HYGENAIR™ A+H HYGIENIC AIR HANDLER ENGINEERING MANUAL

COLMAC COIL
Manufacturing Inc.

HYGENAIR™ A+H HYGIENIC AIR HANDLER ENGINEERING MANUAL 1ST EDITION

Bruce I. Nelson, P.E.

HYGENAIR™ A+H AIR HANDLER ENGINEERING MANUAL

Colmac Coil Manufacturing, Inc.

Table of Contents

I. Introduction

Heating, ventilation, and air conditioning (HVAC) systems in food processing facilities function to maintain the temperature and humidity in the production spaces. Day-to-day sanitary operations are dependent upon a properly functioning system for prevention of condensation as well as overall employee comfort. In addition, it is desirable to create positive air pressure differentials in critical or sensitive food handling rooms (e.g., packaging rooms). Because of these demands, the facility should have properly sized air handling units and an adequate distribution system to do the job. Because HVAC systems have proven to be a source of contamination with pathogenic microorganisms, certain sanitary construction, design, and installation features need to be considered. Systems should be constructed, designed, installed, cleaned, and maintained so that they are not a source of contamination. For example, the intake air supply should be located to not draw air from nearby sources of contamination (e.g., chemicals, bird droppings); adequate filters should be installed (and changed at appropriate intervals); and duct work should be located outside of the processing areas. Finally, air handling systems should be designed to be completely cleanable. Interior surfaces should be sealed to provide for contamination control, and final air filtration is recommended.

In the United States, the Department of Agriculture (USDA) regulates the production of meat, poultry, and egg products. USDA-regulated businesses are required to always have a USDA inspector onsite. The Food and Drug Administration (USFDA) regulates the production of everything else (non-meat foods, unusual meats, and fish products).

U.S. food processing firms have been subject to manufacturing/processing rules or standards for decades, but the standards continue to evolve. The development and application of Hazard Analysis and Critical Control Point (HACCP) principles to seafood, juice, meat, and poultry processing has been a significant event since the 1990s. More recently, Congress expanded its perception of HACCP principles and directed that all other food processors adopt such practices by developing and implementing a Food Safety Plan. Appendix A provides an overview of past Good Manufacturing Practices (GMP), Sanitary Standard Operating Procedures (SSOPs), HACCP principles, and the components envisioned to be included in a Food Safety Plan.

The Code of Federal Regulations (CFR) Title 21 Part 117 “Good Current Manufacturing Practice, Hazard Analysis, Risk-Based Preventative Controls for Human Food” is often cited in relation to USFDA guidance on airflow, cross-contamination, temperature, humidity management, etc. in food processing cleanrooms. Hazard analysis and risk-based preventative controls leading to a written food safety plan are also important topics covered by Part 117. Selected excerpts related to the room air handling system from Part 117 are shown below. Note the repeated reference to allergen cross-contact and contaminations in many of the sections, which supports the importance of cleanroom pressurization.

US Code of Federal Regulations (CFR), Title 21, Chap I, Subchapter B, Part 117

Subpart B: Current Good Manufacturing Practice

§117.20 Plant and Grounds

(b) Plant construction and design

(6) Provide adequate ventilation or control equipment to minimize dust, odors and vapors (including steam and noxious fumes) in areas where they may cause allergen cross-contact or contaminate food; and locate and operate fans and other air-blowing equipment in a manner that minimizes the potential for allergen cross-contact and for contaminating food, food-packaging materials, and food-contact surfaces.

§117.80 Processes and controls

(a) General

(4) Adequate precautions must be taken to ensure that production procedures do not contribute to allergen cross-contact and to contamination from any source.

(b) Manufacturing Operations

(2) All food manufacturing, processing, packing, and holding must be conducted under such conditions and controls as are necessary to minimize the potential for the growth of microorganisms, allergen cross-contact, contamination of food, and deterioration of food.

(3) Food that can support the rapid growth of undesirable microorganisms must be held at temperatures that will prevent the food from becoming adulterated during manufacturing, processing, packing, and holding.

(4) Measures such as sterilizing, irradiating, pasteurizing, cooking, freezing, refrigerating, controlling pH, or controlling a_w that are taken to destroy or prevent the growth of undesirable microorganisms must be adequate under the conditions of manufacture, handling, and distribution to prevent food from being adulterated.

(5) Work-in-process and rework must be handled in a manner that protects against allergen cross-contact, contamination, and growth of undesirable microorganisms.

(6) Effective measures must be taken to protect finished food from allergen cross-contact and from contamination by raw materials, other ingredients, or refuse. When raw materials, other ingredients, or refuse are unprotected, they must not be handled simultaneously in a receiving, loading, or shipping area if that handling could result in allergen cross-contact or contaminated food. Food transported by conveyor must be protected against allergen cross-contact and against contamination as necessary.

(10) Steps such as washing, peeling, trimming, cutting, sorting, and inspecting, mashing, dewatering, cooling, shredding, extruding, drying, whipping, defatting, and forming must be performed to protect food against allergen cross-contact and against contamination. Food must be protected from contaminants that may drip, drain, or be drawn into the food.

(12) Batters, breading, sauces, gravies, dressings, dipping solutions, and other similar preparations that are held and used repeatedly over time must be treated or maintained in such a manner that they are protected against allergen cross-contact and against contamination and minimizing the potential for the growth of undesirable microorganisms.

(13) Filling, assembling, packaging, and other operations must be performed in such a way that the food is protected against allergen cross-contact, contamination, and growth of undesirable microorganisms.

(14) Food, such as dry mixes, nuts, intermediate moisture food, and dehydrated food, that relies principally on the control of a_w for preventing the growth of undesirable microorganisms must be processed to and maintained at a safe moisture level.

To help facilitate compliance with these requirements, HygenAir™ hygienic air handling units have been designed to condition air supplied to food processing rooms with features which are fully cleanable, and perform the following important functions:

- Air filtration
- Air conditioning (temperature control)
- Room pressurization
- Humidity control

Air Filtration

Filtration of air that is introduced to hygienic food processing rooms is one of the critically important functions of hygienic air handling equipment. Air can carry several contaminants which are undesirable in a food processing environment. These contaminants are classified as either particulate or gaseous types and can be introduced to the air circulated in the room either with outdoor air or from various processes and activities within the conditioned space. Particulate contaminants can be removed from the airstream with filter media mounted within the air handler. Colmac offers filters with a wide range of filtration efficiencies up to and including HEPA filtration. Particulate filtration is normally accomplished in the air handler via pre-filters which remove larger particles, normally with MERV 8 or 9 rating. Optional intermediate filters with MERV rating between 9 and 13 can be added downstream of the pre-filters to extend the replacement frequency of the more expensive final filters. The supply air downstream of cooling and reheat coils is filtered with final filters having MERV rating of 11 to 16. HEPA H13 filters are also available as final filters for processing rooms requiring very high cleanroom or hospital surgery level filtration. Note that Colmac hygienic air handling equipment is not designed or intended to be used for the removal of gaseous contaminates, such as volatile organic compounds (VOC) and inorganic gases. Removal of gaseous contaminates is possible with other types of equipment, such as packed adsorbent beds, and ionization. The focus of the manual will be on filtration and removal of particulate contaminants.

Air Conditioning

Another important function performed by the hygienic air handler is conditioning (cooling and/or heating) of the air introduced to the processing room. During operation in the process mode, air is normally cooled and dehumidified by cooling coils in the air handler to cool summertime outdoor air, and remove the heat and moisture added to the airstream by human activities and equipment in the processing room. During the sanitation mode, return air dampers are closed, and fresh air and exhaust dampers are fully opened so that 100% outdoor air is supplied to the room to quickly pick up the moisture from cleaning and sanitizing activities and then exhaust it to the outdoors. The air handler is equipped with a heater, normally gas-fired, to maintain a comfortable supply air temperature to the room during winter months in sanitation mode.

Room Pressurization

By introducing an appropriate amount of outdoor air to the room during processing mode, the air handler maintains a positive pressure in the room to prevent infiltration of contaminated air from other parts of the facility. Determining the correct amount of outdoor air is an important design consideration which is discussed elsewhere in this manual.

Humidity Control

Finally, Colmac hygienic air handlers are designed to control humidity in the processing room by using the normal cooling and dehumidifying function of cooling coils in combination with controlled reheat to provide the desired room relative humidity. A unique option available from Colmac is a proprietary

passive reheating heat exchanger which can reduce the refrigeration compressor power required during dehumidifying by as much as 40 to 50% compared to using active reheat coils. Control of supply air relative humidity is important for several reasons, including:

- Insuring proper operation and long life of the final filters
- Maintaining room relative humidity appropriate for processing activities
- Minimizing growth of pathogens and contaminants
- Reducing the risk of contamination of food products by condensation

II. Components and Construction

HygenAir™ air handlers are designed and constructed using materials and components which facilitate the cleaning and sanitizing processes required by food processors to control pathogens in their facilities. The USDA requires that equipment used in food processing facilities be cleaned and sanitized at appropriate intervals to limit the risk of introduction of pathogens to food products to acceptable levels. Cleaning and sanitizing must be performed as two distinct processes. Cleaning the equipment involves the removal of food particles and soils which, combined with moisture, can form harborage spaces or “niches” where pathogens can multiply. To facilitate effective cleaning, air handling equipment must be designed such that all surfaces in and around various components can be accessed by sanitation workers. Inaccessible spaces and areas, and closed cavities (called “hollow bodies”) must be avoided by design. Flat surfaces within the air handler which can allow for pooling of water, must also be avoided. i.e., All surfaces must be pitched to allow for drainage of moisture to the unit drainpans. The second required step, sanitizing, involves applying chemicals which inactivate (kill) pathogens after cleaning. Both cleaning and sanitizing processes typically involve the use of chemicals which in high concentrations can be corrosive to many metals and other materials of construction. Cleaning chemicals may be alkaline (pH > 7) to remove organic soils (fats and oils) or acidic (pH < 7) to remove inorganic soils (mineral scale). Sanitizing chemicals are typically acidic and may consist of combinations of compounds including sodium hypochlorite (chlorine bleach). Whenever cleaning and sanitizing chemicals are applied to the interior of air handlers, correct solution concentrations and temperatures, following of application instructions, and proper rinsing with clean water are critical to avoiding corrosion of the various components in the air handler. Materials of construction typically found in HygenAir™ Air Handlers include:

- 304L Stainless Steel: Interior panels, drainpans, supply fan frames, coil tubes and manifolds, direct-fired burners.
- Aluminum: Supply fan wheels, coil fins, filter frames, dampers.
- Copper Alloys: Coil tubes and fins
- Carbon Steel and/or Cast Iron: Supply fan motor frames (cast iron) and shafts (carbon steel).
- Elastomers: Wire insulation and glands, door gaskets and seals.

