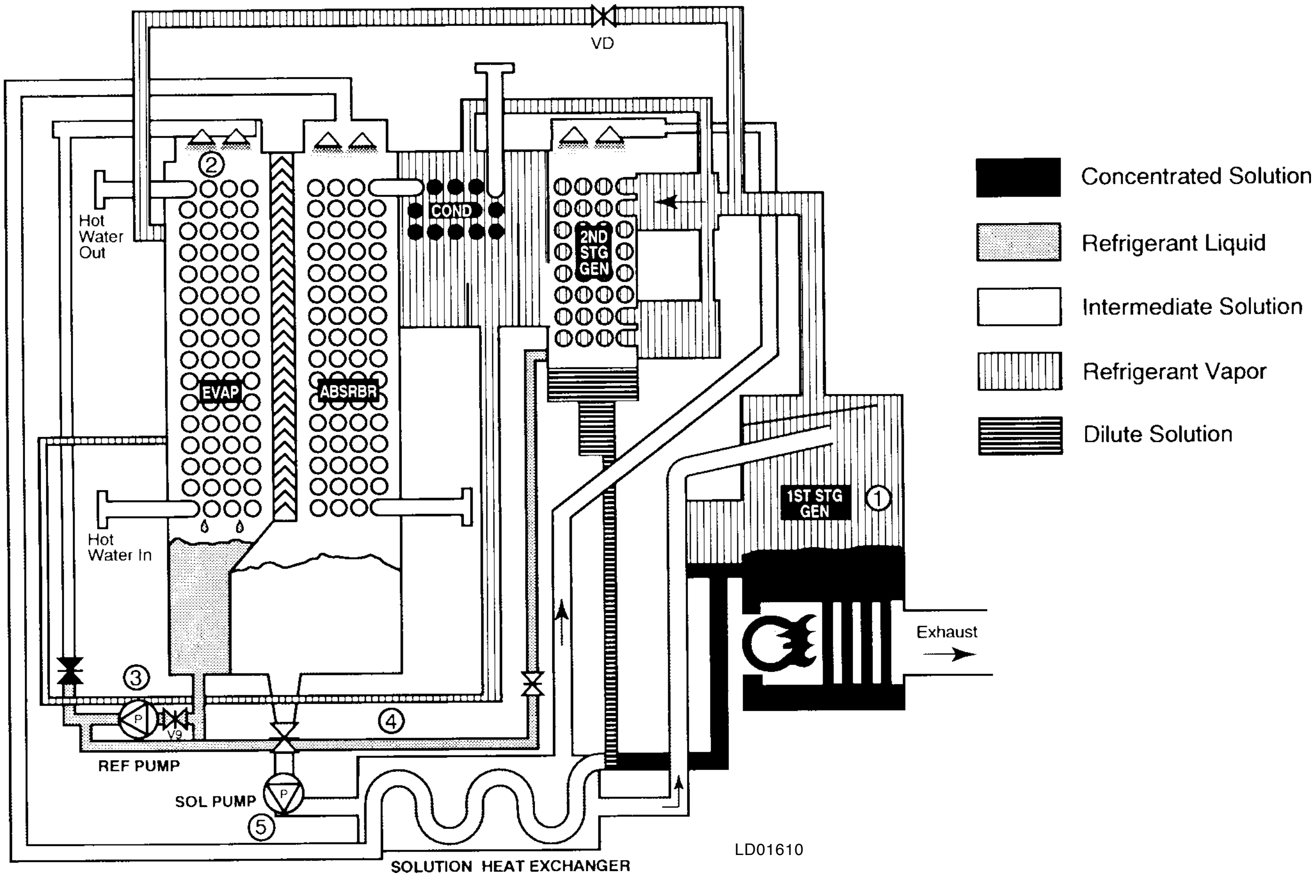


FIG. 3 – HEATING CYCLE WITHOUT HOT WATER HEAT EXCHANGER (TWO-STAGE DIRECT-FIRED ONLY)

1. First-Stage Generator

An energy source heats dilute lithium bromide solution in the First-Stage generator. This produces hot refrigerant vapor which travels through the open valve VD and also moves through the Second-Stage generator and Condenser to the evaporator.

2. Evaporator

The hot refrigerant vapor travels through the evaporator where it gives up heat to the system hot water, and changes state from vapor to liquid as its heat is removed. The refrigerant liquid collects in the bottom of the evaporator.

3. Refrigerant Pump

The refrigerant pump removes the refrigerant liquid from the evaporator and pumps it into the second stage generator where it mixes with intermediate lithium bromide solution to form dilute lithium bromide solution.

4. Heat Exchangers/Absorber

The dilute lithium bromide flows to the solution heat exchanger where it combines with the concentrated solution from the First Stage generator to form intermediate solution. The intermediate solution then flows through the absorber.

5. Solution Pump/Heat Exchangers

The solution pump moves the intermediate solution from the absorber through the solution heat exchangers and into the first and second stage generators and the process begins again.

HEATING CYCLE WITH OPTIONAL HOT WATER HEAT EXCHANGER

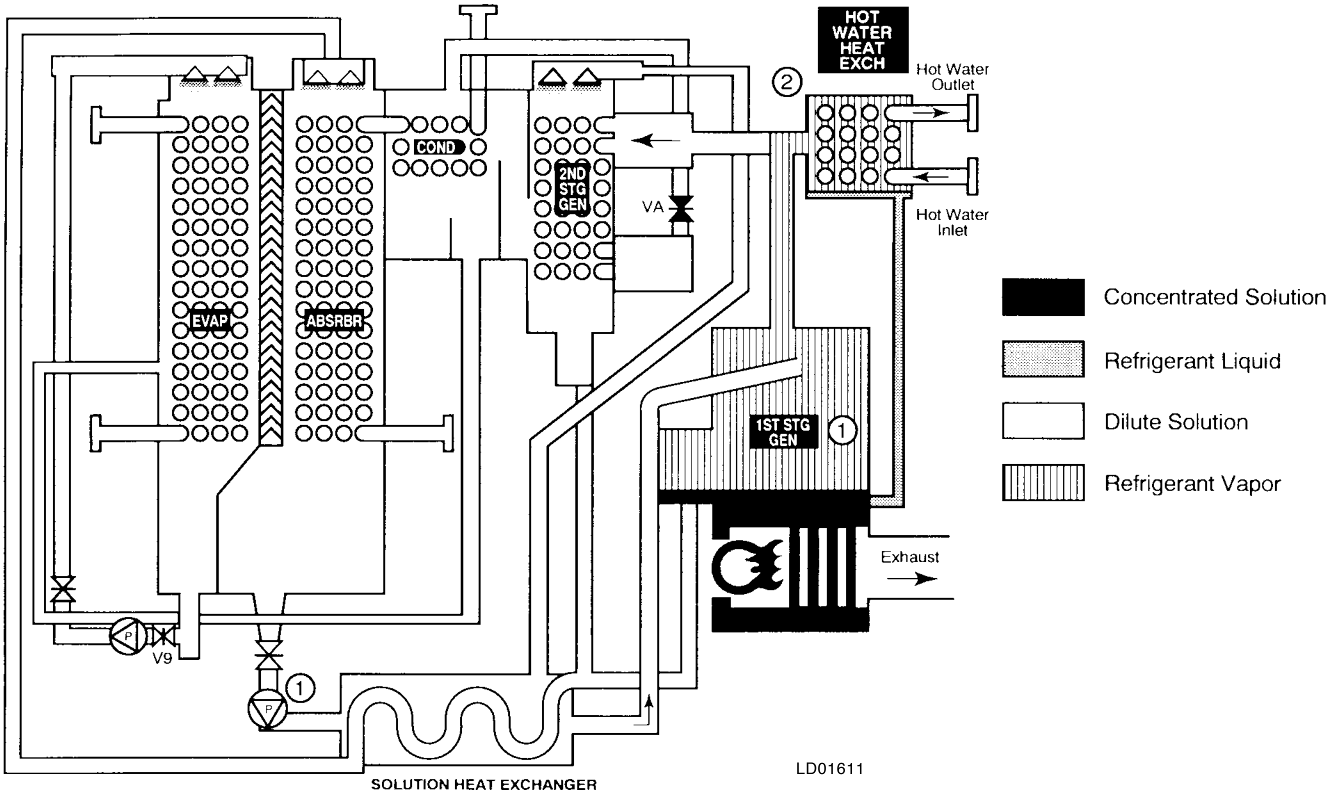


FIG. 4 – HEATING CYCLE WITH OPTIONAL HOT WATER HEAT EXCHANGER (TWO-STAGE DIRECT-FIRED ONLY)

1. First-Stage Generator

Dilute lithium bromide solution is heated by the energy source in the First-Stage Generator. This drives off refrigerant vapor and leaves concentrated solution.

2. Hot Water Heat Exchanger

The hot refrigerant vapor from the first stage generator gives up its heat to the System Hot Water Heat Exchanger and condenses to liquid as the heat is removed. The condensed refrigerant liquid returns to the First-Stage Generator and the cycle begins again.

CHILLER EVALUATION

One of the single most important factors in all purchasing decisions is the economic evaluation of the project. There are a number of reference and calculation tools available to the building designer that will help calculate the total costs involved with a project, whether they be life-cycle costs, operating costs, maintenance costs or real-world energy analyses to name a few. The problem with determining an accurate cost evaluation can be very difficult at times due to the many factors that can effect the results.

Keeping with the intent of this manual, to provide a general understanding of chiller evaluation, the following section provides an explanation of the COP calculation, IPLV, and real world energy performance. These analysis tools enable the design engineer to better evaluate different chiller and mechanical room equipment and systems.

COEFFICIENT OF PERFORMANCE

The Coefficient of Performance (COP) of a chiller is dependent upon the quality of the initial fuel resources. The purpose of this section is to explain why the chiller COP is not a valid means of comparing an electric chiller with a gas-fired chiller.

COP is calculated using the equation shown in Fig. 5.

$$COP = \frac{Q_{output}}{Q_{input}} = \frac{12{,}000\ \frac{Btu}{Ton}\ q_{cooling}}{q_{heat\ input}}$$

where:

Q_{output} = the effective heat output from the chiller in Btu's

Q_{input} = the effective heat input to the chiller in Btu's

$q_{cooling}$ = the capacity of the chiller in tons of refrigeration

$q_{heat\ input}$ = the heat input to the chiller from the thermal heat source (i.e. burner, steam) in Btus.

C_1 = factor to convert from refrigerant tons to Btu's of heat

FIG. 5 – COP EQUATION

As shown in Table 1, the COP of an electric chiller appears much higher than a natural-gas-fired absorption chiller. This difference can be explained by comparing the level of refinement of the natural resource.

TABLE 1 – TYPICAL COP'S OF CHILLERS

Type of Chiller	COP
Electric Centrifugal	6.0
Indirect-fired (Steam) Two-stage Absorption Chiller	1.2
Steam Single-Stage Absorption Chiller	0.68
Direct-Fired Two-Stage Absorption Chiler	1.0

Example: Steam-fired absorption chiller COP calculation. 1000 ton chiller operating at 9.7 lb/ton*hr with 115 PSIG steam, ARI nominal conditions.

$$COP = \frac{q_{out}}{q_{in}} = \frac{12 \times q_{out}}{m \times (h1-h2)}$$

where:

q_{out} = chiller cooling capacity in tons

m = mass flow of steam in lb/hr

h_1 = enthalpy of entering steam

h_2 = enthalpy of leaving condensate

The enthalpies of the steam must be read from saturated steam tables. For 115 PSIG, the enthalpy of the steam (h_1) is approximately 1192 BTU/hr. The enthalpy of the leaving condensate (h_2), hot water at 180°F, 15 PSIG, is approximated at 148 BTU/lb. Now, substituting the figures into the COP equation:

$$COP = \frac{(12{,}000\ BTU/ton)(1000\ tons)}{(9700\ lb./ton*hr.)(1192-148\ BTU/lb.)} = 1.18$$

Except for hydroelectric and nuclear power, most electric power is still produced from fossil fuel burning generators. As Fig. 6 shows, the electric power supplied to the chiller has incurred losses in the production, transmission, and application of power.

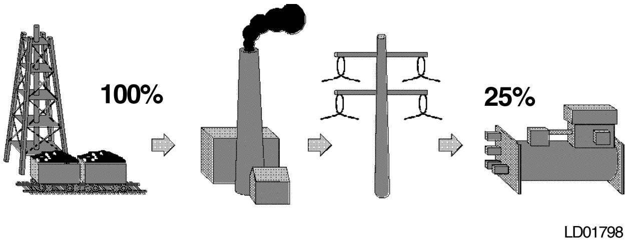


FIG. 6 – ELECTRIC POWER LOSSES BETWEEN THE WELL-HEAD AND THE CUSTOMER'S SITE

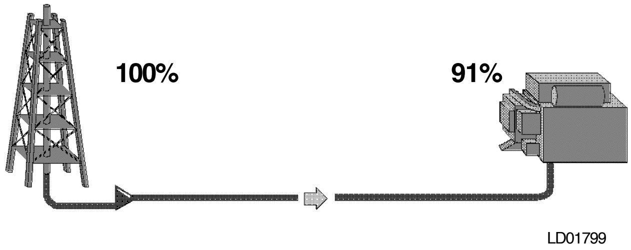


FIG. 7 – GAS FUEL LOSSES BETWEEN THE WELL-HEAD AND THE CUSTOMER'S SITE

A natural gas burning application as shown in Fig. 7 shows the losses associated with the transmission and final combustion of the fossil fuel. The losses associated with the pumping and combustion are small in comparison to the overall energy losses from the electric application.

The COP of the electrical chiller is higher because the chiller requires a highly refined power supply. The initial fossil fuel has been converted from its crude form from the ground to a high quality power source. When comparing COP's the electric chiller COP is calculated based on the input to the chiller, not the input from the ground. Those losses are usually passed on to the customer from the utility companies in the form of higher rates.

A similar comparison can be made between the two-stage and single-stage absorption chillers. The two-stage absorption chiller requires a higher steam pressure and subsequently a higher steam temperature than the single-stage chiller. The difference in the steam temperature and pressure accounts for the 1.2 COP: an increase in the efficiency of about 75% over the single-stage chiller.

In conclusion, the COP of a chiller can remain a useful index to compare similar chillers, however other methods must be employed in order to make an accurate comparison between fossil fuel burning applications and electric powered applications.

INTEGRATED PART LOAD VALUE

The next level of chiller performance evaluation is the Integrated Part Load Value (IPLV). The IPLV calculation was developed to evaluate the performance of the chiller based on four load efficiency levels because experience has shown that the chiller operates at full load for only a fraction of its annual operating hours.

The IPLV calculation is based on four operating ranges: full load, 75%-, 50%-, and 25%-full load. IPLV is calculated using a weighted average of the COPs of the chiller operation at the four index load levels. Fig. 8 defines the equation and components.

$$\text{IPLV} = \frac{1}{\frac{0.17}{A} + \frac{0.39}{B} + \frac{0.33}{C} + \frac{0.11}{D}}$$

where: A = COP at 100% of full load

B = COP at 75% of full load

C = COP at 50% of full load

D = COP at 25% of full load

FIG. 8 – IPLV EQUATION

In order to calculate the IPLV of a chiller, the part load COP's must first be calculated using the equation in Fig. 5.

REAL-WORLD ENERGY PERFORMANCE

Recently, the perspective of chiller-plant energy performance has broadened to encompass the energy-cost impact of ***off-design*** conditions. These conditions exist whenever the building cooling demand is less than full cooling capacity and/or whenever the ECWT is lower than its design value. This wider view also takes into account the energy consumption of auxiliary chiller-plant components: the cooling tower, the chilled-water pump, and the condenser-water pump.

Real-World Energy Performance (REP) is based on existing weather data and actual chiller performance characteristics. Weather data can be obtained from the Department of Defense (DoD), ASHRAE publications, or weather data centers. By using this REP analysis, it can be shown that the chiller only operates at design conditions for one or two percent of its annual operating hours.

Fig. 9 shows historical weather data for Washington D.C. The vertical bars represent the total number of operating hours for a chiller at different temperatures or "bins." For example, a chiller operates at only seven (7) hours annually with dry bulb temperatures of 95°F to 100°F; 82 hours annually with dry bulb temperatures of 90°F to 94°F, etc. By comparing the operating hours at different dry bulb temperatures, it is shown that the chiller actually operates at design full-load for a small fraction of the total annual operating hours.

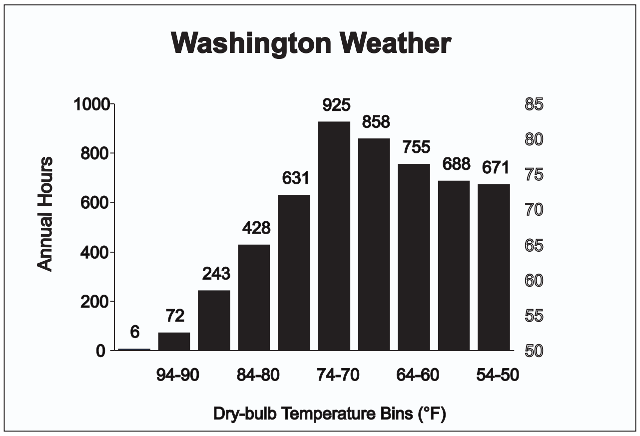


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FIG. 9 – WASHINGTON D.C. WEATHER DATA - COMPILED BY THE U.S. DEPT OF DEFENSE

The graph in Fig. 10 shows the typical cooling tower water temperature available based on tabulated weather data. This calculation assumes a constant speed cooling tower fan. Note that full load operation accounts for less than 2% of annual operating hours.

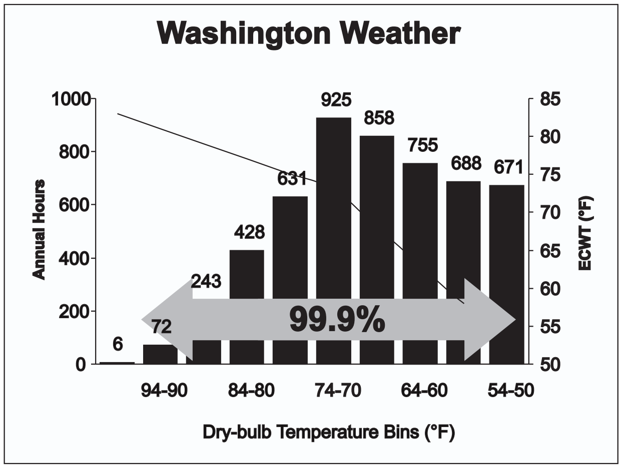

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FIG. 10 – WASHINGTON D.C. WEATHER DATA WITH TOWER WATER TEMPERATURE OVERLAY

The task of deciding which type of chiller to install can often be very difficult and time consuming. We have shown you three different performance evaluation calculations available for determining which chiller or size to choose. It is recommended that an in depth analysis be used when making very important economic and design decisions.

CHILLER ECONOMIC ANALYSIS

A number of factors may influence HVAC projects and not the least of them is the economics of the project. The economics of a cooling project are rarely simple. In order for an economic analysis to be complete, all factors that influence the cost of the chiller and its operation must be considered, including purchase price of the chiller and equipment, gas and electric rates, additonal rate fees such as demand, ratchet charges, etc., operating hours, and weather data.

When making design decisions based on theoretical or calculated data, remember that the data is most often historic and can change over time. It is good practice to build in as much flexibility to the overall design to help offset possible hidden costs and fluctuations in weather and utility rates.

Commercially available software programs perform chiller and HVAC system economic analyses. Many of these software packages include input for typical region or city weather data and utility rates to help the designer get a clearer picture of the total costs of the project.

Now that this manual has explained what an absorption chiller is and how it works as well as a simple means of evaluating chiller performance, the remaining sections will address application considerations specific to absorption chiller applications.

It will help the reader to first think of an absorption chiller in a manner that they may be more familiar with; as a boiler. By thinking of the applications that are particular to a boiler, many of the “special” considerations for absorption chiller applications are addressed. For example, a direct-fired absorber uses a burner, requires a flue for the exhaust gases and requires special consideration for ASHRAE 15. A similar argument may be made for steam-fired absorption chillers. A high-pressure steam absorber may require special pressure vessel codes and piping considerations.

Additional application considerations and features are described in the following sections. Most are common to all absorption chillers, however there are a few features that apply only to YORK absorption chillers. Where applicable, detailed drawings and schematics are provided, or referenced.

MECHANICAL ROOM DESIGN AND BUILDING CODES

Many locations have adopted one of the prevailing model codes; however, these codes may vary slightly from one region to another. It is essential that the building designer understands the importance of all codes and how they apply to their application. All applicable codes must be investigated during the initial phases of a projects design because they may severely impact the overall economics of the installation.

Currently, the Building Officials and Code Administrators International, Inc. (BOCA), International Conference of Building Officials, Inc. (ICBO) and the Southern Building Code Congress International, Inc. are collaborating on an international standards code, the International Mechanical Code (IMC) in an attempt to set a worldwide building code standard.

At the time of this publication, the IMC is still in its draft phase and is not expected to be published for some time. BOCA has already adopted a resolution endorsing “the International Mechanical Code and its successor documents as the recommended codes pertaining to mechanical regulations and [urging] its adoption by governmental units and agencies throughout the United States and the World when available by the International Code Council.”

It is good practice to consider possible future requirements in all building designs. The ANSI/ASHRAE 15 mechanical room design safety standard has been partially adopted into the first chapter of the latest revision of the IMC.

Note, the IMC has not (as of this printing) been published, however it is the current trend towards governing building design standards.

ANSI/ASHRAE 15

ASHRAE 15 is a mechanical safety code for mechanical room and building designs. The scope of the ASHRAE 15-1994 code states that air conditioning and refrigeration equipment using water (lithium bromide absorption chiller/air conditioning equipment) as a refrigerant are exempt from the code, however, direct-fired absorption chiller/heaters require the same considerations as a boiler or similar combustion application.

In accordance with ASHRAE 15, direct-fired absorption chiller/heater applications must be installed in a separate equipment room from air conditioning/refrigeration equipment using classified refrigerants unless one of the following two exceptions apply:

1. The combustion air is ducted from outside the machinery room and sealed in such a manner as to prevent any refrigerant leakage from entering the combustion chamber, or
2. a refrigerant vapor detector is employed to automatically shut down the combustion process in the event of refrigerant leakage.

Due to the increasing demand for safety and environmental protection, the codes may change over the course of an HVAC project. At the time of this printing, a revision to the ASHRAE code is being considered that will include absorption equipment in the scope of the code. Contact ASHRAE for the most recent version of the code. For further information regarding mechanical room safety considerations, refer to the local, state or federal building code agencies.

EMISSIONS

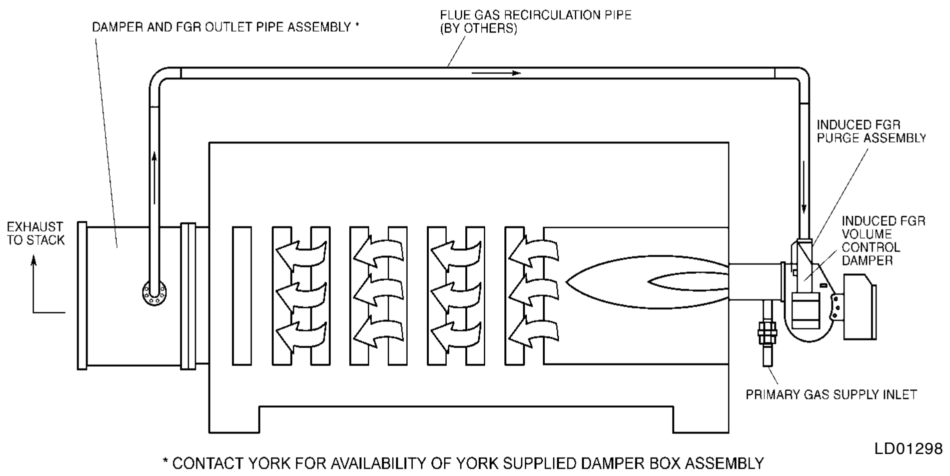
Any combustion application may need an emissions permit depending on ruling local, state, and federal regulations. Some regulations are more stringent than others therefore it is important to consult the state or local air pollution agency for details. The local gas utility may also be a good source of information.

Despite varying levels of emissions controls, YORK YPC absorption chillers operate in some of the most stringent of emissions control zones. Where required, YORK offers low NOx burners designed to meet the lowest emissions levels.

LOW NOx BURNERS

Flue Gas Recirculation (FGR)

YORK’s optional low NOx burner uses an induced flue gas recirculation design (IFGR) on the PowerFlame burners in order to meet low NOx requirements. IFGR can reduce NOx levels as much as 75%. The diagram in Fig. 11 shows a typical IFGR arrangement.



* CONTACT YORK FOR AVAILABILITY OF YORK SUPPLIED DAMPER BOX ASSEMBLY

FIG. 11 – LOW NOx BURNER CONFIGURATION – INDUCED FLUE GAS RECIRCULATION (INDUCED FGR, IFGR)

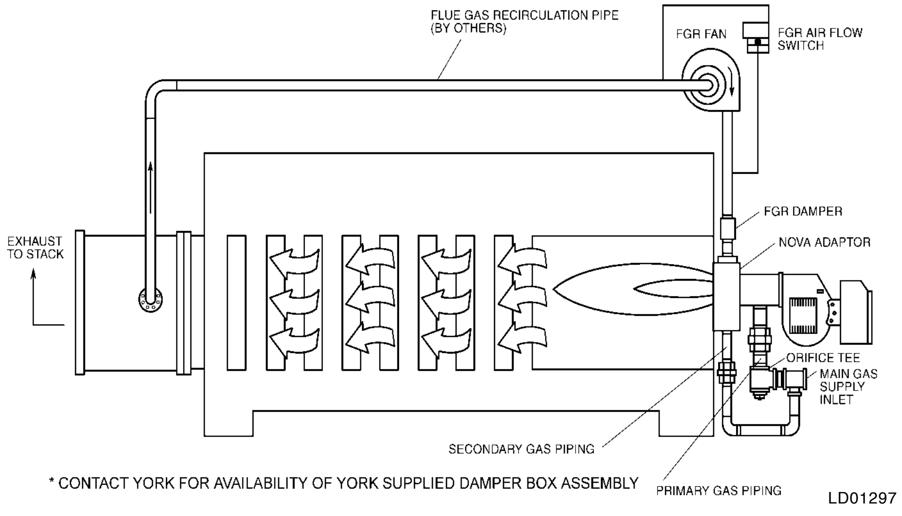
Products of combustion are returned to the flame envelope, reducing both temperature and available oxygen. Incoming air is diluted and the oxygen concentration in the combustion zone is reduced. Approximately 30% of the total flue gas can be recirculated. Higher levels yield unstable flames, high CO levels, and reduced efficiency and capacity.

The installation of FGR low NOx burner equipment requires a field connection between the stack and the burner blower as shown in Fig. 11 above. This connection can be made with seamless schedule 10 steel pipe.

For special applications, YORK can provide a forced flue gas recirculation design (FFGR) for low NOx requirements. The word “forced” implies that a fan in the flue gas recirculation piping to force a quantity of exhaust flow back to the combustion chamber. A typical FFGR arrangement is shown in Fig. 12 below.

With FFGR, a flow switch must be installed in the flue gas recirculation pipe to provide safety lockout of the burner in the event that the FGR blower motor shuts down.

Table 2 shows typical flue product emissions for YORK Direct-Fired Absorption Chillers using natural gas (approximately 1020 Btu/cu. ft.) and #2 fuel oil. Low NOx burner NOx emissions are located at the bottom of the table.



* CONTACT YORK FOR AVAILABILITY OF YORK SUPPLIED DAMPER BOX ASSEMBLY

FIG. 12 – LOW NOx BURNER CONFIGURATION – FORCED FLUE GAS RECIRCULATION (FORCED FGR, FFGR)

TABLE 2 – TYPICAL FLUE PRODUCT EMISSIONS

FLUE GAS EMISSION PRODUCT	EMISSION LEVEL	
	Natural Gas	#2 Fuel Oil
Carbon Monoxide – CO	0.037 lb CO per 10^6 Btu input (50 ppm)	Negligible
Sulfur Dioxide – SO_2	(1.05) x (% Sulfur by weight in fuel) = lb. SO_2 per 10^6 Btu input	
Particulate Matter	Negligible	0.10 lb. PM per 10^6 Btu input
Hydrocarbons	0.025 lb. HC's per 10^6 Btu input	0.038 lb. HC's per 10^6 Btu input
Nitrogen Oxides – NOx Standard Burner	0.092 lb. NOx per 10^6 Btu input (75 ppm)	0.14 lb. NOx per 10^6 Btu input (110 ppm)
Nitrogen Oxides – NOx Low NOx burner	0.031 lb. NOx per 10^6 Btu input (25 ppm)	0.073 lb. NOx per 10^6 Btu input (60 ppm)

LEGISLATION

For discussion purposes, this manual uses codes and legislation of the United States. The Clean Air Act Amendments of 1990 are the driving force in air quality regulation in the United States. They apply to all major air pollution sources with a potential to emit 100 tons per year of any regulated pollutant, any source subject to acid rain, air toxins, Prevention of Significant Deterioration Regulations (PSD), or any source the Environmental Protection Agency (EPA) determines to be applicable by rule making.

PERMITTING

Registrations mandated by the Clean Air Act establish minimum elements of a permit program to be administered by the state or local air pollution control agency. A permit is an authorization to operate combustion driven applications on a specific site. The elements of the permit must include:

1. A standard application form and process
2. Monitoring and reporting requirements
3. A permit fee
4. Provisions for adequate personnel and funding
5. A fixed permit term, not to exceed 5 years
6. Emissions limitations and standards
7. Frequency for submitting compliance certifications (not less than annually)
8. Methods for determining compliance
9. Procedures for permit renewal

Under the amendments each state is required to establish and submit to the EPA a State Implementation Plan. This guarantees that each state will have different requirements, specific to their individual needs.

An example permitting flow chart is shown in Fig. 13.

NON-ATTAINMENT AREAS

In 1991 the EPA designated a number of non-attainment areas and a number of different non-attainment classifications. There are five non-attainment classifications. Each is determined by the Ozone Design Value measured in PPM. Table 3 below summarizes non-attainment classifications and the emissions requirement for new installations in each area.

If a new source is classified as a major source for a given non-attainment status, a New Source Review (NSR) is required by local air quality agencies. The result of this review is typically the requirement for lowest achievable emissions reduction (LAER) technologies. In all cases, classification as a major source should be avoided whenever possible to prevent NSR review.

TABLE 3 – NOX EMISSIONS NON-ATTAINMENT CLASSIFICATION AND SAMPLING

NO. OF AREAS	41	32	18	8	1
Sampling of areas	Columbus Dayton Indianapolis Kansas City Phoenix Pittsburgh Tulsa	Cincinnati Cleveland Dallas Detroit Miami Richmond San Francisco	Atlanta Boston El Paso Fresno Baton Rouge Bakersfield Washington	Baltimore Chicago Houston Milwaukee New York Philadelphia San Diego	Los Angeles
Major source NOx (tons/yr.) new source	100	100	50	25	10
Allowable Net Increase (tons/yr.)	40	40	25	25	NONE

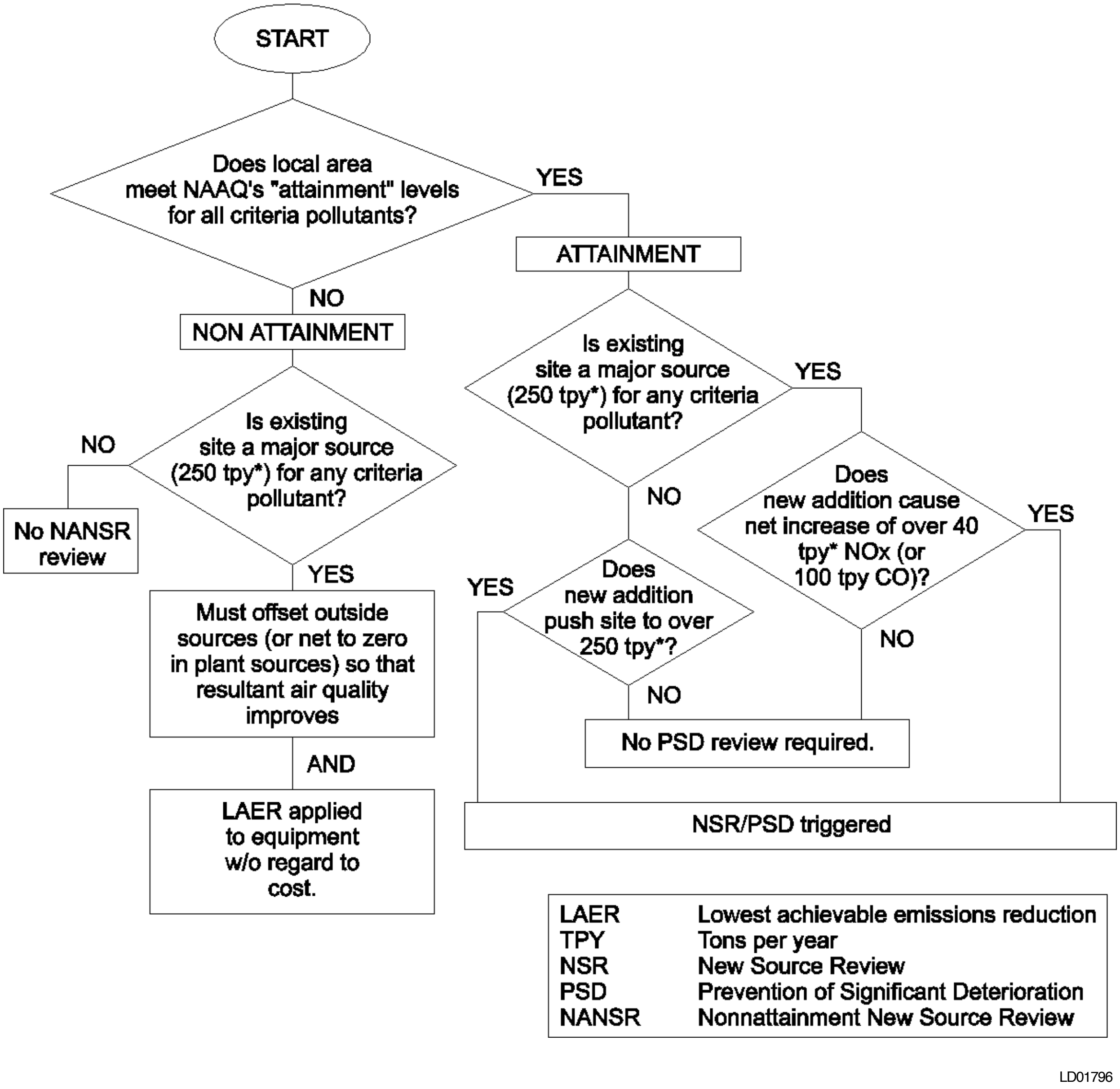


FIG. 13 – EXAMPLE PERMITTING FLOW SHEET FOR COMBUSTION APPLICATIONS

AIR SUPPLY AND CHIMNEY SYSTEM

AIR SUPPLY

As with any natural gas or No. 2 oil-fired appliance, an adequate combustion air supply must be provided for proper and complete combustion. As a general rule, 12 scf (standard cubic feet) of air are required for every 1000 Btu of fuel burned, assuming combustion with 20% excess air.

In addition to direct combustion air, outside air is also needed for ventilation and as make up to any machine room air which is drawn up the chimney through a barometric draft regulator.

Air supply openings of adequate size must be provided to the machine room in accordance with local codes and standards. The specific location and size should be sufficient to allow an unobstructed flow of fresh air to the burner. Under no circumstances should the static pressure in the mechanical room become negative (below atmospheric pressure).