Suppliers of cleaning and sanitizing chemicals should be consulted and informed regarding the various materials of construction found in the air handling equipment to ensure that corrosion problems are avoided. Note that Colmac can provide, in many cases, optional or alternative component construction (all stainless steel, for example) required to resist degradation by many commonly used cleaning and sanitizing chemicals.

III. Refrigeration

As mentioned above, one of the important functions of the hygienic air handler is to remove heat from the airstream which has been added by the room load and, in summertime, by the outdoor air being introduced to pressurize the room. This heat is removed by the cooling coils which are piped to an external refrigeration system. The type of refrigeration system may be direct or indirect. Direct refrigeration systems use refrigerants which are volatile, that is, the refrigerant evaporates (boils) inside the coil tubes and absorbs heat from the airstream in the process. This evaporated refrigerant is piped to

a compressor, then to a condenser, and on to an expansion device which feeds the refrigerant into the cooling coil, forming a closed piping loop. Coils in these direct refrigeration systems are called evaporators and can be configured in several ways as explained below. Indirect refrigeration systems use a second ("secondary") refrigerant which is pumped from a chiller to the cooling coils and back again. This secondary refrigerant may be a single-phase fluid such as propylene glycol, or it may be a volatile secondary refrigerant such as CO2. Cooling coils for use with secondary refrigerants must be configured appropriately for the refrigerant used.

Direct Refrigeration Systems

The refrigerant in direct refrigeration systems can be fed to the evaporator coils in one of three ways:

1. Gravity Flooding
2. Pump Recirculated
3. Direct Expansion

Gravity Flooded Evaporators

With gravity flooded evaporator feed, liquid refrigerant is expanded through a make-up valve into a gravity separation vessel (the "surge drum") mounted above the evaporator coil(s). The make-up valve is controlled to maintain an operating liquid level in the surge drum via a liquid level sensor. Liquid from the bottom of the surge drum is fed to the evaporator coils below through individual hand expansion valves. As the refrigerant boils in the evaporator coil, it rises due to buoyant forces and returns to the surge drum through a vertical suction riser pipe. This natural circulation of the refrigerant is referred to as the thermosyphon effect and depends on the correct elevation of the surge drum above the evaporator coils, and the sizing of the descending liquid line, expansion valve, and ascending suction riser, to balance the static pressure created by the height of the liquid column in the liquid line against the frictional pressure drops and opposing static pressure in the suction riser. Figure 1 below shows a gravity flooded evaporator, surge drum, and associated piping.

Up to three evaporators can be fed and controlled from a single surge drum. The use of motorized ball valves installed at the outlet of each evaporator allows evaporating pressure (and temperature) to be independently controlled for each evaporator. The importance of this feature will be explained below in the section on bypass air defrost.

Gravity flooded control of evaporators offers the following benefits compared to other refrigerant feed methods:

- Simple, automatic controls
- Excellent capacity control over a wide load range
- No refrigerant pumps are required
- Simple hand expansion valves are used to regulate flow to each evaporator

The main disadvantage of gravity flooding evaporators is the additional refrigerant charge needed for proper operation. Gravity flooded evaporators must be designed with large diameter tubes (7/8 to 1") to operate at the low pressure drops (0.5 to 1 psi) needed for proper thermosyphon operation. The larger coil tubes required along with the added volume of liquid in the surge drum and liquid leg increase the refrigerant charge in these evaporators compared to pump recirculated and direct expansion evaporators.

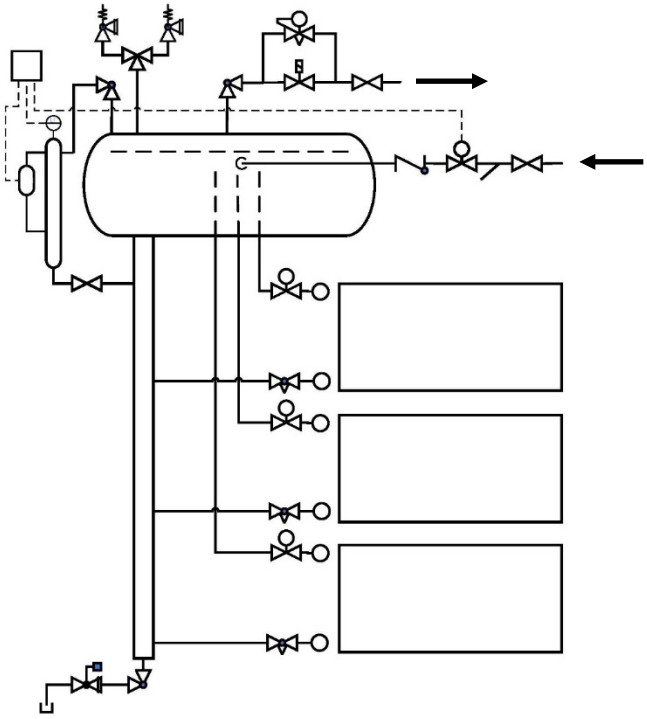
Gravity flooding of evaporators should be avoided with refrigerant blends which exhibit fractionation, or "glide". These blends are referred to as "zeotropic" mixtures and are designated with a "4" as the leading digit in the refrigerant number, for example R404A is a zeotropic refrigerant blend with glide. When used with gravity flooding, zeotropic blends will fractionate (the most volatile component of the blend will boil

first) such that the composition of the liquid remaining in the surge drum will shift over time toward the less volatile components, increasing the evaporating temperature and reducing evaporator capacity. Only pure refrigerants such as ammonia should be used in gravity flooded systems.

Gravity flooding should also be avoided for refrigeration systems and refrigerants which use miscible compressor oil. i.e., Only apply gravity flooding with refrigerants which are either oil-less or use immiscible oil (such as ammonia). In a system which uses miscible oil, some percentage of oil will be in solution with the refrigerant liquid (usually 2-5%) and will circulate throughout the system. The surge drum will act as a distillation vessel retaining whatever oil is carried there with the makeup liquid. It is possible to add an oil rectifier heat exchanger which uses heat to further distill oil that has accumulated in the surge drum to allow it to be returned to the compressors, but this approach is complicated and may have reliability issues returning oil during low loads or in the wintertime.

Colmac Coil makes use of software tools developed specifically to design gravity flooded evaporator circuiting, surge drums, and interconnecting piping and control valves. Surge drums sized properly to account for liquid surge and correct separation velocities can also be provided by Colmac.

Figure 1: Gravity Flooded Evaporators and Surge Drum Piping



Pump Recirculated Evaporators

Pump recirculated evaporator feed is very common in industrial refrigeration systems. With this type of evaporator feed, liquid refrigerant from the condenser is first fed into a vessel (“low pressure receiver” or “recirculator vessel”) through a make-up valve controlled on liquid level in similar fashion to the surge drum in a gravity flooded system. The liquid taken from the bottom of the receiver vessel is fed to refrigerant pumps which pressurize a liquid line which then feeds multiple evaporator coils. The amount of liquid refrigerant fed to each coil is controlled by individual hand expansion valves. The refrigerant pumps are sized to deliver more liquid to the evaporator coils than will be completely evaporated, hence these

systems are also referred to as “liquid overfeed”. The ratio of the total mass flow of refrigerant delivered to the evaporator divided by the mass flow of vapor generated in the evaporator is referred to as the “recirculation ratio”, or the “circulation number”, and is calculated as follows.

$$Recirculation\ Ratio, n = \frac{\dot{m}_{tot}}{\dot{m}_{vap}} \quad [1]$$

$$\dot{m}_{tot} = \dot{V}_{liq} \cdot \rho_{liq} \quad [2]$$

$$\dot{m}_{vap} = \frac{\dot{q}}{h_{fg}} \quad [3]$$

where:
$\dot{V}_{liq} = volumetric\ flowrate\ of\ liquid\ supplied\ to\ evaporator, ft^3/h$
$\rho_{liq} = density\ of\ liquid, lbm/ft^3$
$\dot{q} = evaporator\ capacity, Btu/h$
$h_{fg} = latent\ heat\ of\ vaporization, Btu/lbm$

The recirculation ratio is established by the refrigeration system designer and typically varies between 1.5 to 4. It affects the design of the evaporator coils; hence this is important information which must be provided at the time the air handler is selected.

Since incomplete evaporation is taking place in the evaporator the suction line returning to the recirculator vessel will contain both liquid and vapor, called “two-phase flow”. Because some liquid is present, suction lines in pump recirculated systems are also called “wet” suction lines. The fraction of the mass of refrigerant flowing in this wet suction line which is vapor is referred to as the “vapor mass fraction” or “quality” and can easily be calculated as the reciprocal of the recirculation ratio.

$$Vapor\ Mass\ Fraction\ (quality), x = \frac{1}{n} \quad [4]$$

As with gravity flooded evaporators, use of refrigerant blends which exhibit fractionation, or “glide” should be avoided with pump recirculation. These blends are referred to as “zeotropic” mixtures and are designated with a “4” as the leading digit in the refrigerant number, for example R404A is a zeotropic refrigerant blend with glide. When used with pump recirculation, zeotropic blends will fractionate (the most volatile component of the blend will boil first) such that the composition of the liquid remaining in the recirculation vessel will shift over time toward the less volatile components, increasing the evaporating temperature and reducing evaporator capacity. Only pure refrigerants such as ammonia should be used in pump recirculated systems.