Contacts are provided in the burner panel (supplied mounted on the YORK chiller-heater) to tie in the opening and closing of fresh air dampers into the start/stop sequence of the chiller. If the fresh air dampers do not open within 60 seconds, the Control Panel will shut the chiller down, preventing an unsafe operating condition.

CHIMNEY DRAFT THEORY

Draft control serves two important functions for YORK Direct-Fired chiller-heaters: 1. It removes the combustion products from the living or work space, and 2. Minimizes excess draft which pulls useful heat out of the unit and lowers its efficiency.

Draft depends on two important factors:

1. The temperature difference between the flue gas and the outside air, and
2. The height of the chimney. Higher temperature differences between the flue gas and the outside air, and higher chimneys both create more draft.

Chimney draft is the force created by the difference in temperature between the flue gases and the outside ambient air. The magnitude of this temperature difference is directly proportional to the draft created. Temperature difference causes draft because gases, such as air, occupy different volumes at different temperatures. For example: One cubic foot of air weighs 0.0834 pounds (37.8 g) at a temperature of 0°F (-18°C). This same cubic foot of air at 450°F (232°C) weighs only 0.0422 lb. (19.4 g). The amount of mass per specific volume is referred to as density. Density decreases as temperature increases and lighter (lower density) air rises while heavier, more dense air sinks. This is the same phenomena that leads to air stratification in buildings. The temperature in a room at the thermostat may be 70°F but the temperature near the ceiling may be 80°F.

Heated combustion gases, being less dense than the cooler outside air, rise and flow out of the top of the chimney and create a partial vacuum. This causes a negative pressure at the chimney inlet that pulls in more gas for venting. This pulling force is referred to as chimney draft.

Because YORK direct-fired chiller-heaters are capable of operating in both the heating and cooling modes, the outdoor air temperatures will change significantly from the summer (cooling season) to the winter (heating season). These wide temperature swings must be accounted for during system design. The larger the temperature difference the greater the draft. Therefore, when the unit is operating during the colder months more draft will be produced. It is essential that the chimney system be designed using summer ambient conditions so as to avoid undersizing the draft system.

CHIMNEY DESIGN THEORY

The following is a discussion of the basic terms and approach used in chimney design. It is not the intent of this section to address the fine details of proper chimney design. Because of the large number of variables, this must be addressed on an application specific basis by an experienced designer knowledgeable in chimney systems, draft control, and local code requirements.

Theoretical Draft (Dt) – The definition of Theoretical Draft is the natural draft or “Chimney Effect” produced by difference in densities of hot exhaust gas relative to cooler ambient air.

Available Draft (Da) – The Available Draft is the draft required at the outlet exhaust flange of the YORK High Temperature Generator.

Pressure Drop (dP) – Frictional losses in the chimney system which act against theoretical draft. The chimney draft needed to overcome chimney frictional losses is described as follows:

$$Dt = dP + Da$$

Proper chimney design balances the theoretical draft (Dt) against the pressure drop (dP) of the chimney system in order to provide the required available pressure(Da) at the outlet of the chiller-heater.

Proper chimney design must provide the required Da under all operating conditions. Because the difference between summer and winter ambient conditions can result in Dt variations of 50% and greater, some method of draft control is usually required in order to maintain Da.

CHIMNEY APPLICATION

The YORK direct-fired chiller/heater is equipped with a forced draft burner capable of firing on a variety of fuels, including natural gas and/or No. 2 oil and/or propane. As such, the unit will require a properly designed chimney system to control draft and discharge flue gases from the unit to the atmosphere.

The combustion chamber of the chiller-heater is engineered to produce a positive gauge pressure of 0.05 to 0.15 in. of water at the outlet of the first stage generator with an exhaust temperature of 400°F+/-50°F. As such the chimney must be designed to maintain that pressure at all ambient conditions. Because YORK chiller-heaters operate at "high fire" throughout the summer months, it is important to design the chimney system for summer ambient design conditions to avoid undersizing. It is recommended that the chimney itself should be designed for a Da of 0 (zero) in. of water column. This will prevent the chimney from becoming pressurized at any point along the flue gas path.

CAUTION: If the stack/chimney pressure is ever above 0 (zero) in. of water column there is a chance of flue gas being leaked into the equipment room.

There are two commonly used ways to maintain the pressure at the outlet of the chiller-heater. Either manual or automatic/motorized draft control can be used. All YORK direct-fired chiller-heaters will come standard with a manual draft control damper. This damper can modified for motorized operation either in the factory (if ordered) or in the field if site conditions require.

MANUAL DRAFT CONTROL

Manual draft control is suitable for applications where each gas fired appliance will have its own dedicated chimney and draft control system. Fig. 14 depicts a simple yet effective means of controlling draft. With this system, a (field supplied) barometric draft regulator is used in series with the factory supplied manual backdraft damper. With maximum economy employed in the chimney design, Dt would exactly equal dP + Da during the summer design ambient conditions with the barometric regulator closed. In reality, some degree of conservatism should exist in the design, causing the barometric regulator to be open slightly even during summer design conditions. As ambient temperatures drop, Dt would increase, if not for the barometric draft regulator. With the regulator in place, mechanical room air is introduced into the chimney system in response to the impressed draft, thus stabilizing the gauge pressure just upstream of the barometric regulator. Most barometric regulators can maintain 0.06 in. water column gauge pressure when properly sized for a particular application.

With the gauge pressure stabilized upstream of the barometric regulator, the factory supplied manual backdraft damper can be adjusted to a fixed position which will provide the pressure drop to yield the required positive pressure at the exhaust flange of the chiller-heater.

MOTORIZED DRAFT CONTROL

Motorized draft control is suitable for applications where multiple gas fired appliances will be ducted into one common chimney system. In this case each unit will require its own draft control system. (Motorized draft control may be used for one chiller-heater/one chimney applications as well if it is desired over manual control.) Fig. 15 depicts a sequential draft control system which incorporates a motorized damper whose position is automatically adjusted as a function of available draft at the outlet of the chiller. The YORK supplied backdraft damper at the outlet of the chiller can be modified (in the factory if ordered, or in the field) to mount the motor driver. The motor is controlled from a draft control panel which senses the pressure at the outlet of the chiller-heater. The draft control panel is available from YORK to ship with the chiller. The panel is wired to the burner panel and damper motor in the field, and the pressure is sensed through a small line field-connected to the outlet of the chiller-heater.

SPECIAL PROBLEMS AND MAINTENANCE

Factors causing draft variations during normal operation include: wind and weather factors, inadequate chimney construction or system installation, location of installation, or inadequate system maintenance.

WIND AND WEATHER

Windy conditions will tend to increase the draft in the chimney as the wind helps to remove the combustion products leaving the chimney at a much faster rate. Down draft may occur causing a temporary positive pressure in the chimney system. The stack should be designed to prevent not only wind, but rain and snow from entering the stack. A flue cap should be installed.

INADEQUATE SYSTEM INSTALLATION

If the diameters of the chimney system are too restrictive, the combustion products and flue gases may not be allowed to leave the system. On the other hand if the flue passages are too large, the chimney is never given a chance to completely warm due to the large surface area of the flue. This situation may cause poor draft and flue gas condensation which can corrode the chimney. To allow the chimney system to heat up faster, insulation should be installed on all exposed flue piping. Insulation is also a good safety measure (often required by code) as the breaching and flue pipes will heat to temperatures in excess of 400°F (204°C).

LOCATION OF INSTALLATION

Consideration should be given to the location of the stack in comparison to building intake and exhaust vents, cooling towers, etc. The effect of wind patterns around a building can create surface pressures and eddy currents that could lead to draft problems or contamination of other systems.

INADEQUATE SYSTEM MAINTENANCE

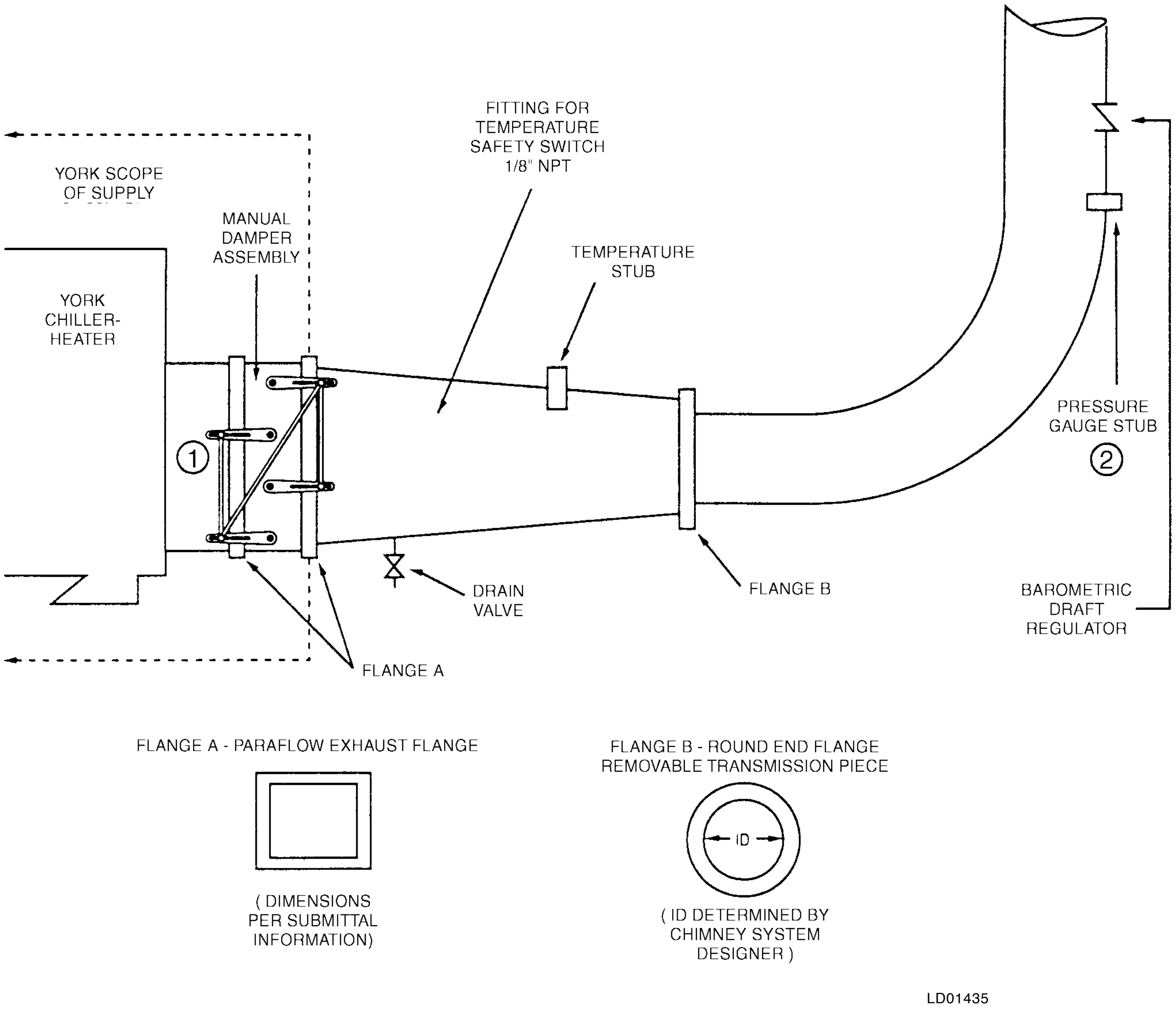
Inadequate system maintenance may lead to burner sooting. If left unattended for long periods of time, the flue passages can become restricted reducing the draft. Flue gas should be sampled on a regular basis to check for proper fuel/air ratios.

FOR MORE INFORMATION

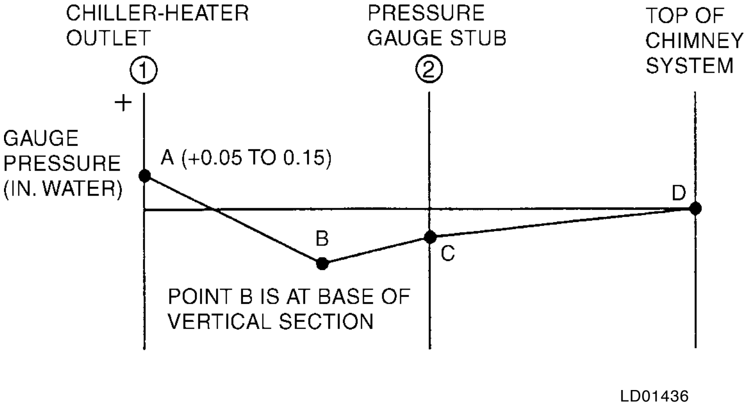
For information regarding chimney and breaching design procedures, refer to ASHRAE 1992 Systems and Equipment Handbook, Chapter 31 and the National Fuel gas code (NFPA 541992). For information regarding the effects of airflow around buildings refer to ASHRAE 1993 Fundamentals Handbook chapter 14.

Also, independent companies exist which design and supply stack materials. These companies have engineering programs to design chimneys for specific application considerations. When contacting such a company expect to need certain information including: Fuel type and consumption, design ambient temp. (it is best to design for the hottest summer day), flue gas temp., expected height of stack, number of expected fittings (elbows, T's, etc.).

FIG. 14 – MANUAL/BAROMETRIC DRAFT



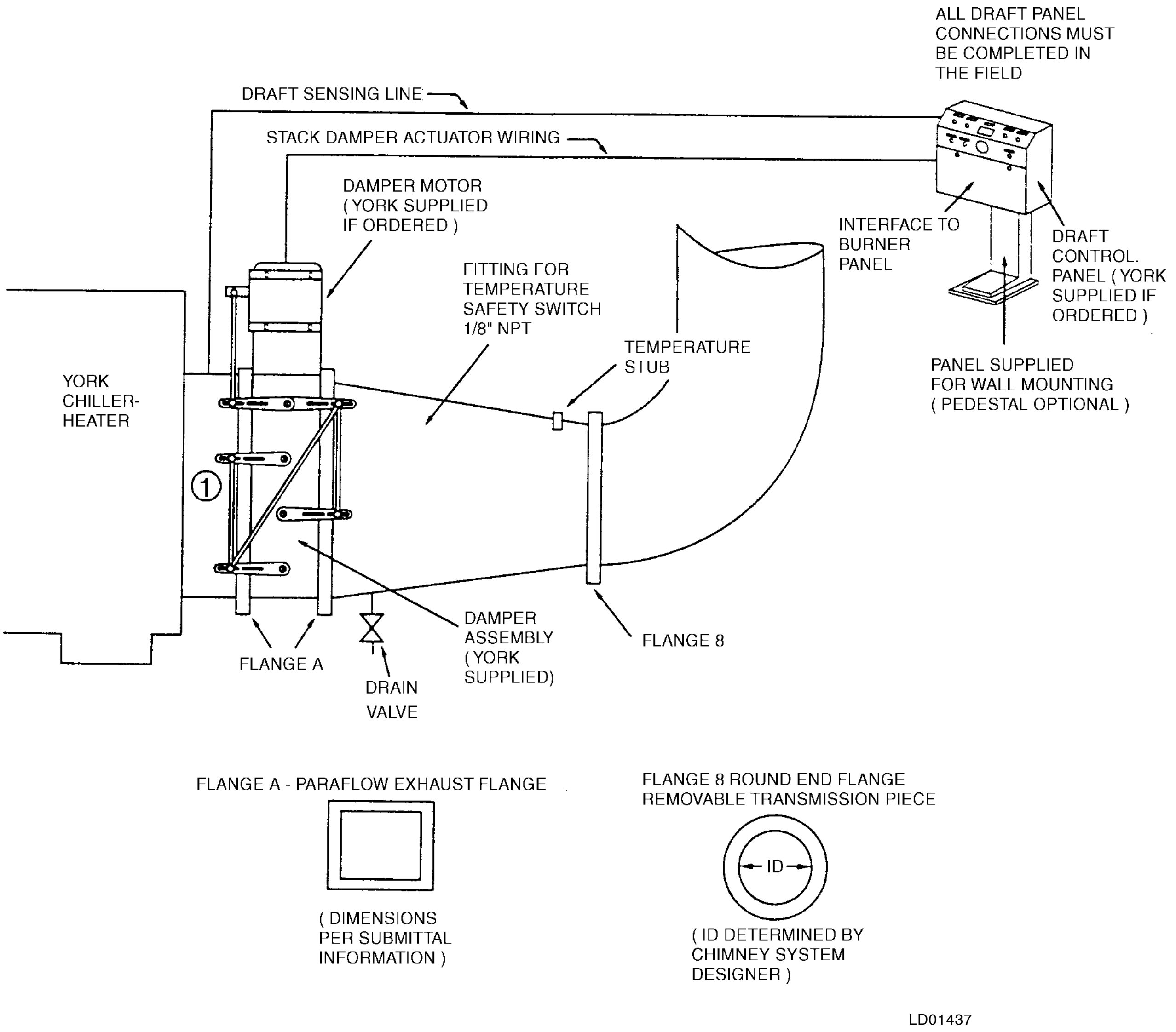
Gauge Pressure Profile – Chimney System with Barometric Control



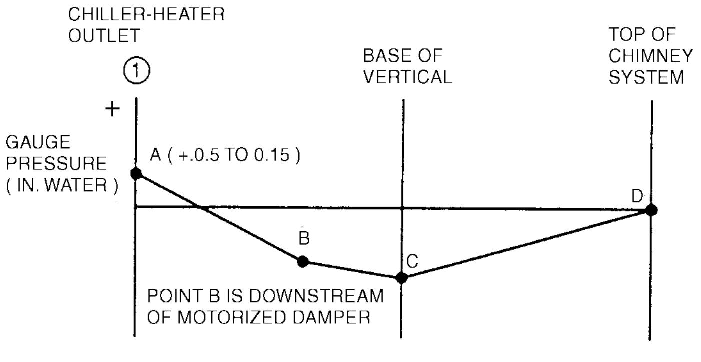
NOTES:

1. dP between A and B due to transition piece and properly positioned backdraft damper. Damper adjusted to maintain +0.05" to +0.15" water at A.
2. Maximum draft (minimum gauge pressure) occurs at base of vertical section of chimney (B). Barometric regulator will maintain steady gauge pressure at C. Maximum gauge pressure attainable with a barometric draft regulator is typically -0.06" water.

FIG. 15 – MOTORIZED DRAFT CONTROL



Gauge Pressure Profile – Chimney System with Motorized Draft Control



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NOTES:

1. dP between A and B due to transition piece and motorized damper. Damper automatically controls to maintain +0.05" to 0.15" water at A. The actual dP is variable and depends on the momentary gauge pressure C.
2. Maximum draft (minimum gauge pressure) occurs at base of vertical section of chimney (C). With sequential draft control, this value is allowed to drift with prevailing ambient conditions. Motorized damper controls to maintain steady gauge pressure at A.

STEAM

General – The YPC unit is nominally rated for dry steam with minimal superheat, and a pressure of 115 PSIG (8 bar) (at the steam valve). The inlet steam must not have a temperature higher than 363°F (184°C) and cannot have a saturation pressure higher than 128 PSIG (8.8 bar).

Pressure – Since a lower steam saturation pressure corresponds to a lower temperature in the generator, a YPC chiller's available capacity varies with the steam pressures at the steam valve. Fig. 16 graphically depicts the relationship between saturation pressure and available capacity.

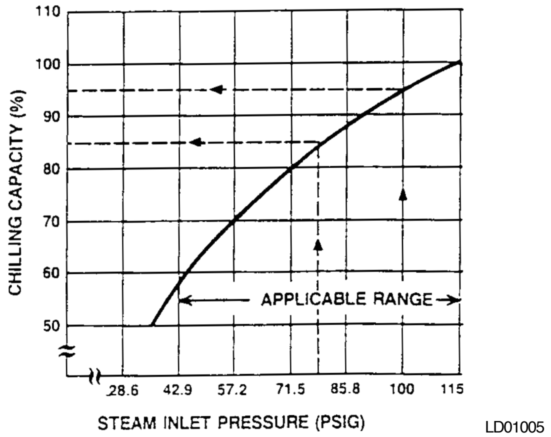

FIG. 16 – AVAILABLE COOLING CAPACITY AS A FUNCTION OF INLET STEAM PRESSURE

Piping – The steam piping must follow the diagram found in Fig. 17. Beginning at the steam supply, the automatic shutoff valve (by others) must be failsafe and is designed to protect the unit in case of a power failure. A steam separator should be installed after the automatic shutoff valve to ensure the unit receives only dry steam. A steam trap below the separator will allow draining of only condensate. Beyond the separator, a 50# mesh strainer removes foreign matter from the steam. A regulator is only necessary if the steam supply pressure to the unit will exceed 128 PSIG (8.8 bar). The pressure relief valve should be set to open at 128 PSIG (8.8 bar). The YORK supplied steam control valve must be no further than 200 inches (5 m) from the first-stage generator steam inlet flange in order to minimize the pressure drop from the valve exit to the generator inlet.

Dimension H1 is specified to prevent condensate backflow from the generator into the steam piping. Dimension H2 is specified to prevent excessive backpressure to the drain cooler. During factory testing, the unit is calibrated to provide design condensate flow to a condensate system operating at 15 PSIG (1 bar) (as measured immediately downstream of the drain solenoid valve). To duplicate this condition in the field, an adjustable condensate system backpressure valve (by others) must be installed such that:

$$\Delta P_{H2} + \Delta P_{PIPING} + \Delta P_{VALVE} = 15.0 \text{ PSIG (1bar)}$$

Where:

ΔP_{H2} = Pressure drop due to height, H2

ΔP_{PIPING} = Pressure drop due to condensate piping elbows, bends, etc.

ΔP_{VALVE} = Pressure drop due to condensate system backpressure valve

The drain cooler installed on the unit effectively eliminates the need for an additional condensate cooler or a steam trap. The drain solenoid valve is a factory installed device to insure zero steam flow through the unit during shutdown.

All steam piping should be adequately supported and braced independent of the chiller. The support system must account for the expansion and contraction of steam piping, avoiding the imposition of strain on the chiller components.

Control Valve Sizing – Control valves are sized for each job based upon the available steam pressure, required steam pressure at the unit, and the full load required steam flow. During start-up, YORK technicians establish the minimum load steam valve position (20%) and set the limit switch accordingly, eliminating possible problems at start-up.

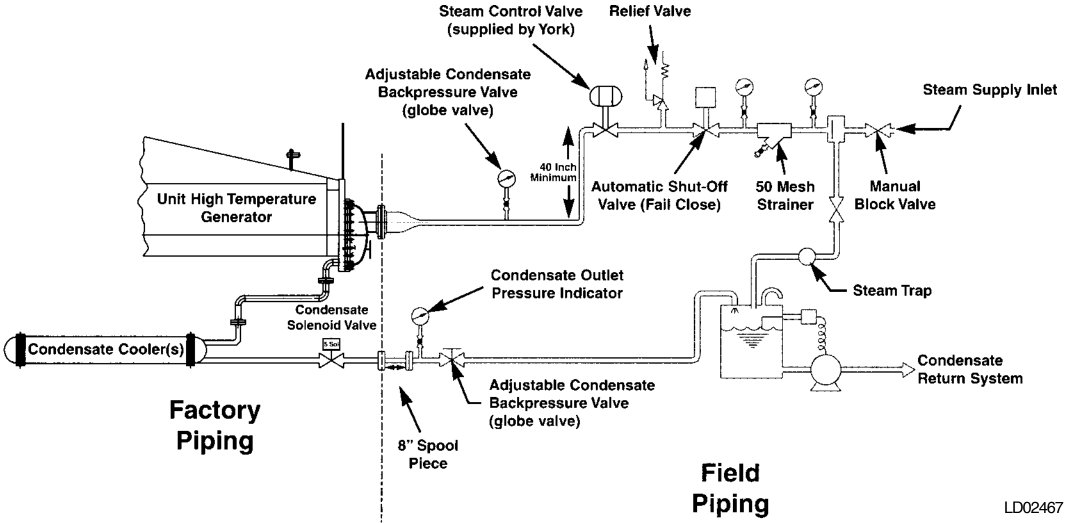

FIG. 17 – TYPICAL PIPING LAYOUT

PURGE SYSTEMS

An absorption chiller has a purge system that allows the chiller to remove non-condensible gases from the chiller. Non-condensibles inside the chiller reduce the cooling capacity of the chiller and, in the presence of the lithium bromide salt solution, enhance corrosion. Therefore it is imperative that an absorption chiller be purged regularly.

DEFINITIONS

Non-condensibles – A non-condensible is defined as a gaseous substance that cannot be liquefied or condensed at the pressure and temperature surrounding it.

Non-condensibles appear in two forms in absorption units:

1. Internally generated non-condensibles are formed as a by-product of corrosion.
2. Air that may be drawn into a unit via leaks.

Non-condensibles that collect in the absorber section of the unit blanket the heat transfer tubes and reduce the absorber's ability to capture the refrigerant vapor. Inside the absorber, the evaporator pressure rises with a corresponding rise in the boiling temperature of the refrigerant spraying on the tubes. The absorber pressure decreases because the solution concentration rises.

This loss of absorber performance can be compared to a plugged suction strainer in a conventional compression-type system. The evaporator pressure rises but the actual compressor suction pressure becomes lower in the compression system.

Eventually, if the non-condensible gases are not purged from the high side of the unit, they end up in the condenser where they blanket the condenser tubes. These non-condensibles raise the pressure inside the condenser, reducing its capacity. Full load capacity will be prevented by high condensing pressure.

Non-Condensible Quantities – An absorption unit's general health can be determined by both the quantity and quality of the non-condensibles it produces. A properly maintained absorption unit will produce very few non-condensibles – the fewer the better. A small amount of internally generated gases will always be present and should be considered normal. Air leaks, no matter how small, will almost always cause noticeable increases in the amount of non-condensibles a unit produces.

Since it is important to correct any air leaks as soon as possible, it is essential to develop a disciplined method of purging a unit so that any abnormalities can be discovered quickly. On SmartPurge™ equipped units, the purge tank is automatically evacuated only when necessary and the frequency of evacuation is continuously monitored.

TYPES OF NON-CONDENSIBLES

The most common type of non-condensible is air. Air enters the unit via leaks or accidentally during maintenance procedures. Air is made up of two main gases: nitrogen and oxygen. Other gases may collect inside the unit due to chemical reactions, some of which are caused by the oxygen in the air.

Nitrogen – Nitrogen, a colorless, odorless gas considered inert from a corrosion stand point, comes from air entering the unit. Air is comprised of approximately eighty per cent nitrogen. Although nitrogen causes no chemical reactions or corrosion within the unit, it does cause capacity problems if it collects in the absorber or the condenser.

Where the nitrogen accumulates will depend on the actual location of the leak. A leak in the absorber or evaporator side of the main shell will generally result in the nitrogen collecting in the absorber. High side leaks, i.e. leaks occurring in the high temperature generator, low temperature generator or the condenser will normally result in the nitrogen collecting in the condenser.

Oxygen – As air enters a unit via a leak or an accidental event, oxygen in the air reacts with the steel and copper materials inside the unit. If too much oxygen leaks into the unit, the reaction may reduce the lifetime of the chiller. This oxidizing reaction produces small amounts of hydrogen gas inside the chiller shell.

Hydrogen Gas – Hydrogen is created as a by-product of corrosion. How much hydrogen is being generated and when it is being generated are important clues to the source of many service problems. The rules are slightly different depending on the type of corrosion inhibitor present in the solution.

Lithium nitrate inhibited units seldom produce large quantities of hydrogen unless the inhibitor quantity is low or non-existent. Small air leaks will increase the rate of inhibitor depletion but as long as the lithium nitrate inhibitor levels are within the allowable range, nitrogen and/or an oxide of nitrogen (NOx) will be present in the purge gas.

Lithium molybdate inhibited units will produce hydrogen when an active air leak is present even if the inhibitor level is within the acceptable range. Air leaks will cause increased depletion of the corrosion inhibitor which will accelerate the generation of hydrogen. After a leak is repaired, lithium molybdate inhibited units may still produce some hydrogen for approximately forty eight hours of unit operation.

No matter which type of inhibitor is present, if a unit is producing hydrogen, the hydrogen amounts will increase as the unit firing rate is increased. High temperature, high concentration areas of the unit produce the most hydrogen since the rate of corrosion will be higher in these areas.

Palladium cells can be used in an absorption chiller to remove hydrogen, reducing the number of purge cycles. However, constantly removing hydrogen from an absorption chiller has the effect of masking corrosion inside the chiller. YORK absorption chillers allow the hydrogen gas to collect in the purge tank where it is stored until the next purge cycle.

Ammonia (NH_3) – Ammonia is a colorless, pungent smelling gas which is extremely soluble in water. It is lighter than air. Most of the ammonia in a unit will remained dissolved in the refrigerant water or solution, however depending on the amount of ammonia in a unit, some will find its way into the purge system.

NOx – Several different oxides of Nitrogen may be produced by a absorption unit. Although not nearly as common as hydrogen, one or more of these compounds may be produced by a unit where lithium nitrate inhibitor is being consumed at a higher than normal rate. The most common cause of high inhibitor consumption is an air leak.

Nitrous Oxide (N_2O) – Colorless, somewhat sweet smelling although the odor of the alcohol in the purge gas may overpower any odor it may have. N_2O is 1.5 times as heavy as air. Supports combustion much like oxygen. Commonly referred to as laughing gas.

Nitric Oxide (NO) – Colorless, slightly heavier than air, does not support combustion, and usually is oxidized to NO_2 when exposed to air.

Nitrogen Dioxide (NO_2) – Reddish brown gas, 1.6 times heavier than air, very soluble in water, unpleasant odor, and is considered noxious to breathe.

Nitrogen Tetroxide (N_2O_4) – Pale yellow gas formed as two molecules of NO2 combine to form one molecule of N_2O_4.

OPERATOR-LESS PURGING WITH YORK'S SMARTPURGE™

The purge tank must be evacuated by the unit purge pump. This can be done either manually or automatically depending if the unit is equipped with SmartPurge™ or not.

SmartPurge™ is YORK's solution to operator-less purging. It is a series of pressure transducers, solenoid valves and connections that automatically run a purge cycle without the need for an operator's supervision. The micropanel constantly monitors the purge tank pressure and evacuates the purge tank when the tank pressure reaches 60 mm Hg absolute. The automatic purge system stops evacuating the purge tank when its pressure is reduced to 30 mm Hg.

FREE COOLING

In regions where low temperature tower water is available for “free cooling”, there are operating and design considerations necessary for proper and reliable absorption chiller performance.

YORK YPC absorption chillers require a minimum entering tower water temperature (ECWT) of 68°F during normal operation. If cooling tower water temperatures drop below 68°F, the absorption chiller will eventually perform a safety shutdown on a low refrigerant temperature (LRT). During a LRT shutdown, the dilution cycle is bypassed in order to prevent possible tube-side freeze-up.

When the ECWT is changing from a “free cooling” situation to cooling operation, the absorption chiller cannot start with ECWT below 59°F. YORK YPC absorption chiller can use ECWT as low as 59°F at start-up, but must reach 68°F within 30 minutes of operation. A tower bypass piping system as shown in Fig. 18, is recommended to raise the ECWT temperature above 68°F (minimum temperature required for operation).

When the ECWT is decreasing, tending towards a “free cooling” condition, the absorption chiller must shutdown when the ECWT cannot be maintained above 68°F using a tower water bypass. The unit shutdown must include the complete dilution cycle, initiated by the chiller micropanel, before the chilled water and tower water can be diverted away from the absorption chiller. If the chiller is not permitted to run a full dilution cycle, solution crystallization may occur.

CAUTION: The dilution cycle requires both chilled and tower water flow through the chiller during the entire dilution cycle. An interrupted dilution cycle may result in chiller crystallization or possible tube freeze-up in the chilled water circuit.

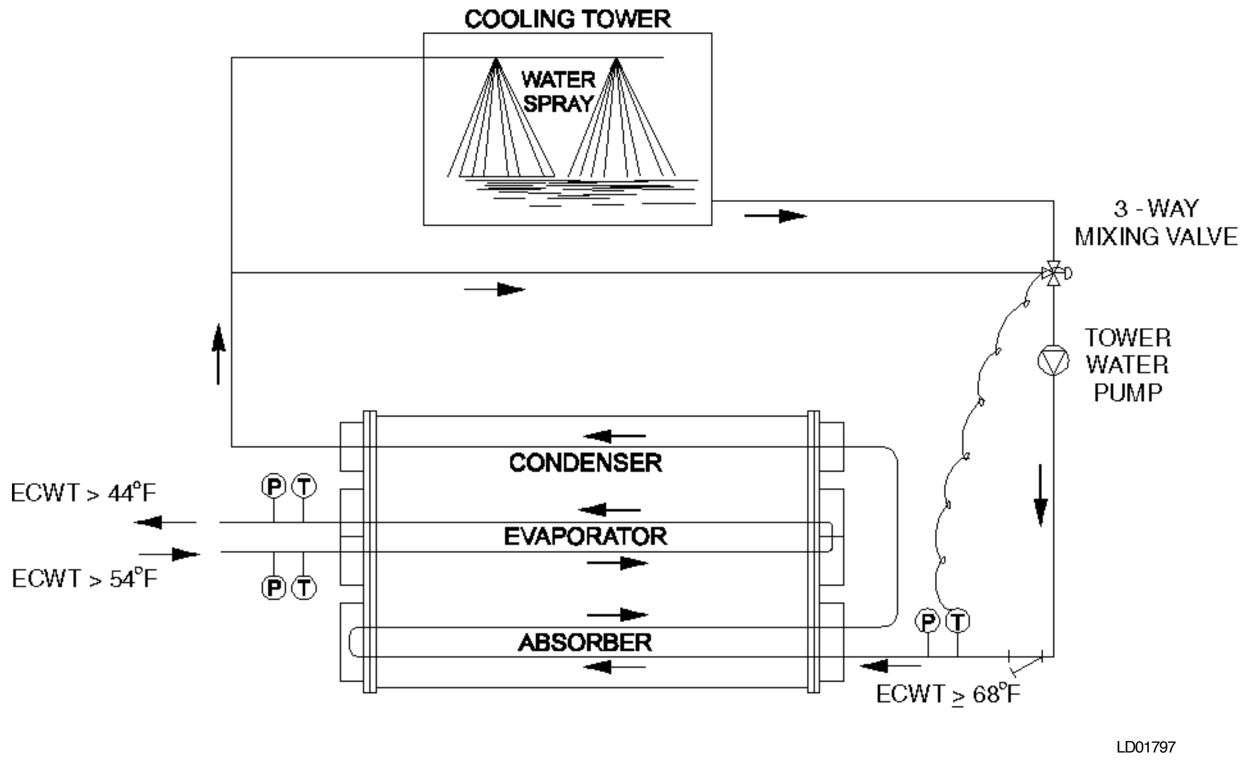

FIG. 18 – TOWER BYPASS PIPING SCHEMATIC

CHILLER STARTUP AND SHUTDOWN

This section addresses the start-up and shutdown of the YORK two-stage absorption chiller. Start-up and shutdown problems may be prevented provided the system is designed to allow the absorption chiller to have water flow during its critical times: start-up and shutdown.

It is universally accepted that absorption chillers are slower to react to changing conditions than electric chillers. With this in mind, it is easy to understand that certain issues must be considered when designing a multiple chiller system that includes absorption cooling.

Two critical times for the chiller operation are during start-up and shutdown. If certain criteria (i.e. chilled and tower water flow or the minimum ECWT) are not met during start-up, the chiller will not start. Likewise, during shutdown, if the dilution cycle is interrupted, crystallization may occur. Although these problems may sound drastic, they are remedied by properly following the installation instructions.