Pump recirculation should also be avoided, if possible, for refrigeration systems and refrigerants which use miscible compressor oil. i.e., Apply pump recirculation with refrigerants which are either oil-less, or use immiscible oil (such as ammonia). In a system which uses miscible oil (such as CO2), some percentage of oil will be in solution with the refrigerant liquid (usually 2-5%) and will circulate throughout the system. The recirculator vessel will act as a distillation vessel retaining whatever oil is carried there with the makeup liquid. It is possible to add an oil rectifier heat exchanger which uses heat to further distill oil that has accumulated in the recirculatory vessel to allow it to be returned to the compressors, but this approach is complicated and may have reliability issues returning oil during low loads or in the wintertime.

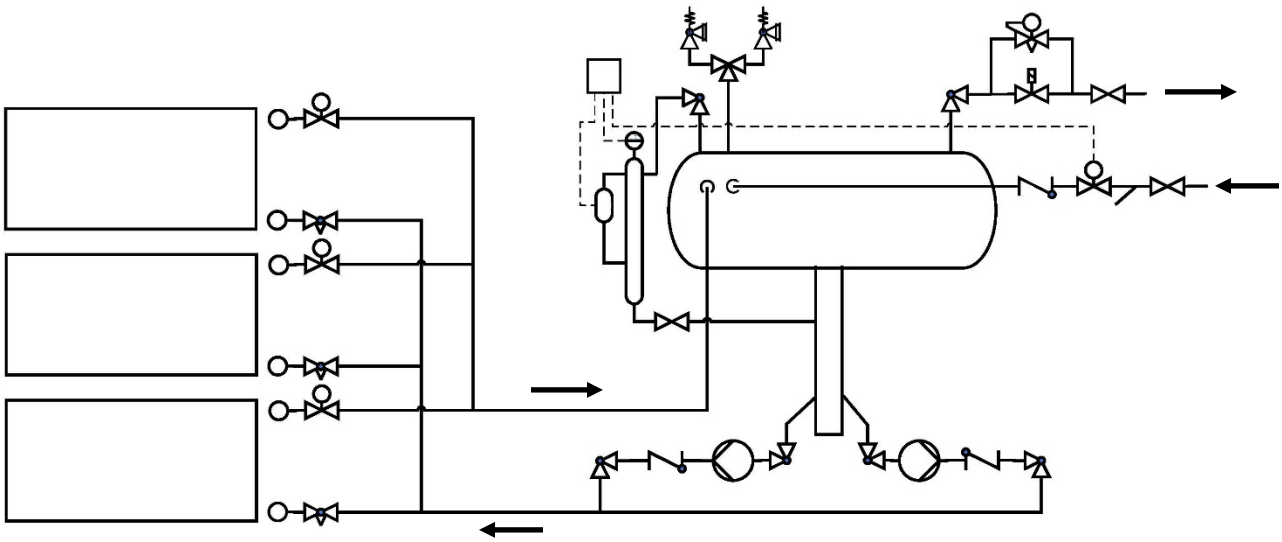
Like gravity flooded evaporators, pressure and therefore evaporating temperature, can be controlled for each individual coil via a motorized pressure regulating valve installed at each coil outlet. Pump recirculated feed of evaporators offers the following benefits:

- Simple, automatic controls

- Excellent capacity control over a wide load range
- Many evaporator coils can be fed from a single receiver
- Simple hand expansion valves are used to regulate flow to each evaporator
- Smaller diameter tubes (lower refrigerant charge) can be used compared to gravity flooding

Figure 2 below shows a typical pump recirculated system with multiple evaporators fed by a single receiver vessel.

Figure 2: Pump Recirculated Receiver and Evaporator Piping



Direct Expansion Evaporators

In a direct expansion ("DX") evaporator, high pressure subcooled liquid is fed directly to each evaporator through an electronic expansion valve as shown in Figure 3. The degree of opening of the electronic expansion valve is regulated by a feedback loop controller which measures and maintains an amount of superheat in the refrigerant vapor leaving the evaporator corresponding to the superheat setpoint. Controlling the flow of refrigerant in this way automatically ensures that the refrigerant fed into the evaporator is completely evaporated and leaves as vapor only. Direct expansion feed eliminates the need for surge drums, and recirculator vessels and pumps found in gravity flooded and pump recirculated systems. Direct expansion can be applied in many types of systems, both large and small scale with both immiscible and miscible compressor oil.

As the refrigerant expands through the electronic expansion valve, some amount of flash gas will form prior to entering the evaporator. This small amount of flash gas entering the evaporator combined with complete evaporation of the refrigerant results in very low refrigerant charge levels in the evaporators compared to both gravity flooding and pump recirculation. This greatly reduced charge level in DX evaporators has been shown to reduce the overall system charge by as much as 75-80% compared to pump recirculation. Line sizes as well as the number and size of vessels in a DX system are also typically smaller than for a pump recirculated system. These two benefits of DX systems, reduced first cost and reduced system charge, have been demonstrated in many commercial and industrial scale projects with all types of refrigerants, including ammonia.

As with gravity flooded and pump recirculated evaporators, pressure, and therefore evaporating temperature, can be controlled for each individual coil via a motorized pressure regulating valve installed at each coil outlet. Direct expansion feed of evaporators offers the following benefits:

- Simple, automatic controls
- Excellent capacity control over a wide load range
- Many evaporator coils can be fed from a single high pressure receiver
- Significantly smaller refrigerant charge compared to gravity flooded and pump recirculated systems

While there are several benefits associated with DX evaporators, there are also several special considerations that must be accounted for in the system design [24].

Subcooling. The liquid leaving the high-pressure receiver must be subcooled to the degree that flash gas does not form in the liquid line upstream of the expansion valve. The amount of subcooling must be sufficient to offset the pressure loss due to lift (elevation gain), frictional pressure drop, and heat gain to the liquid line. If flash gas is allowed to form in the liquid line it will severely reduce the capacity of the expansion valve and effectively “starve” the evaporator coil of liquid reducing system performance.

Water. Water in the refrigerant always penalizes evaporator performance. The presence of even small amounts of water in ammonia causes a large shift in the evaporating temperature and reduction in performance [6]. Liquid to supply DX evaporators should always be taken from the high-pressure receiver and sufficiently subcooled. This approach, particularly with ammonia, insures that the driest refrigerant in the system is delivered to the expansion valves free of flash gas.

Oil. The presence of immiscible oil in the refrigerant (particularly ammonia) reaching DX evaporators will cause a severe reduction in performance due to fouling of the tube surfaces. Effective oil separation is important whether the oil is miscible or immiscible with the refrigerant. DX evaporator performance with small amounts of miscible oil is not pronounced as long as oil concentration in the refrigerant are kept to less than 3-4%.

Dirt and Scale. Regardless of the refrigerant used, dirt and scale from liquid line pipework can plug and foul the control surfaces of expansion valves. Pulse-width expansion valves are particularly sensitive to dirt and scale. Strainers of the mesh size recommended by the expansion valve manufacturer should always be installed directly upstream of the valves and should be cleaned multiple times during startup and commissioning, then at regular intervals thereafter.

Temperature Difference (TD) and Minimum Stable Superheat (MSS). The expansion valves used on DX evaporators normally use a suction superheat signal to control the position of the valve and regulate the flow of refrigerant to the coil. Minimum Stable Superheat is a term used to define the minimum superheat required to avoid control instability and “hunting” (rapid opening and closing of the valve). Most electronic expansion valves and controllers are capable of operation with MSS as low as 5-8 deg F. This implies that the minimum TD (the difference between entering air temperature and saturated evaporation temperature) must be at least 10 deg F. i.e. The TD must be at least 3 to 5 deg F above the evaporation temperature. Designing the refrigeration system to operate with a 12 to 15 deg F TD normally insures stable operation of the expansion valve.

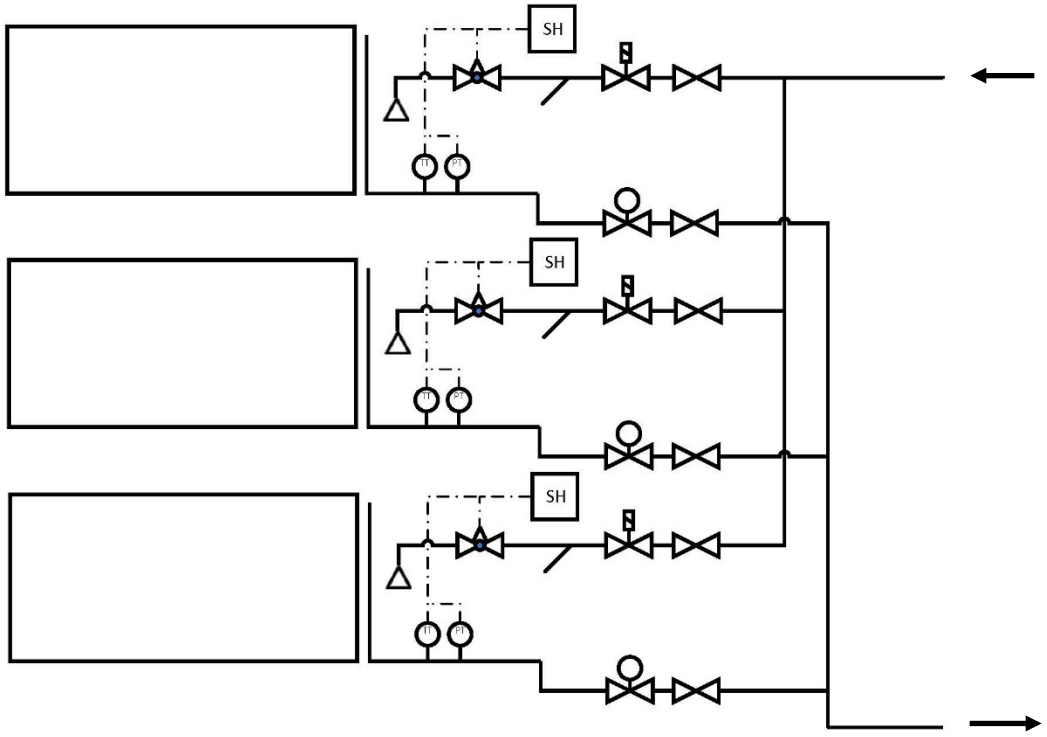
Capacity Control. As the cooling load falls off, capacity of DX evaporators can be controlled by raising the evaporation temperature with an evaporator pressure regulating valve mounted at the

exit of the evaporator (see Fig 3). This approach to capacity control works as long as the TD at reduced load is not allowed to fall below approx. 8-10 deg F to avoid superheat control instability. If capacity turndown of greater than 25 to 30% is required, then multiple expansion valves and coils circuits will be required on each coil.