The literature uses the term "pump contact" referring to contacts that are used to control flow of water to the chiller. For installations with building automation systems, these contacts may be used as a means of water flow control to the chiller, whether they control a pump or a valve. Some contacts are hot, meaning they carry a current. The power to these contacts may be removed in some instances to provide a dry contact to use a building automation system signal. Consult the installation, field modifications and wiring diagrams for details.

CONTROL OF BUILDING WATER FLOW DURING CHILLER START-UP

During chiller start-up, the YORK micro panel performs a "systems" check to insure that it is ready to start. Before the chiller starts, the first checks it makes are for system setpoint configurations: local/remote start, chilled water setpoint, and steam demand limiting (steam-fired only). The next step in the start-up procedure is to close the chilled and tower water pump contacts, followed by a display message "COOLING START SEQUENCE INITIATED," allowing flow to the chiller. Fig. 19 shows a simple control flow schematic that should be followed to insure that the chiller operates properly within a building cooling system.

NOTE: Note that the chiller allows for a 30-second delay between the closing of the pump contacts and the return signal from the chilled water flow switch showing flow. If the chiller does not receive a chilled water flow signal from the flow switch, it will shut down on a safety shutdown in order to prevent freezing in the evaporator.

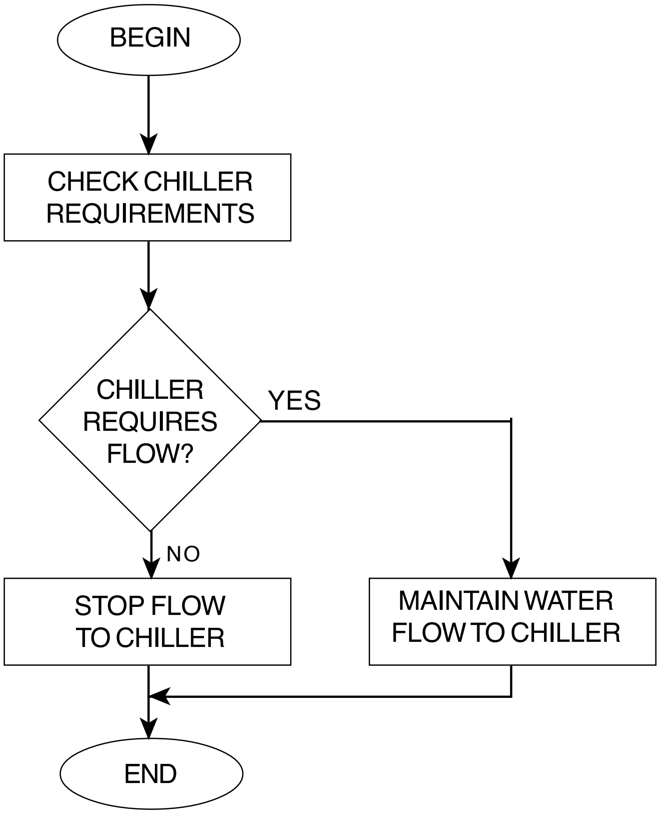

FIG. 19 – PUMP CONTROL FLOW CHART

CONTROL OF BUILDING WATER FLOW DURING CHILLER SHUTDOWN

Absorption chillers rely on a dilution cycle to reduce the concentration of the solution in the chiller and to dissipate the heat from the solution. The combination of the heat and the concentration may cause crystallization in the chiller when the machine cools to ambient room temperatures.

The chiller's means of protecting itself from crystallization during a chiller shutdown is the dilution cycle. The absorption chiller's dilution cycle requires tower and chilled water during the entire process. If the chiller loses flow at any point during the dilution cycle, it will exit the dilution cycle and perform a safety shutdown, requiring a manual restart.

To summarize, it is necessary to emphasize the importance of giving the chiller control of water flow to the chiller when using a building automation system. If a building automation system is not used, it is necessary to control the building pumps via the control contacts in the chiller micro panel. The shutdown sequence of the chiller requires the same control flow as shown in Fig. 19 for the start-up of the chiller.

POWER FAILURE APPLICATIONS

With absorption cooling equipment, the primary concern during a power failure is eliminating the risks of crystallization. This can be accomplished by either reducing the solution concentration with the dilution cycle or maintaining a solution temperature above the crystallization point for the solution concentration.

Power failures that occur during chiller operation limit the options available to keep the chiller from crystallizing. Without main building power supply, the options to protect the chiller are limited to insulating the chiller from the ambient room air, providing an emergency power supply sufficient to either maintain operation of the chiller or, at a minimum, to perform a "low power" dilution cycle. In order to fully understand why the following methods of protecting the chiller should be used during a power failure the designer must understand the relationship between lithium bromide and water, and the function of the dilution cycle.

LITHIUM BROMIDE AND WATER DURING POWER FAILURE

The PTX chart displays the relationship of lithium bromide and water inside the absorption chiller. The concentration of the solution in the chiller depends upon the solution temperature and refrigerant temperature (or pressure).

During a power failure, the solution cannot be diluted unless adequate emergency power is supplied to the chiller and pumps. The provisions taken during design of the chiller system will determine the risk of crystallization during a power failure. For example, the solution reference point A in Fig. 20 reflects the operating conditions of the solution in one location of the absorption cycle prior to power failure. During a power loss, the solution reference point A will move in one of two directions depending on the power failure application:

1. Line A-B shows a line of constant concentration which shows that the solution temperature is decreasing; however, the concentration remains the same. This demonstrates what would happen if the chiller was shut down without running the dilution cycle. As the solution cools, it approaches the solution crystallization zone. From Point A to Point B this is not a great concern; however, if Point A was closer to the crystallization zone, as it may during full-load operation of the chiller, the risk of crystallization is greater. Therefore, the best defense against crystallization resulting from the solution cooling is to insulate the chiller to retain the heat in the solution.

2. Line A-C shows a line of changing solution concentration and temperature. This line demonstrates what happens when the chiller is shut off and runs the dilution cycle.

The general trend of conditions 1 and 2 above hold for the solution throughout the entire chiller. The whole cycle diagram as shown in the Fig. 1 will shift in the direction of the two line A-B or A-C depending on power failure designs.

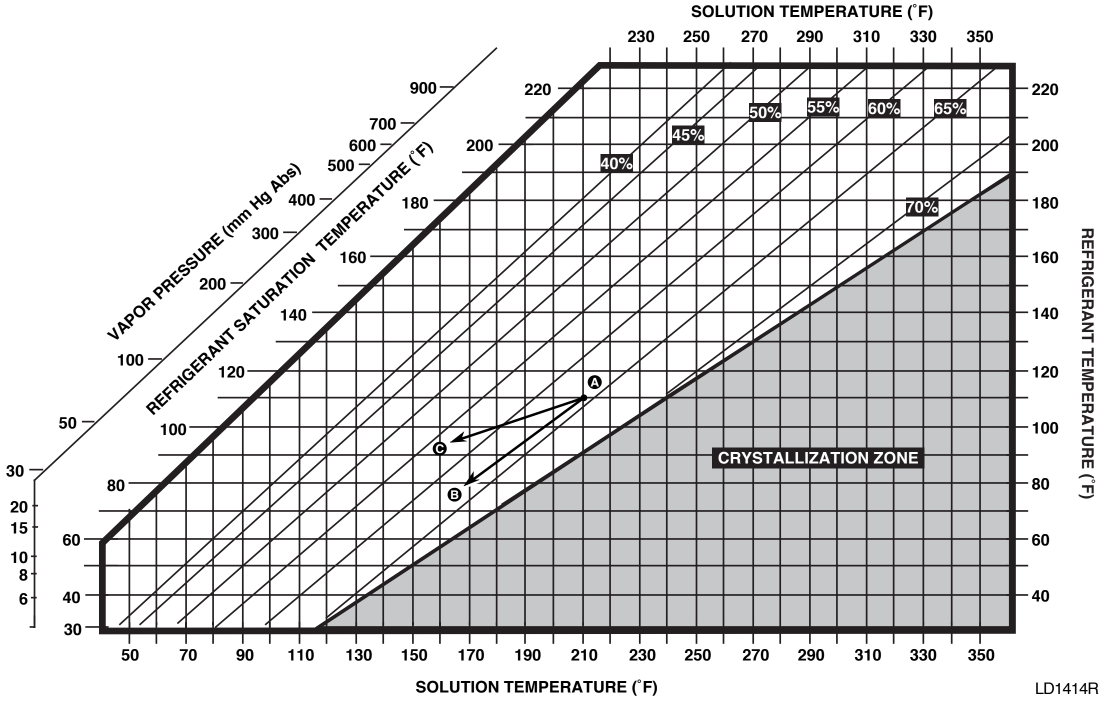

FIG. 20 – EFFECTS OF POWER FAILURE ON SOLUTION PROPERTIES

CRYSTALLIZATION PROTECTION DURING POWER FAILURES

In the event of a power failure, a chiller does not possess the means to protect itself against crystallization by running a dilution cycle. York has devised a three-prong attack at eliminating the risks of crystallization as a result of power failures. This system is implemented in three levels of protection. Listed below are the three solutions, ranked by their installation costs: Insuation only, backup power for a limited dilution cycle and full-emergency power to the chiller. Any three may be used, or all three in conjunction, however each method has its own benefits.

SOLUTION #1: INSULATION

Lowest initial cost, lowest level of protection

The primary reasons an absorption chiller is insulated are to prevent heat gain in the evaporator, eliminate condensation on the evaporator surface in humid climates, reduce heat loss from the generator to the equipment room and to protect personnel working on or around the absorption equipment from the high temperature surfaces on the generator. Insulation serves an additional purpose for absorption chillers in the event of a power failure. During a power failure, insulation can protect the chiller against crystallization for up to eight (8) hours provided the insulation is applied to the appropriate areas and the ambient room temperature is 70°F (21.1°C) or above.

In the event of a power failure, special attention should be paid toward the hot sections of the chiller that contain lithium bromide solution. These are the critical areas of the chiller that may be subject to crystallization. During a power failure, the solution remains stagnant and is allowed to cool to room temperature. Depending on the concentration and the temperature, this solution may crystallize if it is allowed to cool. Some areas of the chiller, such as small diameter solution piping will cool quicker than a larger thermal mass such as the generator. By insulating these critical areas, the heat in the solution may be retained long enough for the customer to restore power, allowing the chiller to perform a proper dilution cycle, or restart the cooling process.

Insulation diagrams are located in Appendix A. As shown in the example diagram in Figure 21, these diagrams show three different areas of the chiller that require insulation: cold, hot and intermediate sections in three different shades of gray.

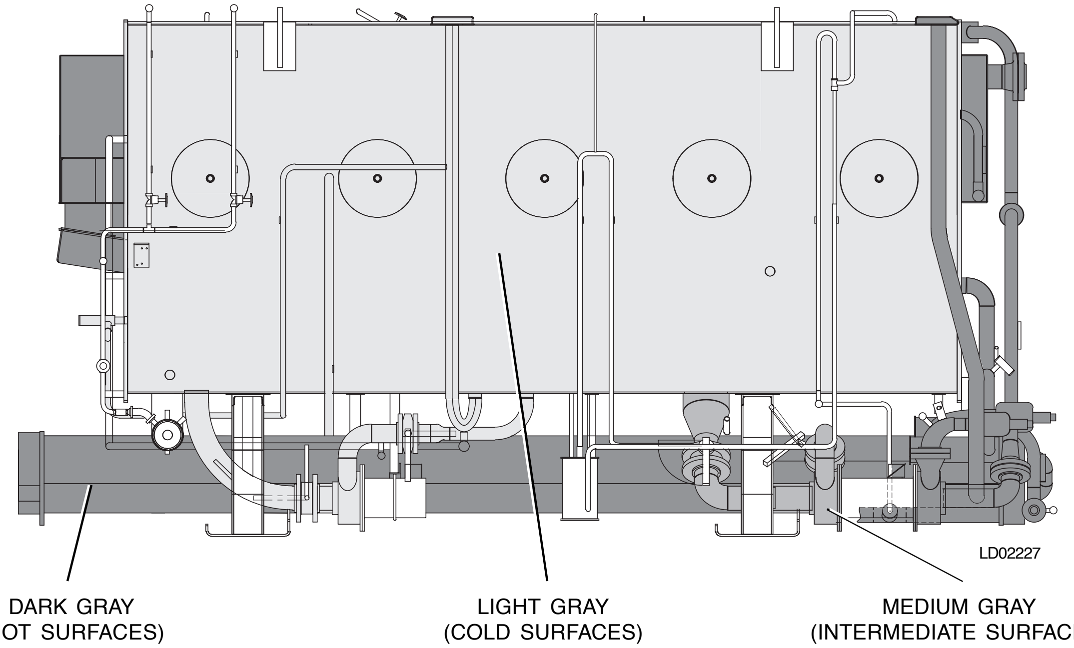

FIG. 21 – EXAMPLE INSULATION REQUIREMENTS FOR CRYSTALLIZATION PROTECTION

The cold surfaces, shown in the lightest gray, are insulated primarily to prevent condensation from forming on the cold surfaces in humid climates. A slight increase in system performance may be observed, however it may be negligible depending on the accuracy of the measuring equipment. Three-quarter-inch (3/4" or 1.91 cm) thick closed-cell foam is recommended to properly insulate cold surfaces of the chiller.

Hot surfaces, shown in dark gray, are areas that will have high temperature lithium bromide solution such as the generator, heat exchangers or hot water heater. These areas are insulated to 1.) protect personnel working on or around the equipment or 2.) minimize heat loss to the equipment room. Two-inch (2" or 5.08 cm) thick three pound density fiberglass insulation is recommended for the proper insulation of hot surfaces.

Intermediate temperature surfaces of the chiller contain lithium bromide solution at warm temperatures, however the concentration may be at a level where it is subject to crystallization at room temperature. In order to minimize the exposure to crystallization, these surfaces should be insulated. One-inch (1" or 2.54 cm) thick two pound density fiberglass insulation is sufficient for this type of protection.

All three sections of the chiller must be insulated in this manner to protect the chiller against crystallization during a power failure. Eight hours at 70°F is the limitation of this design, therefore additional protective measures must be taken if it is expected that a power failure may occur that would last longer than eight hours. For actual installation instructions, refer to the product service literature Installation Instructions.

SOLUTION #2: EMERGENCY POWER FOR LIMITED DILUTION CYCLE

Second lowest initial cost, middle level of protection

The next level of protection is York's exclusive limited power dilution cycle. The limited power dilution cycle is a modified version of the full dilution cycle that uses only the solution pumps to circulate the solution allowing the heat to dissipate more evenly throughout the machine and to reduce the solution concentration to lower levels, less subject to crystallization.

The Millennium™ control panel will recognize a signal from an automatic transfer switch when utility power is lost and switch over to the limited power dilution cycle. The display will inform the operator that the chiller is using emergency power by running the limited dilution cycle. This cycle will continue until the solution concentration has reached a crystallization-safe level.

The limited power dilution cycle requires power for the chiller power and control panels and the solution pumps. Emergency power requirements are listed in Tables 4a and 4b.

TABLE 4A – STEAM-FIRED YPC CHILLERS

CHILLER MODEL	VOLTAGE	EMERGENCY GENERATOR (MIN KW)
14SC	200V/3Ph/60Hz	20
	230V/3Ph/60Hz	25
	380V/3Ph/50Hz	20
	460V/3Ph/60Hz	25
15SL	200V/3Ph/60Hz	25
	230V/3Ph/60Hz	30
	380V/3Ph/50Hz	25
	460V/3Ph/60Hz	30
16S	200V/3Ph/60Hz	25
	230V/3Ph/60Hz	30
	380V/3Ph/50Hz	25
	460V/3Ph/60Hz	30
16SL	200V/3Ph/60Hz	30
	230V/3Ph/60Hz	30
	380V/3Ph/50Hz	15
	460V/3Ph/60Hz	20
17S	200V/3Ph/60Hz	30
	230V/3Ph/60Hz	30
	380V/3Ph/50Hz	15
	460V/3Ph/60Hz	20
18S	200V/3Ph/60Hz	30
	230V/3Ph/60Hz	30
	380V/3Ph/50Hz	15
	460V/3Ph/60Hz	20
19S	200V/3Ph/60Hz	30
	230V/3Ph/60Hz	30
	380V/3Ph/50Hz	15
	460V/3Ph/60Hz	20
19GL	200V/3Ph/60Hz	40
	230V/3Ph/60Hz	45
	380V/3Ph/50Hz	35
	460V/3Ph/60Hz	45
20G	200V/3Ph/60Hz	50
	230V/3Ph/60Hz	60
	380V/3Ph/50Hz	45
	460V/3Ph/60Hz	60
21G	200V/3Ph/60Hz	45
	230V/3Ph/60Hz	60
	380V/3Ph/50Hz	35
	460V/3Ph/60Hz	60
22G	200V/3Ph/60Hz	45
	230V/3Ph/60Hz	60
	380V/3Ph/50Hz	35
	460V/3Ph/60Hz	60

TABLE 4B – DIRECT-FIRED YPC CHILLERS

CHILLER MODEL	VOLTAGE	EMERGENCY GENERATOR (MIN KW)
12SC	200V/3Ph/60Hz	20
	230V/3Ph/60Hz	25
	380V/3Ph/50Hz	20
	460V/3Ph/60Hz	25
13SC	200V/3Ph/60Hz	20
	230V/3Ph/60Hz	25
	380V/3Ph/50Hz	20
	460V/3Ph/60Hz	25
14SC	200V/3Ph/60Hz	20
	230V/3Ph/60Hz	25
	380V/3Ph/50Hz	20
	460V/3Ph/60Hz	25
15SL	200V/3Ph/60Hz	25
	230V/3Ph/60Hz	30
	380V/3Ph/50Hz	25
	460V/3Ph/60Hz	30
16S	200V/3Ph/60Hz	25
	230V/3Ph/60Hz	30
	380V/3Ph/50Hz	25
	460V/3Ph/60Hz	30
16SL	200V/3Ph/60Hz	35
	230V/3Ph/60Hz	40
	380V/3Ph/50Hz	30
	460V/3Ph/60Hz	40
17S	200V/3Ph/60Hz	35
	230V/3Ph/60Hz	40
	380V/3Ph/50Hz	30
	460V/3Ph/60Hz	40
18S	200V/3Ph/60Hz	35
	230V/3Ph/60Hz	40
	380V/3Ph/50Hz	30
	460V/3Ph/60Hz	40
19S	200V/3Ph/60Hz	40
	230V/3Ph/60Hz	50
	380V/3Ph/50Hz	35
	460V/3Ph/60Hz	50
19G	200V/3Ph/60Hz	30
	230V/3Ph/60Hz	35
	380V/3Ph/50Hz	25
	460V/3Ph/60Hz	35
19GL	200V/3Ph/60Hz	40
	230V/3Ph/60Hz	45
	380V/3Ph/50Hz	35
	460V/3Ph/60Hz	45
20G	200V/3Ph/60Hz	50
	230V/3Ph/60Hz	60
	380V/3Ph/50Hz	45
	460V/3Ph/60Hz	60

SOLUTION #3: FULL EMERGENCY POWER

Highest initial cost, highest level of protection

The highest level of protection against crystallization is for the site to provide dedicated emergency to the chiller, sufficient to run the chiller and building water pumps. This method may be used simply to allow the chiller to perform a full dilution cycle or to continue normal operation providing cooling to a process or building space. Operational energy requirements of the chiller at normal operation are located in the product engineering guides (Forms 155.19-EG2, 155.19-EG3, 155.17-EG3 or 155.17-EG4).

PLANT DESIGN APPLICATIONS

The following section of this manual presents a general discussion of various plant design applications. In particular, the manual will focus on cogeneration, heat recovery and district energy applications.

COGENERATION

Cogeneration is defined by ASHRAE as the sequential production of two forms of useful energy from the same fuel source. Typically, cogeneration involves the production of electricity and heat. The heat is recovered in the form of steam or hot water. Absorption systems can provide an economical use for the recovered heat energy. Although cogeneration may involve applications other than cooling, it is the main concern of this chapter to single out the cooling applications associated with cogeneration.

Applying indirect-fired (steam or hot water) absorption chillers is currently the most common approach. The main component of a cogeneration system is the prime mover, which converts fuel or thermal energy to shaft energy. The conversion devices are normally reciprocating internal-combustion engines, combustion gas-turbines, expansion-turbines, and steam boiler -turbine combinations. Heat recovered from the exhaust gas stream or the engine coolant system is used to power an absorption machine. When the heat from a cogeneration system is used to power an absorption chiller, this not only meets the cooling load but also reduces the electrical demand on the system. Several common applications include:

- cooling occupied spaces (HVAC system)
- cooling inlet air for gas-turbines (which increases gas-turbine efficiency and capacity)
- cooling industrial process

The application of absorption systems has typically been economical for many industrial facilities with both large heat rejection and cooling requirements. The heat source for the cooling load in industrial facilities vary, depending on the process. Examples of possible heat sources include

- exhaust from a drying process, such as in a paper plant
- hot water produced in a wastewater treatment facility
- gas vapor produced in an oil refinery from a distillation process
- low-pressure steam from a back pressure steam turbine
- incinerator exhaust/coolant water

There are four major components to a cogeneration system: prime mover, generator, heat recovery, and controls. The prime mover furnishes two kinds of energy: mechanical energy from the shaft, and unused heat energy that remains after the fuel or steam has acted on the shaft. Shaft loads (generators, centrifugal compressors, process equipment) require a given amount of rotating mechanical energy. Once the prime mover is selected to provide the required shaft output, it has a fixed relationship to heat availability and system efficiency, depending on the prime-mover fuel versus heat-balance curves. Cogeneration provides an opportunity to use the fuel energy that the prime mover does not convert into shaft energy. The ability to use the prime mover waste heat determines overall system efficiency and is one of the critical factors in economic feasibility.

Gas turbines are available for cogeneration systems in sizes from 50 -175,000 brake horse power (bhp) (37.3 - 130,500 kW). Gas turbines are typically 20-25% efficient. Gas-Turbine cogeneration systems have several advantages over other types of systems. The larger gas-turbines are very reliable and often require less maintenance under continuous operation. Gas-turbine systems also have high heat recovery capability, require no cooling water, have clean exhaust, and are attractive when all the heat recovered can be used all the time. Gas turbines operate at a much lower efficiency than gas engines. In order to get a comparable system efficiency, heat recovery is essential. Exhaust-gases from a turbine range from 800 - 1000°F (427 - 538°C) and allow for high quality heat to be recovered in the form of low or high pressure steam or hot water. When used as part of a total energy system, the thermal efficiency of a gas turbine system can exceed 90%. Heat exchangers are available specifically for heat recovery from gas turbine exhausts.

Steam-turbines can be used in a variety of applications. They can be the prime movers in a cogeneration system or drive centrifugal compressors and chilled water pumps for air conditioning systems. Steam-turbines typically have a fuel to electricity efficiency of 25%. Recovering heat from steam turbines for use in absorption chillers is done using a back pressure or extraction turbine. Low pressure exhaust steam is suitable for use in a single stage absorption chiller.

Reciprocating-engines are the most common type of prime mover used in cogeneration plants because of their high efficiency (Btu/bhp hr) and their relatively low coast ($/hp) compared to gas turbines. Engines are typically 30-35% efficient. Engines can operate on a number of fuel sources, the most common being diesel and natural gas. Engines are available in sizes up to 27,000 brake horsepower (bhp) (20,130 kW). Engines operate between 360 and 1800 RPM. Operating speed is based on size, characteristics of the manufacturer, the generator, and length of time required between overhauls. Heat can be recovered from the engine jacket water system (used to cool the engine block, heads, and exhaust manifold) and from the engine exhaust. Engine jacket systems provide glycol between 210-260°F (99 - 127°C) depending on engine manufacturer and model. Exhaust gas temperatures are normally between 700 - 800°F (371 - 427°C).

HEAT RECOVERY

It is recommended that indirect heat recovery be used for recovering heat for powering absorption chillers. The most widely used method of recovering heat from absorption chillers is through the use of a heat recovery steam generator. The use of this type of heat exchanger to generate steam or hot water reduces the problem associated with the direct use of the heat source stream.

The temperature and quality of heat recovered will determine the type of absorption chiller most suited for the application. Steam pressures of less than 15 PSIG (0.103 mPa) are the most common heat source for single effect absorption units. Generating this steam by using recovered heat is fairly common. Heat sources that can generate steam for single effect absorption machines includes reciprocating engine cooling jackets and exhaust, gas turbine exhaust, incinerators, steam turbine exhausts, and industrial processes. Steam pressures of 45-125 PSIG (310 - 862 mPa) are required for two stage absorption. Steam at these pressures is advantageous because the two stage absorption unit is 40% more efficient than the single stage unit. Steam at these pressures may also be required when it must be transported long distances, as in a campus situation or where it is required for industrial processes that need a higher temperature. Generating steam at these pressures from a source heat stream is most often done with high temperature exhaust gases from gas-turbines or incinerators. Heat recovery boilers are available to meet most applications.

While steam is the most popular heat transfer fluid, medium-temperature, high-pressure water (210-260°F, 99-127°C) is often used as well. Hot water systems can be operated and installed at a lower cost than steam systems because less equipment and less maintenance is required. High temperature hot water can be generated by the same technique used to generate low pressure steam. Reciprocating engines are the most common type of prime mover found in cogeneration applications. These engines (similar to those used in YG product) provide an excellent source of hot water through the jacket water system.

Heat Recovery Equipment

There are many different ways to generate steam or hot water needed for the absorption process. A variety of heat exchangers are available to recover heat to make steam and hot water. The following equipment is available from a wide range of manufacturers and can be used alone or in combination to provide the required steam pressures or water temperatures:

- Shell-and-tube heat exchanger (liquid- or steam-to hot-water)
- Exhaust gas boilers (exhaust gas-to-steam or hot water)
- Steam separators

Shell-and-tube heat exchangers are used to transfer heat from one liquid to another. An example is shown below.

To promote heat transfer, the most viscous fluid is usually passed on the outside of the tubes to increase turbulence. However, if one fluid tends to foul the tubes more easily than the other, that fluid should be passed on the outside of the tubes to ease cleaning.

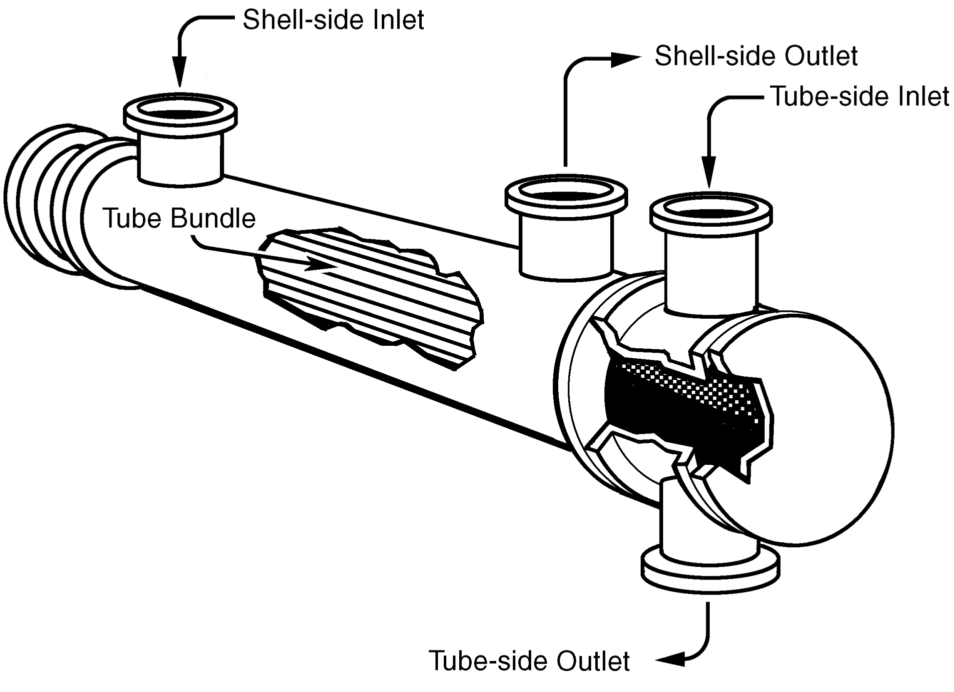

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FIG. 22 – SHELL AND TUBE HEAT EXCHANGERS

Exhaust Gas Boilers

Exhaust gases from gas-turbines or stack gases from industrial processes can be used as the heat source to generate either steam or hot water for use in an absorption machine. Exhaust gas boilers can be constructed as either water-tube or fire-tube units and may also be combined with exhaust as silencing. Two Examples are shown below.

Heat recovery from reciprocating engines can be the most complicated because there are two sources of waste heat. A typical engine converts fuels in the following percentages

- 33% Shaft HP
- 7% Radiation
- 30% Exhaust
- 30% Jacket Water

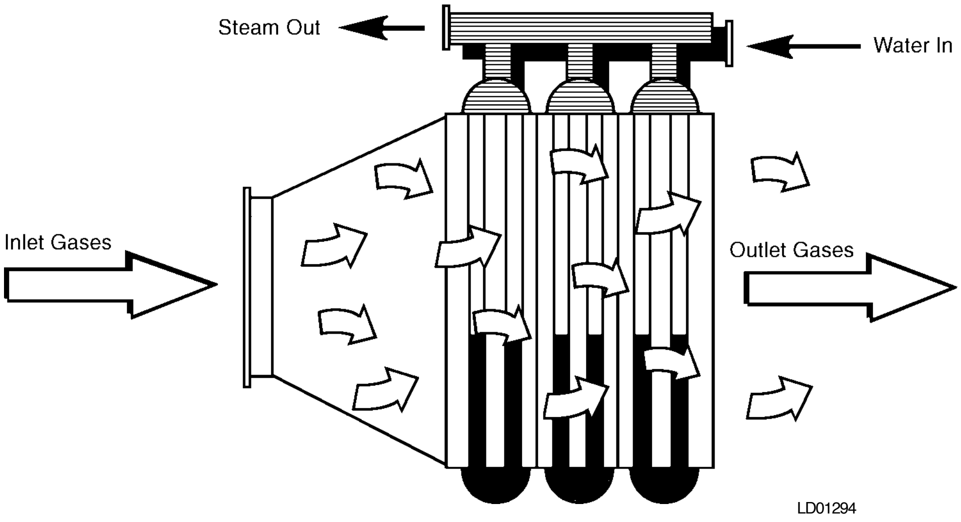

FIG. 23 – WATER TUBE BOILER USING EXHAUST GASES

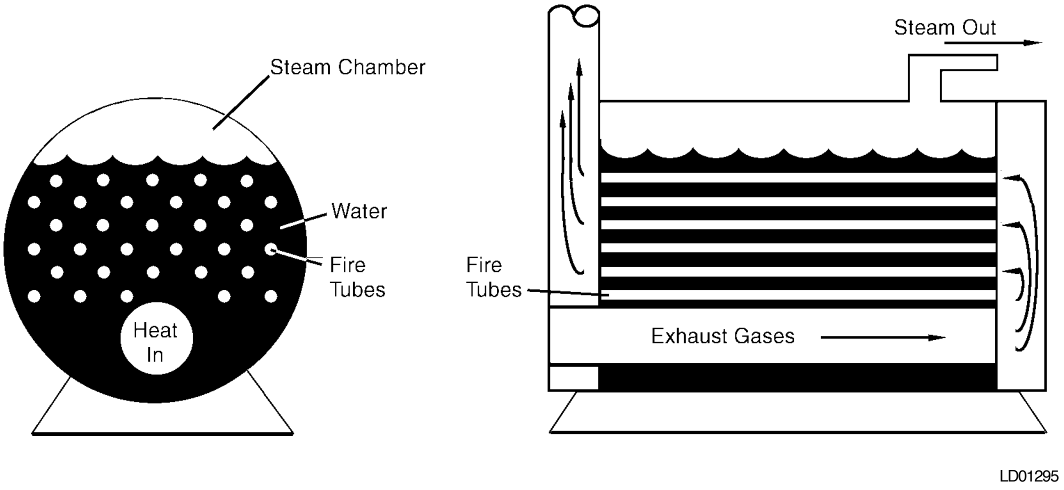

FIG. 24 – FIRE TUBE EXHAUST GAS BOILER

In order to maximize the heat recovered from an engine the exhaust and the jacket water and exhaust heat should be recovered. This could involve a shell-and-tube heat exchanger to recover jacket water heat and a heat recovery boiler to recover exhaust heat or jacket water piped in series with exhaust heat recovery equipment as illustrated below.

Exhaust heat can be recovered down to 350°F (177°C). If the temperature is allowed to get below 350°F (177°C) condensate could form in the exhaust system and cause corrosion problems. Heat recovery equipment manufacturers have developed a single heat recovery boiler to recover heat from the jacket water and exhaust gas.

An example is illustrated in Fig. 25.

Cooling Estimates

The flow rate and the temperature of a recovered heat stream determine if sufficient heat is available to produce the required cooling output. A typical single effect absorption chiller will consume 18 pounds (8.2 kg) of steam per ton hour of cooling. A typical double effect absorption chiller will consume 10 pounds (4.5 kg) of steam per ton hour of cooling.

A typical single effect machine will consume varying amounts of hot water depending on the water temperature and the delta T of the water across the absorption generator. The heat input for a single effect machine using hot water ranges from .5 to 1.5 gpm (.03 to .09 L/s) per ton of cooling. The greater the temperature difference the lower the gpm/ton required. The nominal hot water temperature for absorption chillers is 240°F (116°C). Units can operate with entering hot water as low as 190°F (88°C) with a significant reduction in capacity.

A steady flow rate of recovered heat at a constant temperature is required by an absorption machine to maintain constant cooling output. Supplemental boilers may be required to provide additional heat. Radiators or load balancing condensers can be used to dissipate excess heat. Balancing of the shaft requirement, heat recovery equipment, and cooling load is required for proper application of the system. The quantity of cooling available can be calculated if the quantity of recovered heat is known. The amount of cooling can be estimated using the following equation:

$$q_{cooling} = (COP_{Abs})\left(\frac{q_{recovered}}{C_1}\right)$$

where:

$q_{cooling}$ = cooling output (tons)

COP_{Abs} = coefficient of performance of the absorption machine

$q_{recovered}$ = heat recovered from jacket/exhaust gas (Btu/h)

C_1 = constant, 12,000 Btuh/ton

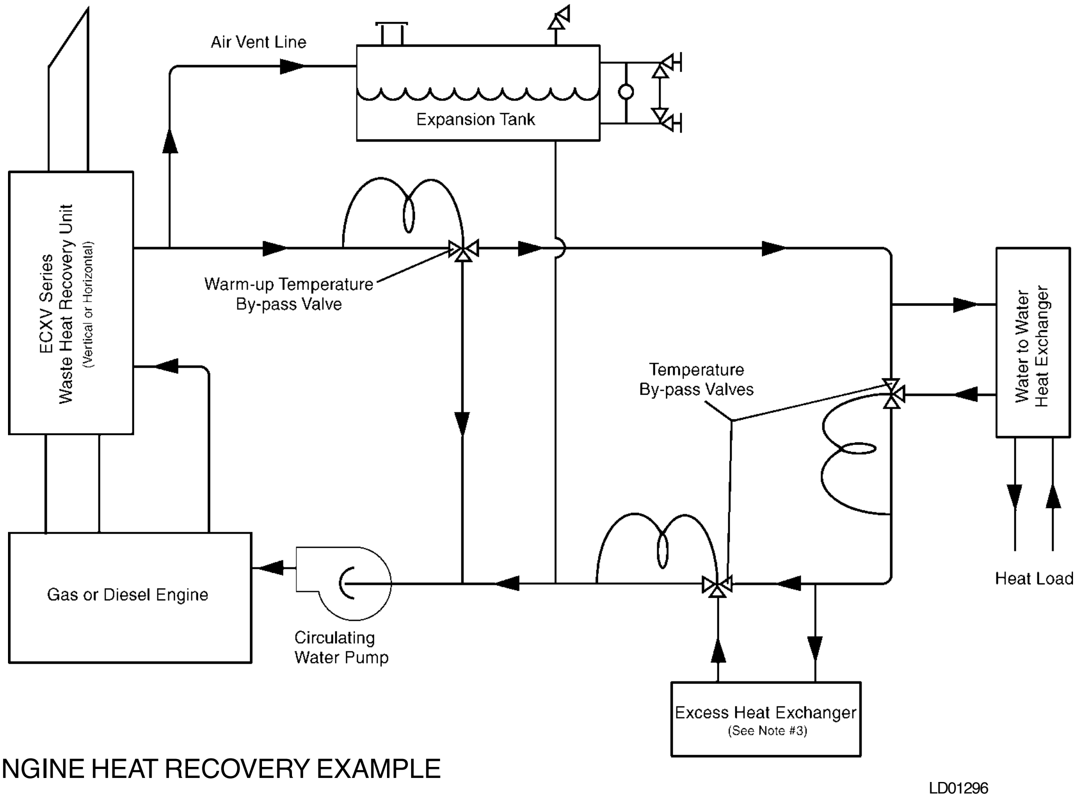

FIG. 25 – ENGINE HEAT RECOVERY EXAMPLE

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There are numerous sources of technical information available regarding cogeneration and heat recovery systems. The **ASHRAE Handbook** and **ASHRAE Application Guide for Absorption Cooling / Refrigeration Using Recovered Heat** are excellent sources of information.