Gravity Draining Suction Lines. Care must be taken to ensure that the outlet connection of the evaporator coil in the vicinity of the superheat temperature sensor is kept clear of any refrigerant liquid that may leave the coil. i.e., Flooding of the evaporator suction outlet connection must be avoided by properly pitching suction lines back to the compressor, and installing p-traps where necessary. Figure 3 shows the evaporator suction connections and suction lines arranged in such a way to avoid flooding the superheat temperature sensors.

Figure 3 below shows typical direct expansion piping and control valves for multiple evaporators.

Figure 3: Direct Expansion Evaporator Piping and Controls

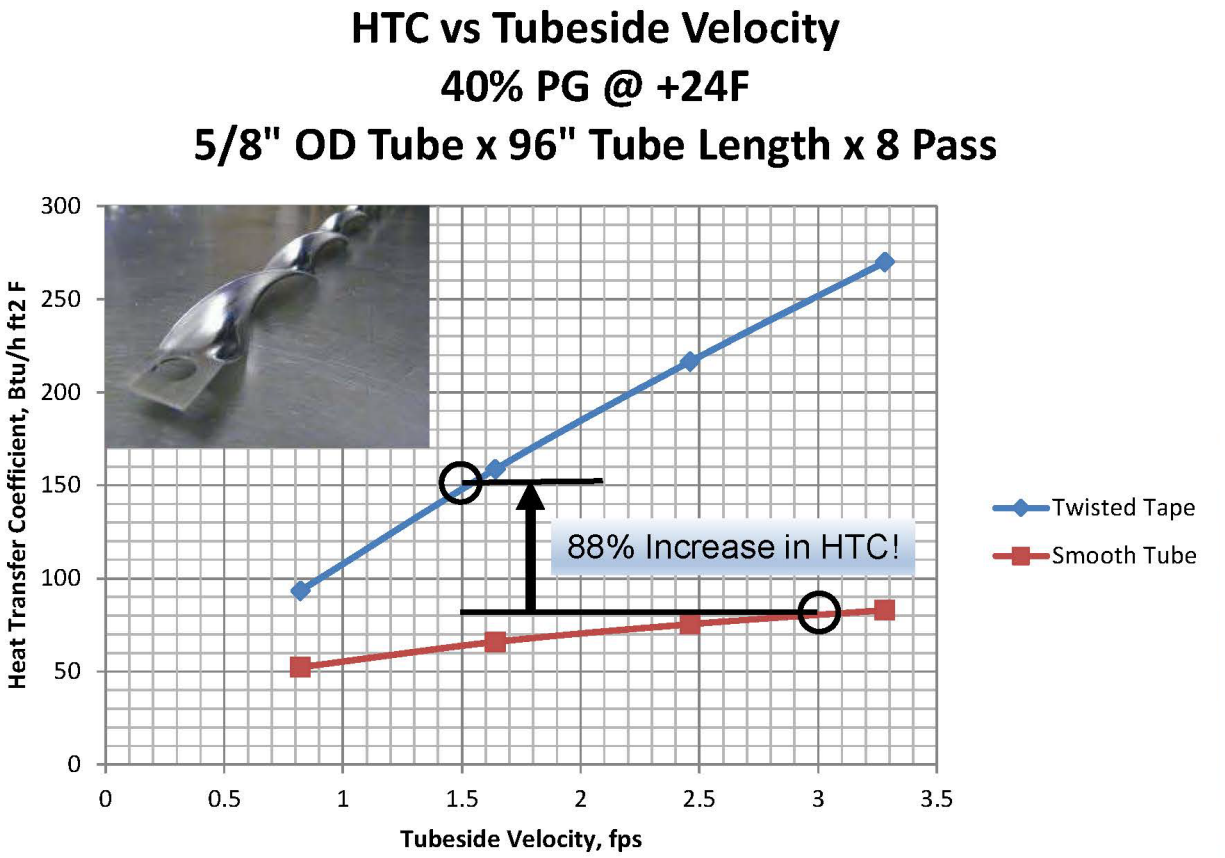


Indirect Refrigeration Systems

Indirect refrigeration systems make use of a second (or “secondary”) refrigerant which is circulated to and from the cooling coils and the system chiller. The secondary fluid can transfer heat in the cooling coils as a single-phase liquid, such as propylene glycol, or as a volatile evaporating refrigerant, such as carbon dioxide. Single phase secondary refrigerant (glycol) systems are typically designed in a similar fashion to a chilled water air conditioning system, which are well covered elsewhere.

To improve the performance of glycol cooling coils, Colmac Coil offers a proprietary tube side heat transfer enhancement device. This unique turbulator device shown in Figure 4 below, significantly increases the tube side heat transfer coefficient when applied in propylene glycol cooling coils and is included in Colmac Coil selection software.

Figure 4: Propylene Glycol Heat Transfer Enhancement



Volatile secondary refrigerant systems are typically designed in a similar fashion to pump recirculated direct refrigeration systems (see above). Volatile secondary systems require the use of a condensing heat exchanger positioned above the recirculator vessel. The condensing heat exchanger acts as an evaporator on the direct refrigeration side and condenses the wet returning refrigerant on the secondary side. The condensed secondary refrigerant drains by gravity into the recirculator vessel which feeds the recirculating pumps which in turn supply pressurized liquid to the cooling coils.

One advantage of these volatile indirect systems is the elimination of circulated oil. That is, the secondary refrigerant is oil-less and as such eliminates the possibility of oil fouling and reduced performance (which can be significant) of the cooling coil surfaces as well as the elimination of the need for oil rectification or oil removal/draining from the recirculator vessel.

Volatile secondary refrigerant systems are particularly attractive compared to single phase systems because of the significant reductions in refrigerant flow rates, line sizes, and pumping power, all resulting

from making use of the latent heat of vaporization to absorb heat in the cooling coils. Carbon dioxide is increasingly being used as a volatile secondary refrigerant because it has low gwp, is non-toxic, non-flammable, and is unregulated.

IV. Cooling Coil Construction

Tubes and Tube Patterns

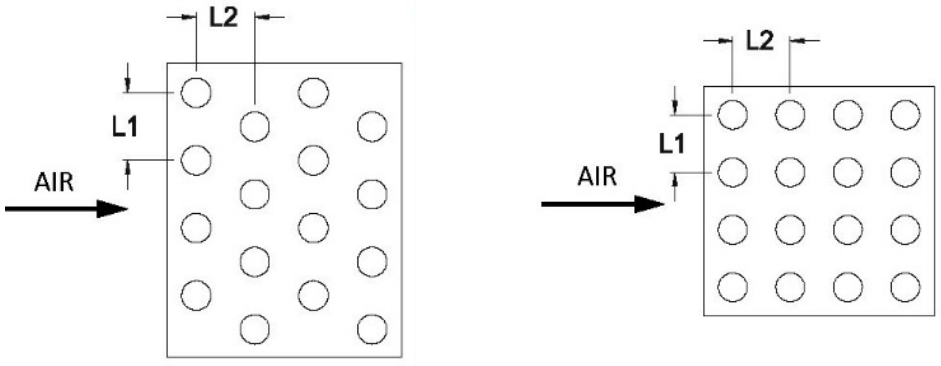
Cooling coils used in hygienic air handlers are inherently one of the most challenging components to clean and sanitize due to the large amount of surface area involved and limited accessibility to the interior surfaces of deep (8 to 16 rows) coils.

Cooling coil tubes can be arranged in either a staggered or inline arrangement in the direction of airflow. Staggered tubes are commonly used in commercial air conditioning cooling coils due to increased heat transfer efficiency, reduced volume, and reduced first cost. Staggered tubes are also inherently more difficult to clean, particularly interior rows of deep coils as mentioned above. Inline tubes, on the other hand offer a number of benefits compared to staggered tubes:

- Interior row surfaces are more accessible and easier to clean
- Lower air pressure drop and fan power for the same cooling capacity
- Increased frost carrying capacity and runtime between defrosts (low temperature rooms)

Colmac Coil HygenAir™ Air Handlers are available with either staggered or inline tube cooling coils. Figures 5 and 6 below illustrate staggered and inline tube differences in arrangement and frost carrying characteristics.

Figure 5: Staggered vs Inline Tube Patterns



Staggered In-Line

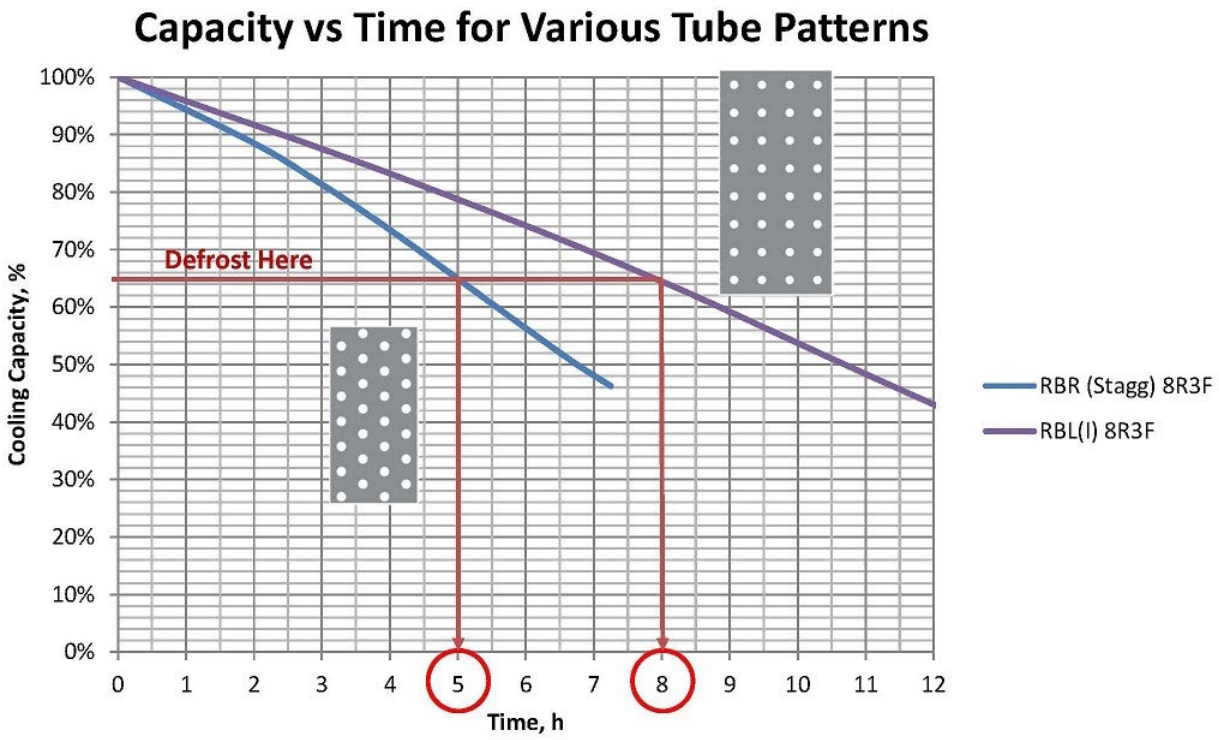
Four different tube patterns are available for cooling coils installed in HygenAir™ Air Handlers, which are designated as shown in Table 1 below.

TABLE 1: TUBE PATTERN DESIGNATIONS

Designation	Tube Diameter, in	L1 Spacing, in	L2 Spacing, in
I	5/8	1.5	1.299
T	5/8	1.97	1.97
M	7/8	2.25	1.949
U	7/8	2.36	2.36

Low temperature rooms will require that the refrigerant temperature, and so the surfaces of the cooling coils, operate below freezing. The frost that consequently accumulates on the coil fins reduces heat transfer and increases air pressure drop over time, requiring that the coils be periodically defrosted. Colmac Coil has developed a unique sequential bypass air defrost system for these frosted coil conditions which allows continuous operation of the air handler without an interruption in service. Nonetheless, maximizing runtime between defrosts is desirable and is made possible by using one of the inline tube patterns shown above (“T” or “U” pattern coils). An example of this difference in runtime between defrosts comparing an inline and staggered tube pattern is shown in Figure 6.

Figure 6: Staggered vs Inline Runtime Between Defrosts



Materials of Construction

Cooling coils in HygenAir™ hygienic air handlers Industrial evaporator coils can be manufactured by Colmac Coil with a number of different tube and fin materials to match the requirements of the working fluid (refrigerant), the operating environment, and the project’s first cost requirements.

Tube Materials

Cooling coil tubes can be supplied made from several different materials. Selection of an appropriate tube material will depend on refrigerant compatibility, required design pressure, and environmental corrosion resistance requirements. The compatibility of various tubing materials with refrigerants is shown below in Table 2.

TABLE 2: REFRIGERANT COMPATIBILITY

Refrigerant	Copper	Aluminum	304/316 SS	Carbon Steel
Ammonia	No	Yes	Yes	Yes
CO2	Yes	Yes (1)	Yes	No (2)
HCFC, HFC	Yes	Yes	Yes	Yes
Glycols	Yes	No (2)	Yes	Yes
CaCl	No	No	No	Yes (3)
Potassium Formate	Yes	No	Yes	No

NOTES:
1. The relatively low strength of aluminum limits its application only to low operating temperatures and pressures with CO2.
2. Potential for corrosion exists.
3. To avoid corrosive conditions: (a) avoid exposure of the CaCl solution to air, (b) maintain saturated solution concentration (approx. 30%), (c) maintain 8.5 < pH < 9.5 and avoid contamination.

Fin Materials

- Aluminum (1100, 8006, AlMg2.5)
- Epoxy Coated Aluminum
- Copper
- Anti-Microbial Alloy
- Stainless Steel (304L, 316L)

Available Combinations (Tube/Fin):
- Stainless Steel / Aluminum
- Stainless Steel / Epoxy Coated Aluminum
- Copper / Aluminum
- Copper / Epoxy Coated Aluminum
- Copper / Copper
- Stainless Steel / Anti-Microbial
- Stainless Steel / Stainless Steel

V. Corrosion

Cooling coils used in hygienic air handlers may come into contact with food, food additives, and food seasonings. All foodstuffs are acidic to varying degrees. Common types of acids in food include:

- Acetic Acid (fruits and berries, as well as vinegar)
- Citric Acid (citrus fruits)
- Fatty Acids (fats and oils)
- Lactic Acid (sour milk products, sourdough bread making)

When onions are sliced, they release sulfur-laden gases which oxidize and combine with water to form sulfuric acid – this is what causes the stinging sensation in your eyes. Onions also contain phosphoric acid. The pickling of vegetables with vinegar (acetic acid) can also produce an operating environment that is quite acidic.

Food additives and seasonings, along with various chemicals that are applied to foods to manage discoloration, shelf life, etc., are too numerous to mention here but can alone or in combination create corrosion conditions. It is important to identify and understand the chemistry of these additives and seasonings and factor them into the coil material selection process. One of the most common seasonings used for flavoring as well as preservation of food is sodium chloride. It will be seen below that certain metals, while they have good corrosion resistance over a wide range of pH conditions, exhibit pitting corrosion in the presence of chloride ions.

Corrosive conditions can also result from the fumigation of rooms with gases for disinfection. Chlorine dioxide is a gas used on equipment and in rooms to disinfect and sterilize surfaces. If not applied correctly, severe damage to stainless steel surfaces can result in the form of pitting and metal loss.

In general, aluminum, copper, and stainless steel all exhibit good corrosion resistance to mildly acidic conditions while galvanized steel will quickly corrode when exposed to the same conditions.

Cleaning and Sanitizing

The cleaning process is defined as the removal of organic soil (fats and oils) and/or inorganic soil (mineral scale or stains). The sanitizing process is defined as treating cleaned surfaces to effectively kill or remove pathogens. The USDA requires that these two processes be done separately.

Cleaning and sanitizing chemicals used in the food processing industry fall into four general categories:

1. Acidic
2. Strongly Alkaline
3. Mildly Alkaline
4. Chlorine Based

There are many manufacturers of cleaning and sanitizing chemicals, each manufacturer having its own array of complex and proprietary formulations designed for specific foodstuffs and processes. It is not the intent of this article to give even a cursory list of manufacturers and their products, but instead to describe in general terms the effects of categories of chemicals and their pH on various metals used in evaporator coils.

The effect of these cleaning and sanitizing chemicals on the metal surfaces of coils will be determined by several factors:

- Concentration of the solution

- Proper and complete application
- Correct sitting/soaking time
- Correct temperature
- Complete and thorough rinsing with clean water after cleaning and sanitizing

Corrosion Resistance of Metals

Aluminum

Aluminum is a lightweight, strong, thermally conductive metal which can be alloyed with a number of different elements to produce variations in strength, ductility, weldability, and so on. It is in wide use in automobile, aircraft, and shipbuilding industries, it is used as a building material, for beverage containers, baseball bats, and so on. Because of its lightweight, high thermal conductivity, strength, and corrosion resistance, it is used extensively in heat exchangers of many kinds. It is compatible with all refrigerants including ammonia, in fact anhydrous ammonia naturally passivates aluminum surfaces.

Figure 7: Aluminum fins severely corroded by application of highly alkaline (pH > 9.0) cleaning chemicals.

In the presence of air, aluminum oxide forms very quickly to protect the surface of the metal. This oxide layer is very stable and tenacious and is very resistant to corrosion when 4.0 < pH < 9.0. However, aluminum oxide dissolves and the surface of the metal will corrode quickly, seen as pitting and metal loss, when exposed to strong alkaline cleaners (pH > 9.0) such as sodium hydroxide (caustic soda). The oxide layer is also attacked by highly acidic (pH < 4.0) cleaners and chlorine-based sanitizing chemicals.

Stainless Steel (304L, 316L)

Stainless steel is widely used in the food processing industry because of its high tensile strength and corrosion resistance. The chromium in stainless steel forms a very dense passive film layer which is generally very stable over a wide pH range. The oxide resists corrosion by strong alkaline cleaners such as sodium hydroxide (caustic soda) and in the presence of most acids.

While stainless steel has a reputation for being practically indestructible and able to handle any corrosive environment, the passive layer on stainless steel can be attacked by certain chemical species. The chloride ion Cl- is the most common of these and is found in everyday materials such as salt and bleach. The halogen salts, primarily chlorides such as sodium chloride, penetrate the passive oxide layer and can result in pitting and/or stress corrosion cracking. Exposure to sodium hypochlorite (chlorine bleach), or hydrochloric acid solutions, in high enough concentrations will result in pitting and/or stress corrosion

cracking. It is interesting to note that this susceptibility of stainless steel to chlorides is practically independent of pH.

Copper

The oxide layer on copper surfaces forms quickly in the presence of air and is very stable over a wide pH range. The color of the oxide layer ranges from brown to green and is referred to as “patina”. Copper is cathodic to most metals (steel, stainless steel, aluminum, and zinc) and so galvanic corrosion typically proceeds very slowly.

Stress corrosion cracking appears quickly when copper is exposed to ammonia and aqua-ammonia. This obviously makes the metal unsuitable for use in piping, heat exchanger tubing, and other components in direct contact with ammonia.

Most copper tubing is joined by brazing with silver-bearing alloys. While the copper is resistant to acidic environments, the components of the brazing alloy – specifically phosphorus – may not be. For example, in low pH environments such as grape storage where sulfur dioxide is used in fumigation, or in citrus ripening rooms, it is important to specify that only phosphorus-free brazing alloys be used for joining copper tubes in coils.