DISTRICT ENERGY SYSTEMS

A district heating and cooling (DHC) system distributes thermal energy from a central source to residential, commercial, and/or industrial consumers for use in space heating, cooling, water heating, and/or process heating or cooling. The thermal energy is distributed over a network of steam or hot or chilled water lines. This allows consumers to obtain thermal energy without using valuable real estate to generate it.

There are three primary components to a DHC system: the production plant, the distribution system and the consumer's on-site systems. Although the selection of the distribution system has a great impact on the cost and operation of the DHC system, the primary concern within the content of this manual is the central plant.

DHC systems consist of three primary components: the central plant to provide the thermal or electrical energy; the distribution system to carry the energy from the central plant to the consumer's site; and the user's on-site system to distribute the energy throughout the facility. Central plants often employ the use of the exhaust heat stream from a cogeneration system to fuel an absorption chiller, however, the central source for the DHC system may be any type of boiler, incinerator, geothermal source, solar energy or thermal energy developed as a byproduct of electrical generation. The distribution systems, a large part of the initial capital investment, consists of a combination of pre-insulated and field-insulated piping to convey the energy to the consumer's site. The distribution piping is usually either buried in the ground, in a non-accessible or accessible tunnel. The last piece of the DHC system is the customer's on-site equipment that handles the incoming thermal and electric loads.

Opportunities for absorption equipment are varied and wide in many cities with existing steam-supply district heating systems. Steam supplied to a building can fuel steam-turbine, steam-fired absorption or turbo-absorption chiller systems located at individual building sites. This type of district cooling is more attractive where underground piping systems do not have adequate clearance for additional piping for central plant supplied chilled water. With steam-fired absorption equipment installed at the steam consumer's site, a district heating plant can, in effect, supply cooling as well. High pressure steam can power steam turbine-driven chillers or two-stage absorption chillers. Low pressure steam less than 15 PSIG (.103 mPa) is suitable for single-stage absorption chillers. Check for rebate opportunities to install steam-powered equipment on district steam circuits.

DHC systems are best suited towards regions where (1) the thermal load density is high and (2) the annual load factor is high. A high load density is important to cover the capital investment for the transmission and distribution system, which usually constitutes most of the capital cost for the overall system, often ranging from 50 to 75% of the initial investment. These two factors make DHC systems attractive to serving (1) industrial complexes, (2) densely populated urban areas, and (3) high-density building clusters with high thermal loads. District cooling systems are best suited for regions with high concentrations of cooling loads, often signifying tall building structures.

An additional benefit of DHC systems is that control over emissions is less expensive as well as easier to control from an environmental standpoint than distributed heating and cooling systems burning fossil fuels.

District Heating and Cooling Economic Factors

The economic factors that make DHC systems attractive include the operating personnel, insurance premiums for in-house boilers, available space within a building for HVAC equipment, maintenance costs, thermal efficiencies, availability of fuels and the major economic driving force, the initial capital investment. The initial capital investment of the DHC systems includes (1) concept planning, (2) design, (3) construction and (4) consumer interconnection.

Due to DHC systems dependence on high-density load applications, the initial design of a DHC plant must include: (1) plans for expansion and future growth, (2) cogeneration when energy and electric energy use are coincident and (3) the availability of solid waste which may provide a use of a solid waste fueled plant.

DHC plants may consist of all electric chillers, all gas- or steam-fired chillers or a combination of the two or "hybrid plant". Special consideration must be taken during the design stage and feasibility study of a hybrid plant installation. Many building codes as well as safety codes such as ASHRAE 15 have special requirements for hybrid plants such as refrigerant detectors and separation walls.

Design Considerations

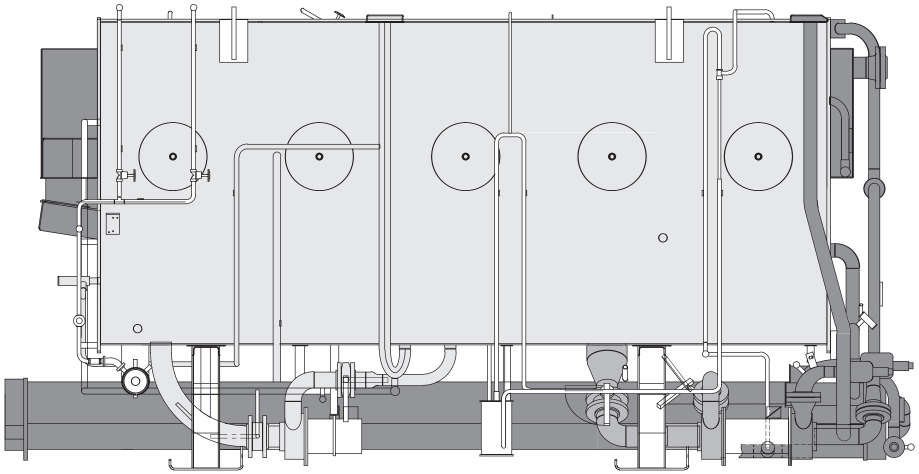
Electric chiller systems allow for variable water flow through the chilled and/or tower water circuits. Absorption chillers require constant flow through the chilled and tower water circuits. This fundamental difference between the two machines will influence layout of the chiller plant, selection and arrangement of the chillers, the distribution system and the methods of building connection to the primary system.

The trade-off between constant flow and variable flow systems must be considered as well. Constant flow systems are typically used in smaller systems where cost and simplicity of design are essential or in in-building systems with separate pumps. In-building systems with constant flow eliminate the flow balance problems between different buildings.

As mentioned above, the implications and application of a district heating and cooling systems are very far reaching. DHC systems have a high initial cost and are very labor intensive to design, however the future benefits are many. Existing district heating and cooling systems can be found throughout the world. As with all other types of chiller applications, there is a great deal of information to consider. For further information regarding district heating and cooling or "district energy" applications, the **ASHRAE Handbook** is an excellent source of information. Also, the International District Energy Association publishes proceedings from their annual meetings that have some of the latest information regarding district heating, cooling and energy supply applications.

APPENDIX A

INSULATION DIAGRAMS



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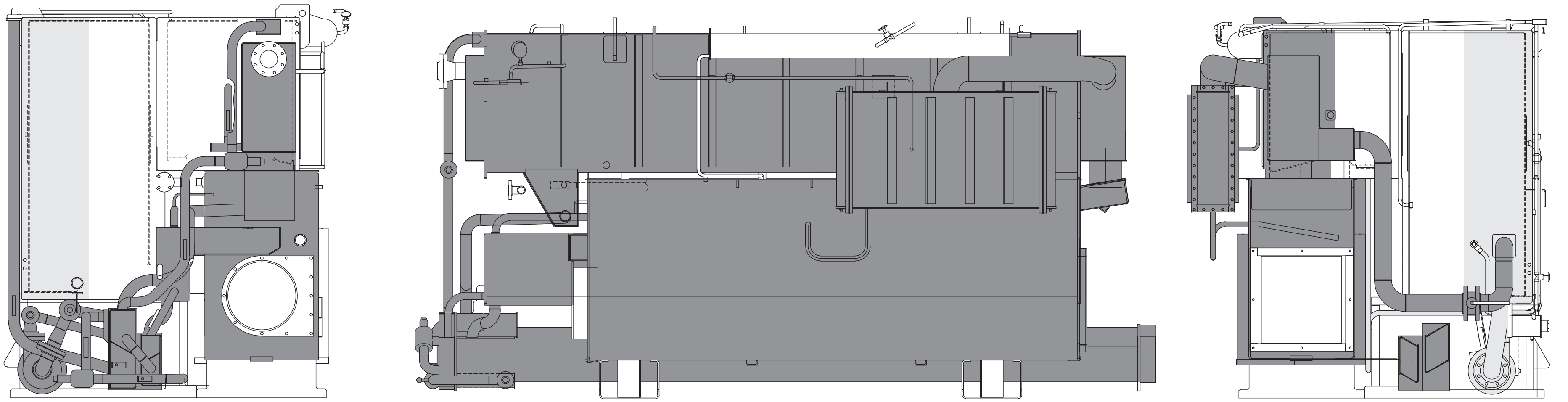

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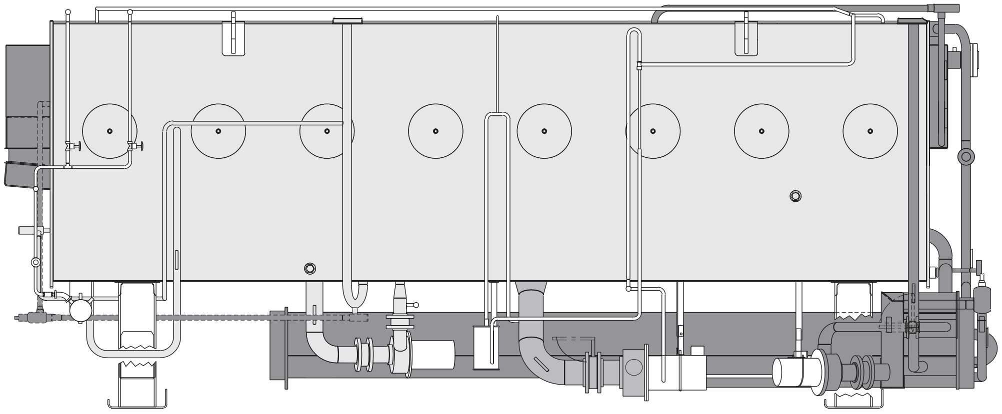
FIG. 26 – 12SC, 13SC, AND 14SC DIRECT-FIRED



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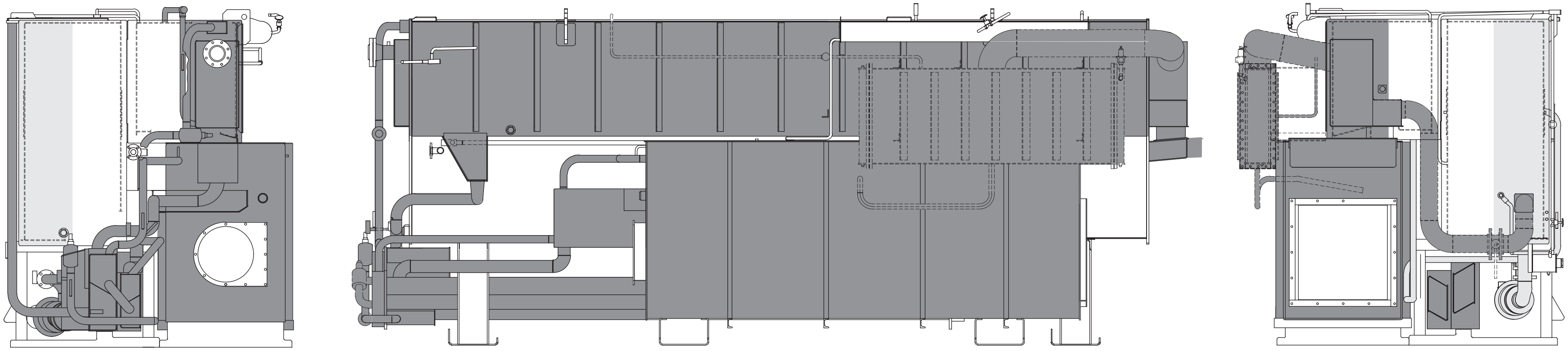

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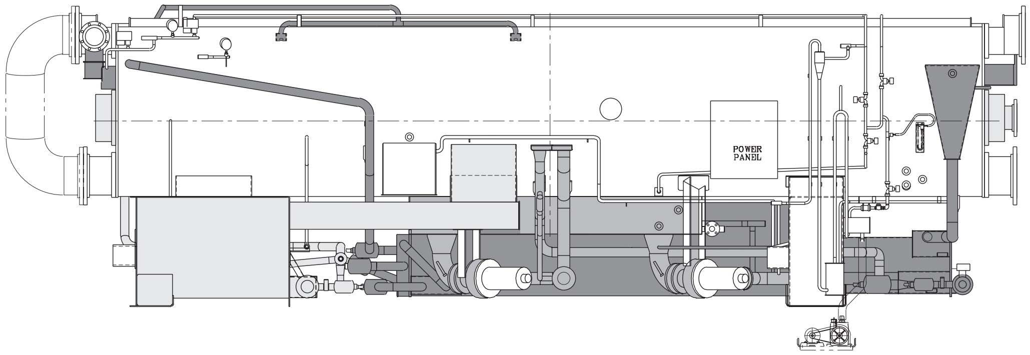
FIG. 27 – 15SL, 16S, 16SL, 17S, 18S, AND 19S DIRECT-FIRED



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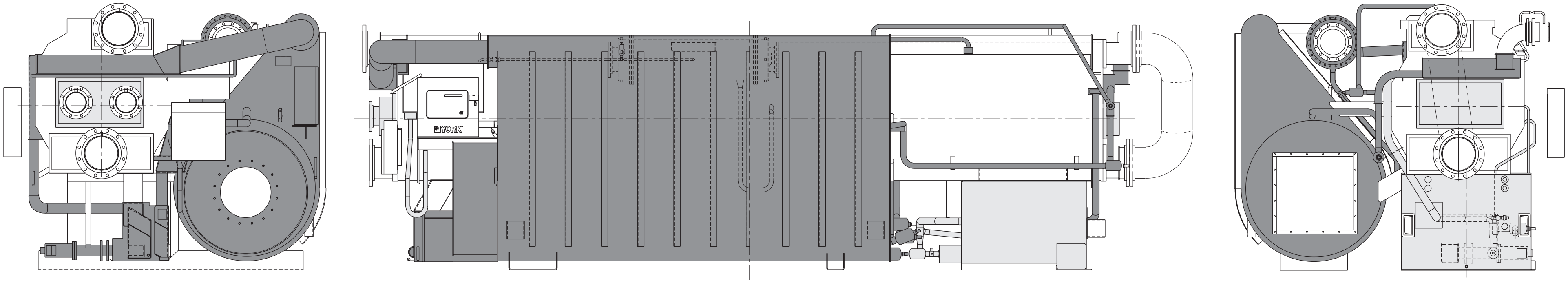