Anti-Microbial Fin Alloy

This proprietary Cu-Ni metal alloy, when used to make fins, exhibits corrosion resistance similar to stainless steel but with conductivity and thermal performance similar to aluminum. This special alloy also displays active anti-microbial behavior. It has been shown by testing that colony counts of cultured E.coli bacteria placed on the surface of this fin material approach zero after only 3 hours [17]. *Pathogens cannot exist on the surface of this fin material.*

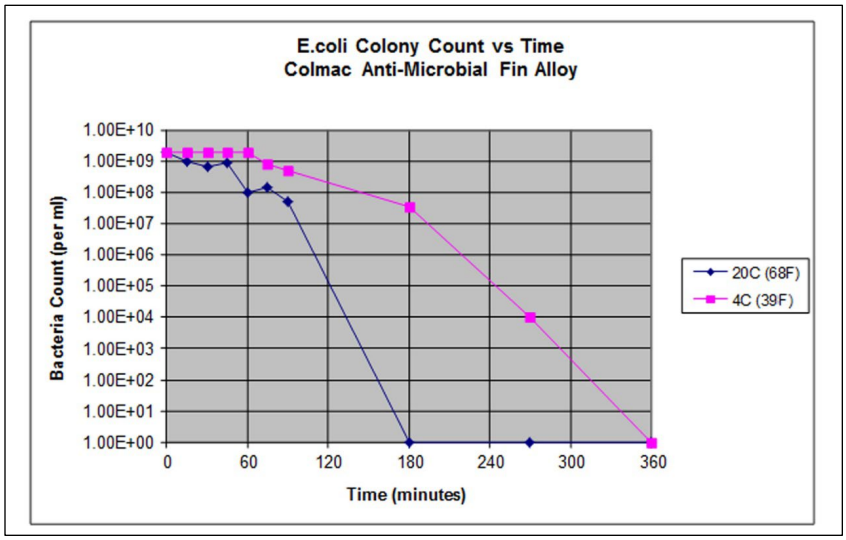


Figure 8: Colmac Anti-Microbial fin alloy actively kills bacterial pathogens [17]

Galvanized Steel

In the past industrial
ammonia evaporators were traditionally made by hot dip galvanizing (application of molten zinc) carbon steel tubes and fins after fabrication. The zinc oxide layer forms quickly in air on coil surfaces and is stable with the range $7.0 < pH < 12.0$.

Galvanized steel construction corrodes very quickly when exposed to acidic solutions, even mildly acidic, such as sodium hypochlorite. Consequently, this type of construction is very difficult to sanitize and so has fallen out of favor in recent years with the increased focus on controlling pathogens.

However, because of the corrosion resistance of galvanized steel to highly alkaline cleaners such as sodium hydroxide (caustic soda) this type of coil construction is still popular in some meat processing facilities where heavy accumulations of fat and oils must be removed during cleaning.

Epoxy Coated Aluminum

Colmac Coil offers fins made from epoxy coated aluminum fin stock as an option on all evaporator coils and air coolers.

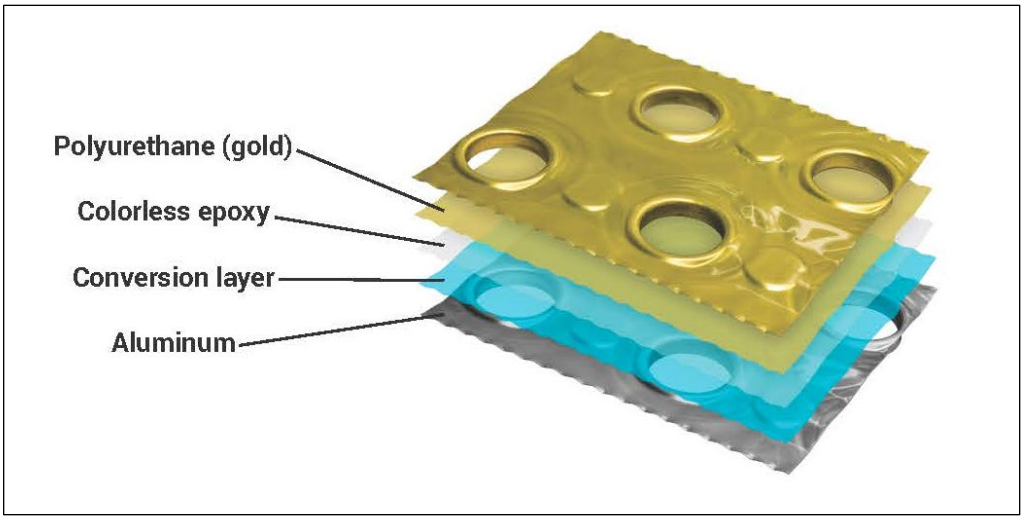


Figure 10: Colmac Epoxy Coated Aluminum fin stock coating

The coated fin material offers the following benefits:

- Over 300% more resistant to corrosion than bare aluminum fins.
- High thermal conductivity with negligible impact on heat transfer.
- Flexible - will not peal crack or chip.
- Highly resistant to abrasion.
- Coating does not support growth of micro-organisms.

The coating system (EPPU) consists of the base aluminum alloy prepared with a chemical conversion layer coated with a colorless epoxy layer and a final polyurethane top coating.

Testing of the coating has shown the following results.

- Heat Resistance: 200 deg C/5 min
- Solvent Resistance: Trichlorethlene (85 deg C/5 min), Perchlorethylene (120 deg C/30 min)
- Salt Spray Test (ASTM B117): >1,000 h
- Kesternich Test (ISO 3231): > 15 cycles
- Humidity Test (DIN 50017): No degradation, no corrosion
- UV Resistance (ASTM G154): OK

Cleaning and Sanitizing Chemicals

Two general groups of cleaning chemicals, alkaline and acidic, are used to remove soil from coil surfaces. Alkaline cleaners are effective at removing fats and grease while acidic cleaners are used to remove minerals and scale. When cleaning coil surfaces these cleaners are usually supplied in a foaming format which causes the chemicals to adhere and 'stick' to vertical surfaces to keep the chemicals in contact longer.

The amount and concentration of the cleaning chemicals used is important along with the temperature of the solution. Longer soaking time requires lower concentration of the chemical. Warmer water/solution temperatures also require lower concentration of the chemical. Rinsing with clean water is critical after cleaning and sanitizing to avoid accumulation and high concentrations of the chemical collecting in crevices and initiating corrosion sites. Even though chemicals are applied in the correct concentration with pH in the acceptable range, neglecting to rinse thoroughly after cleaning and sanitizing can result in increased concentrations of chemicals in crevices as coil surfaces dry out resulting in crevice corrosion. Proper rinsing is critical!

Sodium hypochlorite (chlorine bleach) is widely used as a sanitizing chemical and is effective at killing pathogens on metal surfaces. Unfortunately, high concentrations of sodium hypochlorite will cause pitting corrosion on stainless steel surfaces. Also, if the pH of the sodium hypochlorite solution is allowed to fall below 4.0 then aluminum oxide will begin to dissolve, and aluminum surfaces will corrode. Therefore, sodium hypochlorite is not recommended for use on any Colmac Coil evaporator coil.

Fortunately, there are numerous cleaning and sanitizing chemicals available which are compatible with the various types of evaporator coil construction listed above. A few are listed below according to the metal surface being cleaned and sanitized. WARNING: The supplier of the cleaning and sanitizing chemicals to be used must be consulted and their application guidelines carefully followed regarding chemicals and procedures used for a given type of coil construction. It is the responsibility of the chemical supplier and end user to correctly select and apply cleaning and sanitizing chemicals to avoid damage to coil surfaces due to corrosion.

Recommended Coil Cleaners and Sanitizers

1. Aluminum or Stainless-Steel Tubes with Plain or Epoxy Coated Aluminum Fins
    a. Cleaning
        i. Organic Soil (Fats and Grease)
            1. Use a foaming mildly alkaline cleaner with pH < 9.0
            2. Example: ZEP FS Strike Three (Potassium Hydroxide based)
        ii. Mineral Soil (Minerals and Scale)
            1. Use a foaming mildly acidic cleaner with pH > 4.0
            2. Example: ZEP Formula 7961 (Phosphoric Acid based)
    b. Sanitizing
        i. Spray-on application
            1. Use Peracetic Acid or Quaternary Ammonia with pH > 4.0
            2. Example: ZEP FS Amine A, FS Amine Z
            3. Note: DO NOT use Sodium Hypochlorite-based (Chlorine Bleach) sanitizing chemicals on stainless steel or aluminum surfaces.

2. Stainless Steel Tubes with Stainless Steel or Anti-Microbial Fins
    a. Cleaning
        i. Organic Soil (Fats and Grease)
            1. Use a foaming alkaline cleaner with pH < 12.0

2. Example: ZEP FS Strike Three (Potassium Hydroxide based) or Sodium Hydroxide (Caustic Soda)

ii. Mineral Soil (Minerals and Scale)
1. Use a foaming mildly acidic cleaner with pH > 4.0
2. Example: ZEP Formula 7961 (Phosphoric Acid based)

b. Sanitizing
i. Spray-on application
1. Use Quaternary Ammonia with pH > 4.0
2. Example: ZEP FS Amine Z
3. Note: DO NOT use Sodium Hypochlorite-based (Chlorine Bleach) cleaning or sanitizing chemicals on stainless steel or anti-microbial alloy surfaces.

3. Galvanized Steel
a. Cleaning
i. Organic Soil (Fats and Grease)
1. Use a foaming alkaline cleaner with pH < 12.0
2. Example: ZEP FS Strike Three (Potassium Hydroxide based) or Sodium Hydroxide (Caustic Soda)

ii. Mineral Soil (Minerals and Scale)
1. Very difficult to remove from galvanized steel surfaces
2. DO NOT use Acidic cleaning chemicals!

b. Sanitizing
i. Very difficult to sanitize galvanized steel surfaces
ii. DO NOT use Acidic sanitizing chemicals!

VI. Fin Spacing

Industrial cooling coils can be manufactured with a wide variety of fin spacings, ranging from 2 to 12 fins per inch. Close fin spacings (> 8 fins per inch) are not recommended for use in hygienic air handling equipment due to the difficulty of cleaning and sanitizing interior fin surfaces. Therefore, 4 to 6 fins per inch fin spacing is recommended for hygienic air handler cooling coils to facilitate thorough cleaning. This wide fin spacing combined with an inline tube pattern ('T' or 'U' designation) is the ideal cleanable hygienic coil design. These wide fin spacings are also more efficient for low temperature rooms, when the coils are operating in a frosted condition. The 4 to 6 fins per inch will carry more frost with the result of longer run times between defrosting compared to close fin spacings.