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FIG. 28 – 19GL DIRECT-FIRED

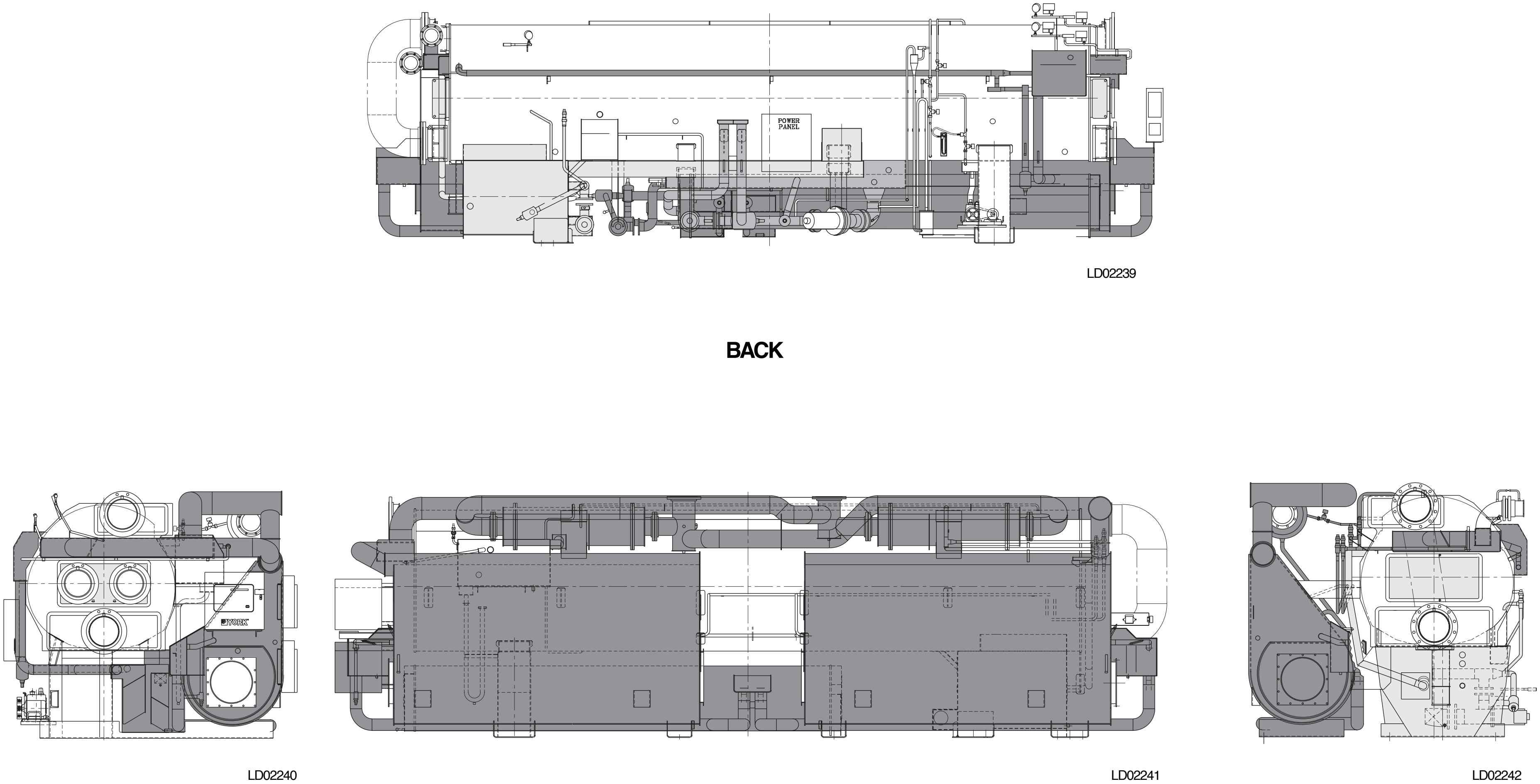

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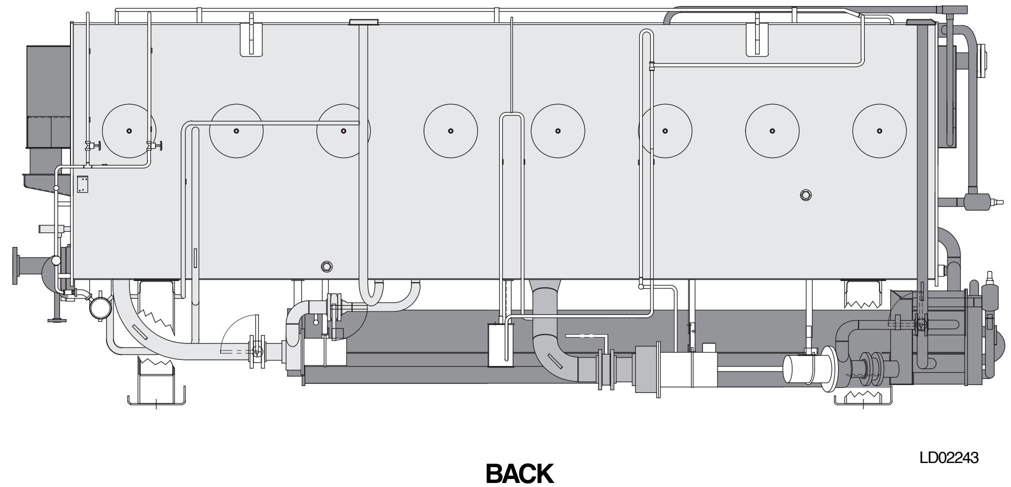
FIG. 29 – 20GL DIRECT-FIRED



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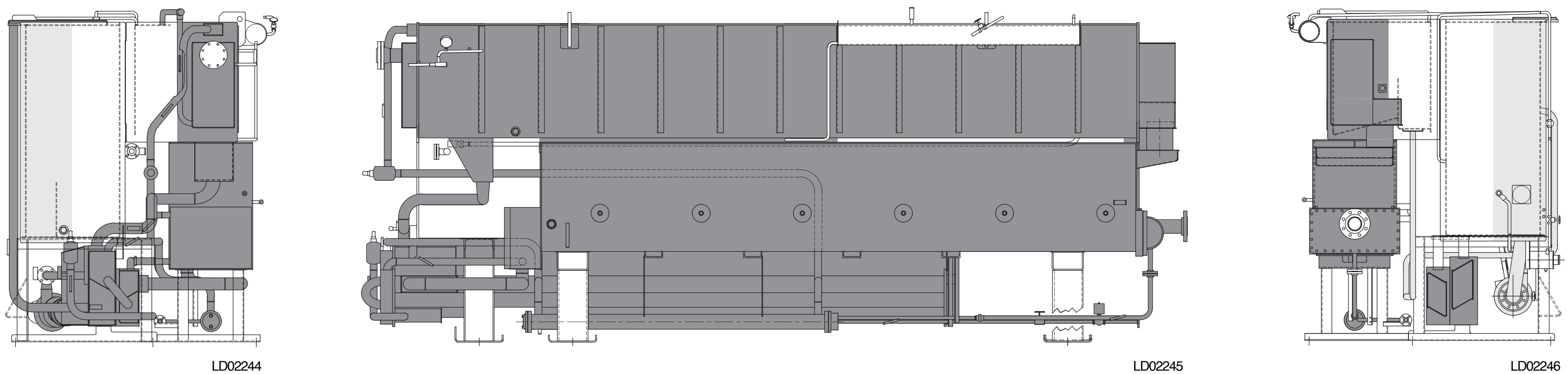
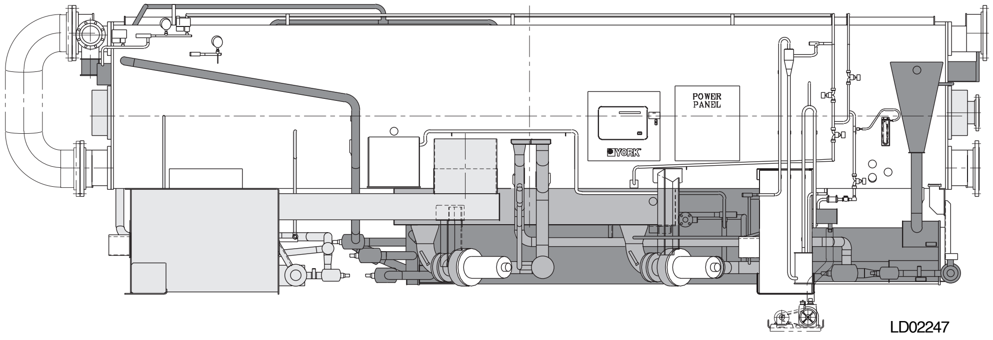


LEFT END FRONT RIGHT END

FIG. 30 – 16SL, 17S, 18S, AND 19S STEAM-FIRED



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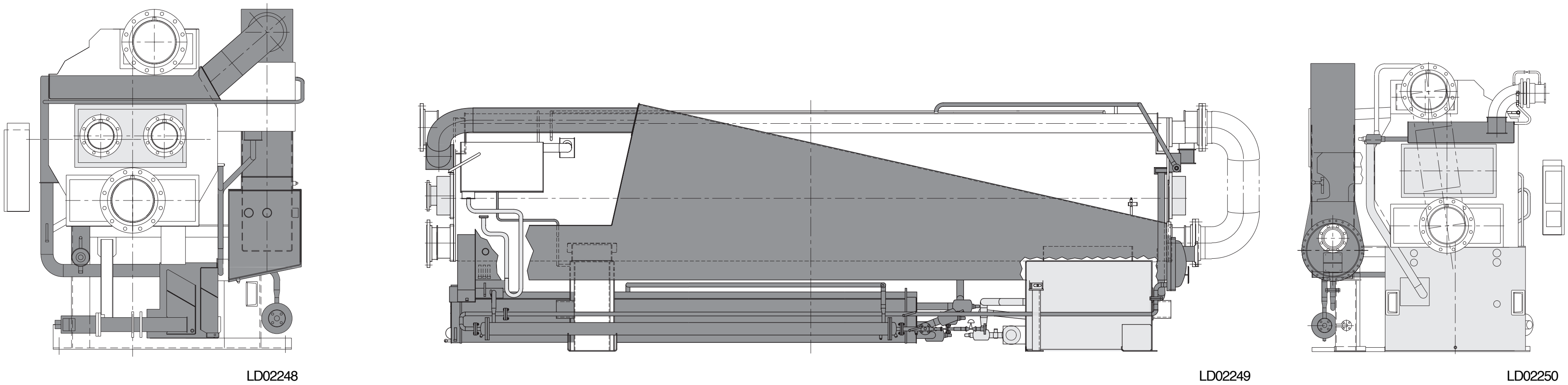
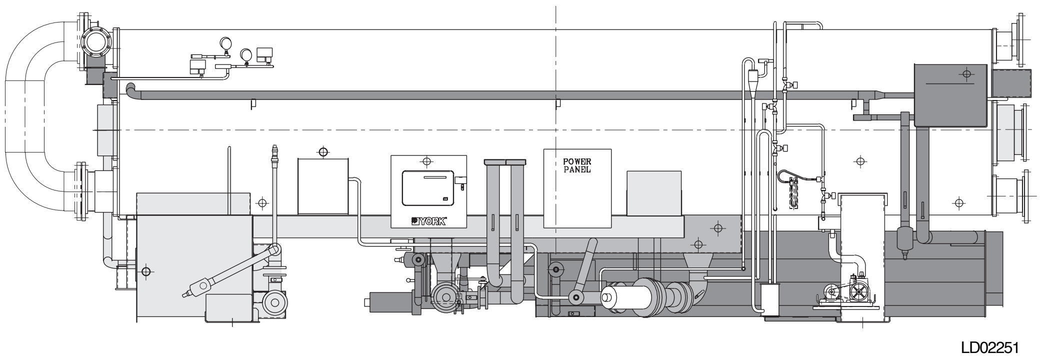


LEFT END **FRONT** **RIGHT END**

FIG. 31 – 19GL STEAM-FIRED



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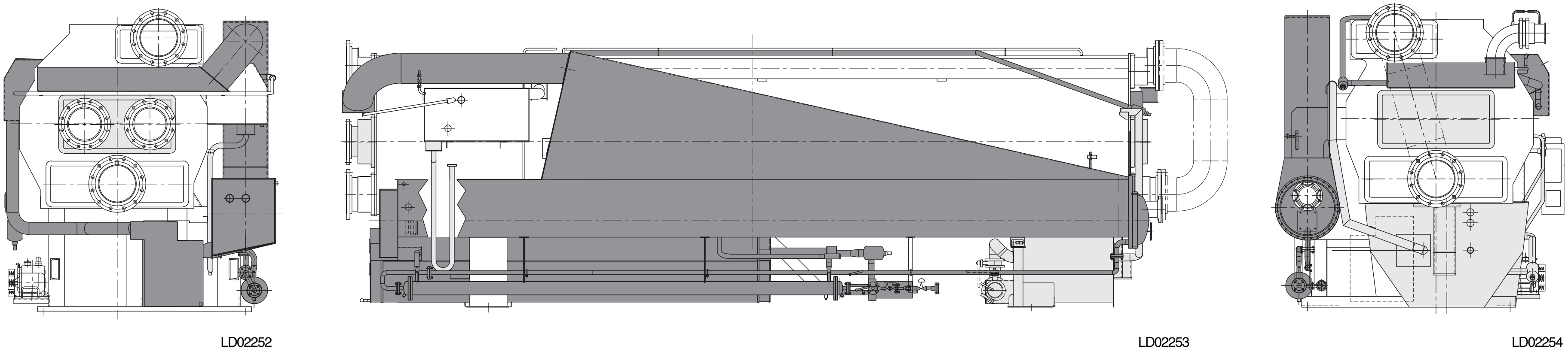
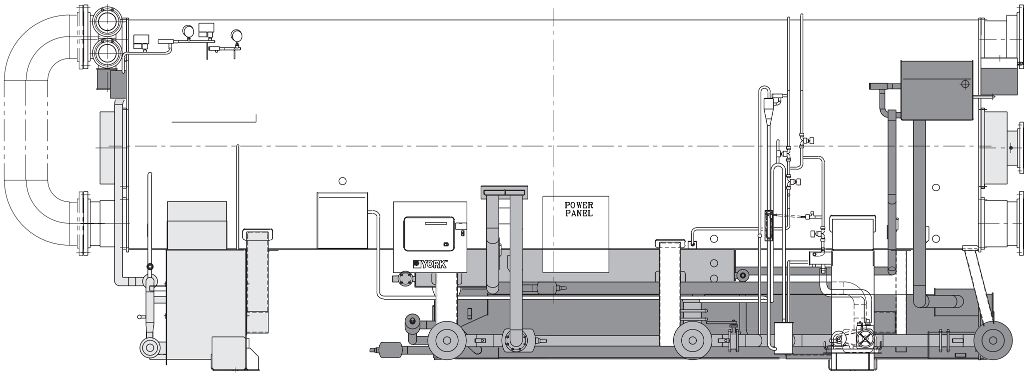


LEFT END

FRONT

RIGHT END

FIG. 32 – 20G STEAM-FIRED



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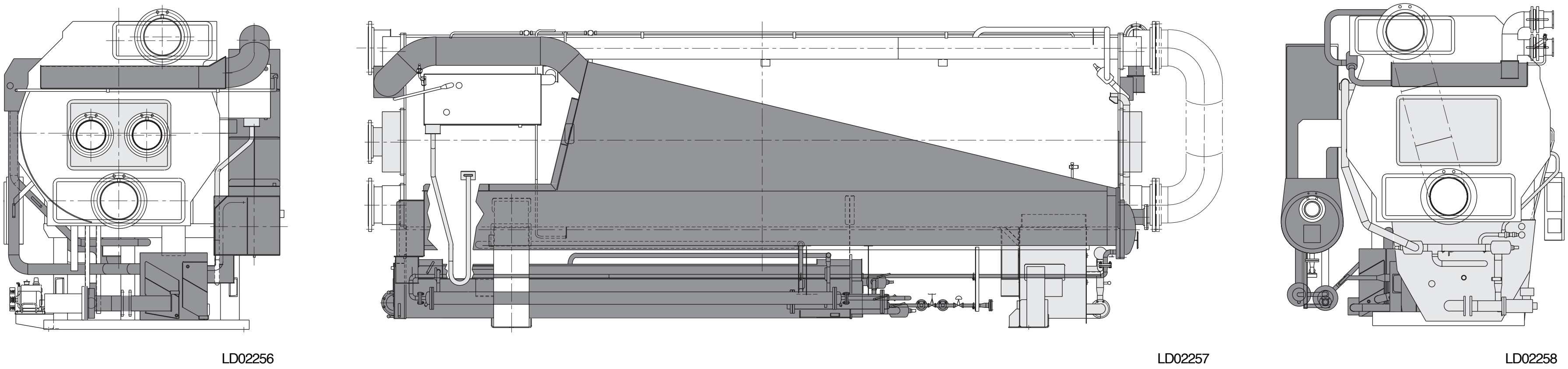
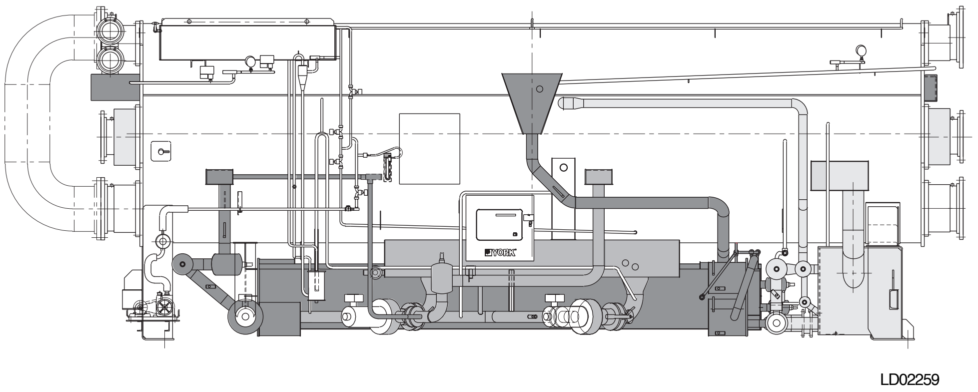


LEFT END

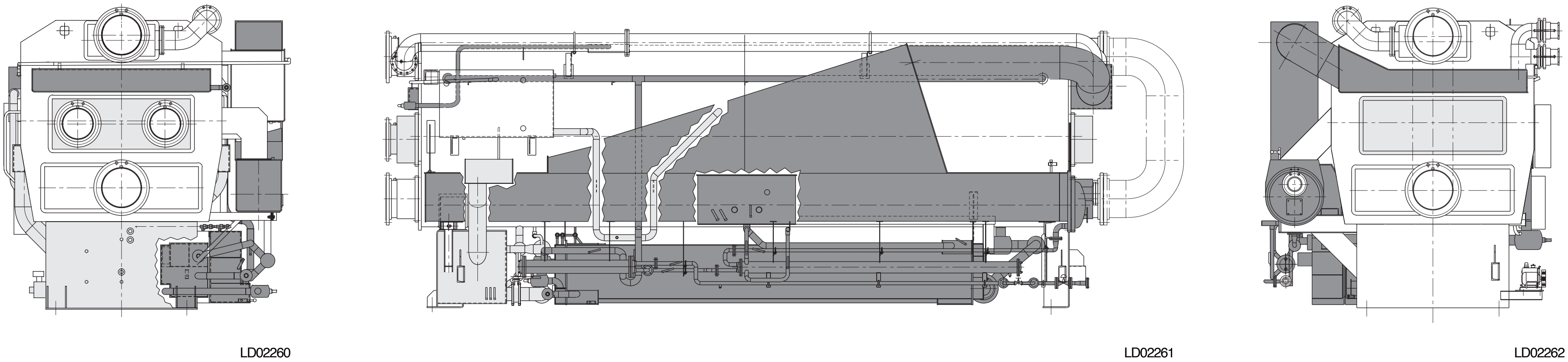
FRONT

RIGHT END

FIG. 33 – 21G STEAM-FIRED



BACK



LEFT END

FRONT

RIGHT END

FIG. 34 – 22G STEAM-FIRED

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Form 155.17-AD1 (897)
Supersedes: Nothing