VII. Psychrometrics

Chapter 1 of the ASHRAE Fundamentals Handbook [1] states that "Psychrometrics uses thermodynamic properties to analyze conditions and processes involving moist air." Chapter 1 goes on to offer the following definitions:

> *Atmospheric air* contains many gaseous components as well as water vapor and miscellaneous contaminants.
> *Dry air* is atmospheric air with all water vapor and contaminants removed. Its composition is relatively constant, but small variations in the amounts of individual components occur with time, geographic location, and altitude.
> *Moist air* is a binary (two-component) mixture of dry air and water vapor. The amount of water vapor varies from zero (dry air) to a maximum that depends on temperature and pressure.
> *Saturation* is a state of neutral equilibrium between moist air and the condensed water phase (a flat liquid water surface). At saturation the air contains the maximum amount of water vapor possible and has relative humidity of 100%.

A basic understanding of the properties of moist air along with the psychrometric processes that take place in the processing room and within the air handler are important for proper selection and application of hygienic air handling equipment.

VIII. Properties of Air

Correct selection of the hygienic air handler depends on the accurate determination of air properties. Some properties can be measured directly, such as temperature, other properties must be calculated, such as enthalpy.

The following properties of air are measured directly.

Dry Bulb Temperature, T_{db}. Measured using a (dry) temperature sensor or thermometer.

Wet Bulb Temperature, T_{wb}. Measured using a temperature sensor or thermometer having a wetted gauze sock covering the sensor element or thermometer bulb. Because of the evaporative cooling effect of the wetted sock, the measured wet bulb temperature will normally be lower than the dry bulb temperature. This difference between dry and wet bulb temperatures is referred to as the “wet bulb depression”. Note that as relative humidity falls, the wet bulb depression increases. Conversely, as relative humidity increases, the wet bulb depression decreases until the air becomes saturated (100% relative humidity), at which point dry bulb and wet bulb temperatures are equal and the wet bulb depression is zero.

Barometric Pressure, P_{baro}. Measured using a pressure gage, this is the atmospheric pressure normally measured in inches or mm of mercury. It is sometimes measured in psia or in Pa. Note that barometric pressure changes with elevation and affects important properties of air, such as density and enthalpy. Hence, elevation is an important input to most psychrometric calculations.

The following properties are calculated or determined using tables, or a software routine. Note that “lbm” are units of pounds-mass.

Specific Volume, v. Given as cubic feet per lbm of dry air [ft3/lbm].

Density, $\rho = 1/v$. Given as lbm of dry air per cubic foot [lbm/ft3].

Specific Enthalpy, h. Given as Btu per lbm of dry air [Btu/lbm].

Specific Heat, C_p. Given as Btu per deg F per lbm of dry air [Btu/F-lbm].

Humidity Ratio, H. Given as lbm wtr per lbm dry air [lbm/lbm].

Dew Point Temperature, T_{dp}. Temperature of the air when saturated at pressure P_{baro} , and humidity ratio, H. On a psychrometric chart the dew point temperature can be found at the intersection of the saturation line (100% rh) and humidity ratio line for a given air temperature and humidity.

Relative Humidity, rh. Is a ratio, expressed in percent, of the amount of moisture present relative to the amount that would be present if the air were saturated, [100%].

Note that the above “specific” properties of air (Volume, Enthalpy, Specific Heat, and Humidity Ratio) are given “per lbm of dry air”. What does this mean and why do these properties appear in terms of dry air? Expressing air properties per pound of “dry air”, even though the air is humid, turns out to be a very convenient way to handle the combined heat and mass transfer which occurs whenever moisture is

removed from or added to an airstream. Examples include removing (condensing) moisture from an airstream during air cooling and dehumidifying with cooling coils or adding (evaporating) moisture to an airstream during an evaporative cooling process. The mass of dry air in the airstream does not change during a heating or cooling process even if the amount of moisture does through condensation or evaporation. As will be seen later, making heating and cooling calculations using the dry air mass flow and specific moist air properties expressed per pound of dry air significantly simplifies convective heat transfer calculations.

IX. Heat Transfer Calculations

Heat can be transferred in three ways, by conduction, convection, or radiation. In the case of hygienic air handling, the primary means of transferring heat to or from the processing room is through circulation (convection) of air. As mentioned above, the air being circulated by the air handler will be a mixture of dry air and water vapor. The heat transferred to or from the airstream which results in a temperature change is referred to as “sensible” heat transfer, that is, the heat that can be “sensed”. The heat transferred to or from the airstream which results in the addition or removal of moisture is referred to as “latent” heat transfer, since no temperature change takes place, and the transfer of heat is “hidden”.

The amount of sensible heat transferred by convection is given by:

$$\dot{q}_{sens} = \dot{m}_{da} \cdot C_{p\,ma} \cdot \Delta T_{air} \qquad [5]$$

Where:
$\dot{q}_{sens} = Sensible\ Heat\ Transfer\ Rate, Btu/h$
$\dot{m}_{da} = Mass\ Flowrate\ of\ Dry\ Air = \dot{V}_{sa} \cdot \rho_{sa} \cdot 60, lbm/h$
$60 = Unit\ conversion\ factor, min/h$
$\dot{V}_{sa} = Volumetric\ Flowrate\ of\ Standard\ Air, scfm$
$\rho_{sa} = Density\ of\ Standard\ Air\ (Air\ at\ 70°F\ and\ Sea\ Level) = 0.075\ lbm\ dry\ air/ft3$
$C_{p\,ma} = Specific\ Heat\ of\ Moist\ Air, Btu/lbm°F$
$\Delta T_{air} = Air\ Temperature\ Change\ (Dry\ Bulb), °F$

The amount of total (sensible plus latent) heat transfer is given by:

$$\dot{q}_{tot} = \dot{m}_{da} \cdot \Delta h_{air} \qquad [6]$$

Where:
$\dot{q}_{tot} = Total\ Heat\ Transfer\ Rate, Btu/h$
$\Delta h_{air} = Air\ Enthalpy\ Change, Btu/lbm\ dry\ air$

The Sensible Heat Ratio, SHR, is given by:

$$SHR = \frac{\dot{q}_{sens}}{\dot{q}_{tot}} \qquad [7]$$

The amount of latent heat transfer is given by:

$$\dot{q}_{latent} = \dot{q}_{tot} - \dot{q}_{sens} \qquad [8]$$

The rate of moisture removal during cooling and dehumidifying of the airstream is given by:

$$\dot{m}_{wtr} = \dot{m}_{da} \cdot \Delta H_{air} \qquad [9]$$

Where:
$\dot{m}_{wtr} = Moisture\ Removal\ Rate, lbm\ wtr/h$
$\Delta H_{air} = Air\ Humidity\ Ratio\ Change, lbm\ wtr/lbm\ dry\ air$

X. Room Load

Accurate estimation of the room cooling load along with the outside air requirement for pressurization, is critically important to determining the proper sizing of the air handling equipment and the correct amount of refrigeration needed to maintain room temperature and humidity. Detailed load estimation methods are well covered elsewhere [3][4][5] and should be used for determination of the final design.

Table 3 below shows various "rules of thumb" which can be used for preliminary load and airflow estimating. The following guidelines should be used for estimating and budget purposes only. They are not intended for final design use. Each application must be individually sized and selected based on its own particular requirement and customer specifications.

TABLE 3: GENERAL ROOM LOAD AND AIRFLOW GUIDELINES

Type of Room	Room Temp.	Air Changes per hr	Sq. Ft. Per Ton Cooling	Outside Air in CFM Per Sq. Ft.	Filtration (MERV)	See the Following Notes
Cottage Cheese Vat Room	80°	21	90	0.50	14/15	1,2,3,6,7
Cheese Packaging Room	65°-75°	18	105	0.50	14/15	1,2,3,6,8
Culture/Starter Room	70°-75°	18	95	0.50	HEPA H13	1,4,9
Milk Process	75°	18	90	0.50	14/15	1,2,3,6,28
Milk Filling Room	75°	18	95	0.50	14/15	1,2,3,6,10
Sterile Filling Room	65°-75°	24	50	0.25	HEPA H13	1,4,11
Yogurt Filling	70°-75°	18	100	0.50	14/15	1,2,3,6,12
Ice Cream Process	75°	18	85	0.50	14/15	1,2,3,6,13
Ice Cream Packaging	75°	18	90	0.50	14/15	1,2,3,6,13,29
Blow Mold	65°-70°	24	50	0.50	9	1,2,4,14
Egg Breaking Room	75°	21	95	0.60	14/15	1,2,3,6,15
40°-50° Meat Room	40°-50°	24	100	0.25	14/15	1,2,4,16
General Packaging	75°-80°	18	105	0.50	14/15	1,2,3,6,17
General Food Processing	75°-80°	18	100	0.50	14/15	1,2,3,6,18
Bakery	----	20	N/A	0.50	8	1,2,3,19
Case Room	----	20	N/A	100% O.S. Air	8	5,20
Powder Bagging Room	65°-75°	21	33	100% O.S. Air	14/15	1,21
Boiler Room	----	20	N/A	100% O.S. Air	8	5,22
Engine Room	----	20	N/A	100% O.S. Air	8	5,23
Warehouse	----	12	N/A	0.50	8	5,24
Poultry Eviscerating	75°	20	100	0.50	14/15	1,2,4,6,7,25,30
Poultry Process	45°-50°	24	100	0.25	14/15	1,2,4,16,26,30
Poultry Packaging	45°-50°	24	100	0.25	14/15	1,2,4,16,27
Beef	40°-45°		80		14/15	
Grind/RTE	38°-40°		60-80		HEPA H13	
Cheese Shred	40°-45°		100			Humid.Control

NOTES:

1. The room must be kept under a pressure at all times.
2. The air should be supplied at the ceiling with a diffuser so the air will flow across the ceiling and down the walls.
3. This application should use an economizer cycle so outside air can be used for cooling whenever possible. Exhaust equipment must be used in this system to exhaust when all outside air is being used.

4. This application should use a fixed amount of outside air, just enough for pressurization and ventilation. No exhaust is required.
5. This application should use all outside air and ventilate the room only. King make-up units usually can be supplied with heating as an option. Matching exhaust equipment is also necessary.
6. The application should have a clean-up cycle. At this time, the unit brings in additional volumes of outside air to purge the room of steam and moisture and help dry the walls, floors, and ceilings.
7. A cottage cheese room is a very high moisture area. If the room is not air conditioned, at least 25 air changes per hour should be used. If the area is air conditioned, approximately 15 to 20 air changes per hour is recommended. Generally, the room load is calculated for non-cooking periods. Either the room is not air-conditioned during cooking period or the room temperature is just allowed to rise. Air is generally supplied at the ceiling and returned near the floor.
8. A cheese packaging room should have between 15 and 20 air changes per hour.
9. In a culture room, there should be from 15 to 20 air changes per hour. The area may be conditioned or not depending on customer preference.
10. A milk filling room is generally air conditioned but is not absolutely necessary. The load in the table includes 6-ton air conditioning load per filler.
11. A sterile filling room requires a unit which has absolute filters. The air handling unit can just provide the sterile atmosphere in the room around the filling machine, or it can supply air directly to the shroud over the filler itself. The filler should be in a separate "clean room." The load in the table includes 7 to 7.5 tons per aseptic filler.
12. In a yogurt filling room, we recommend using 95 percent filters since this is a cultured product, similar to cottage cheese. Normally, the room is air conditioned, but it is not absolutely necessary.
13. An ice cream processing room will require between 18 and 20 air changes per hour. Even though we have a refrigerated product, a 95 percent filter is recommended. Air is supplied at the ceiling.
14. A blow mold room is always air conditioned. Generally, the conditions are 70°-72° and between 40%-50% R.H. The humidity is critical since too high a humidity will cause condensation to form on the blow mold heads which in turn will cause spots on the bottles. Blow mold manufacturers recommend a temperature-controlled atmosphere because they can obtain a consistent container quality without continuous machine adjustment. Forty percent filters are normally used for the air handling unit. However, 95 percent of filters are sometimes used. The load shown on the table includes blow mold machine load at 15 ton per machine. It is advisable to have the grinding done in a separate room to eliminate the dust problem in the blow mold room itself.
15. An egg breaking room must be kept under a very definite positive pressure. The transfer room should be kept under a negative pressure. Airflow out of room openings should be a minimum of 100FPM. Recommended filters are 95% NBS.
16. A 40°-50° meat room on the average will require 30 to 35 air changes per hour. We recommend 95% filtration to remove all bacteria from the air to prevent product contamination. In most rooms we recommend a clean-up cycle which will bring in outside air to purge the room of moisture and dry it faster. USDA guidelines say a room cannot be used until it is dry and free of moisture.
17. General packaging would include such areas as pizza, eggs, fish, candy, and many other products. Specific filtration requirements would be dependent on the product.
18. Example of general food processing areas would include pizza, processed meats, and candy. Filtration requirements would be dependent on the product.
19. A bakery is generally not air conditioned. Air change should be between 20 and 25 times per hour. Outside air is used for cooling and ventilation. Bakeries generally require only 25 percent NBS filters.

20. A milk case room usually has a moisture problem. The pressure within the room should be balanced. Provide heated air with a King make-up air unit to balance the exhaust.
21. A powder bagging room requires a unit which supplies 100 percent outside air and no recirculated air. This is necessary due to the powder in the air. In summer, discharge air must be cooled to provide dehumidification and reheated so the air is dry. Due to a potential salmonella problem, 95 percent NBS filters are necessary. Also, since the unit is supplying 100 percent outside air, a matching exhaust unit is necessary.
22. A boiler room requires only ventilation. King make-up air units are generally used with matching exhausts. Remember, a boiler requires approximately 7.5 CFM/H.P. of make-up air, so reduce your exhaust capacity accordingly. Generally, 20 to 25 air changes per hour is adequate. Filters with a 25 percent NBS rating are used to filter out the dust and dirt.
23. An engine room requires only ventilation using a King make-up air unit. Matching exhaust is required. Air circulation of 20 to 25 times per hour is generally sufficient. Filters of 25 percent NBS rating are used to filter out the dust and dirt.
24. A warehouse requires only ventilation and heating for winter use. An air change of 10 to 15 air changes per hour is sufficient using a King make-up air unit. Filters with a 25 percent rating is adequate in most applications.
25. A poultry eviscerating area is generally a high moisture area. Therefore, air conditioning is suggested, and the use of 95 percent filters is recommended due to the exposed product.
26. Poultry process and further process areas are generally maintained at approximately 60-to-70-degree room temperature. Due to the exposed product, a 95 percent filter is recommended. A standard clean-up cycle is also suggested.
27. In a poultry packaging area, the room must be always maintained under a positive pressure. Due to the large amount of exposed product, a 95 percent filter is recommended. Room temperature control is also normally a requirement.
28. A milk process area is generally a very high load, high moisture content area. Pressurization and 95 percent filtration is generally recommended. Depending upon the customer and application, cooling may or may not be required.
29. An ice cream process area requires room pressurization, 95 percent filter filtration, and room cooling.
30. Typical airflow requirements in air changes (ACH) in poultry plants as follows:
    - Eviscerating: 20 to 30
    - Picking (Pinning): 24 to 40
    - Tub Wash: 30 to 60
    - Oven Rooms: 60 to 120

XI. Particulate Contaminants

Air contaminants are generally classified as either particles or gases. Particles dispersed in air are also known as **aerosols**. In common usage, the terms *aerosol*, *airborne particle*, and *particulate air contaminant* are interchangeable. The distinction between particles and gases is important when determining removal strategies and equipment. Although the motion of particles is described using the same equations used to describe gas movement, even the smallest particles are much larger and heavier than individual gas molecules, and have a much lower diffusion rate. Conversely, particles are typically present in much fewer numbers than even trace levels of contaminant gases.

The **particulate** class covers a vast range of particle sizes, from dust large enough to be visible to the eye to submicroscopic particles that elude most filters. Particles may be liquid, solid, or have a solid

core surrounded by liquid. The following traditional particulate contaminant classifications arise in various situations, and overlap. They are all still in common use.

- **Dusts**, **fumes**, and **smokes** are mostly solid particulate matter, although smoke often contains liquid particles.
- **Mists**, **fogs**, and **smogs** are mostly suspended liquid particles smaller than those in dusts, fumes, and smokes.
- **Bioaerosols** include primarily intact and fragmentary viruses, bacteria, fungal spores, and plant and animal allergens; their primary effect is related to their biological origin. Common indoor particulate allergens (dust mite allergen, cat dander, house dust, etc.) and endotoxins are included in the bioaerosol class.

Particulate contaminants may be defined by their size, such as **coarse**, **fine**, or **ultrafine**; **visible** or **invisible**; or **macroscopic**, **microscopic**, or **submicroscopic**. Particles may also be described using terms that relate to their interaction with the human respiratory system, such as **inhalable** and **respirable.**

The size of particulates will generally determine whether or not they will settle and be deposited on surfaces or remain airborne. Table 4 below [2] shows approximate particles sizes and time to settle 1 m.

TABLE 4: APPROX. PARTICLE SIZES AND TIME TO SETTLE 1 M

Type of Particle	Diameter, µm	Settling Time
Human hair	100 to 150	3 to 1 s
Skin flakes	20 to 40	20 to 80 s
Observable dust in air	>10	<5.5 min
Common pollens	15 to 25	1 to 2 min
Mite allergens	10 to 20	1 to 6 min
Common spores	2 to 10	6 to 128 min
Bacteria	1 to 5	21 to 475 min
Cat dander	1 to 5	21 to 475 min
Tobacco smoke	0.1 to 1	8 h to 13 days
Metal and organic fumes	<0.1 to 1	8 h to >13 days
Cell debris	0.01 to 1	8 h to 171 days
Viruses	<0.1	>13 days

XII. Supply Air Flowrate

The required air flowrate supplied to the room will be determined in standard cubic feet per minute, or “SCFM”, and is a calculated value based on the airflow design criteria for the processing room. Standard Air traditionally used in HVAC calculations is defined as having density of 0.075 lbm/ft3 (density of air at 70°F and sea level). The required supply air flowrate in a cleanroom will be based on the greater of the following:

1. The flowrate required to maintain an acceptable contaminant concentration, or particle count, in the room by dilution, or
2. The flowrate required to maintain the specified (or acceptable) temperature, humidity, and allowable air temperature rise through the room based on the room cooling load.

Note that the supply air flowrate must also be adequate to pressurize the room and account for any process exhaust.

Particle Contaminants, Supply Air Flowrate, and ISO Standard 14644

The amount of air supplied to a cleanroom or clean area by the hygienic air handler must be adequate to provide the contaminant dilution needed to reduce room particle concentrations to acceptable levels. The air supplied must also satisfy the heating and cooling loads in the room as well as room pressurization, which is a separate calculation to be covered later. It is important to realize that particle contaminants found in cleanroom air do not come from the filtered supply air, but from activities inside the cleanroom or clean area.

There is an infinite potential list of contamination sources, however, all contamination can generally be broken down into five basic categories, namely the people, facilities, equipment, product, and fluids. The workers present in a cleanroom are typically the major source of particle contamination, even without any motion. Contamination can arise from hair loss, clothing, or other debris e.g., fibers, cosmetics and perfumes, loss of skin tissue or transmission of oil from skin surfaces, etc. Table 5 below shows particle generation from various human activities [7].

TABLE 5: RATE OF PARTICLE GENERATION FROM OCCUPANTS

Activity	Particles/sec (0.3 µm and larger)
Motionless (standing or seated)	1,667
Walking about 2 mph	83,333
Walking about 3.5 mph	116,667
Walking about 5 mph	166,667
Horseplay	1,666,667

Cleanroom cleanliness can be classified by particle concentration according to the widely accepted international standard ISO 14644. Currently, the standard consists of 14 parts, including:

Part 1: Classification of Air Cleanliness
Part 2: Specifications for Testing and Monitoring
Part 3: Test Methods
Part 4: Design, Construction, and Startup
Part 5: Operations
Part 6: Vocabulary
Part 7: Separative Devices
Part 8: Classification of Airborne Molecular Contamination

Part 1 of the standard classifies cleanroom air cleanliness by particle concentration ranging from Class 1 to Class 9, Class 1 rooms having the lowest allowable concentrations (the cleanest), and Class 9 having the highest allowable concentrations by particle size. Airborne particle contaminants of concern in food processing cleanrooms include contaminants generated from human activity, pathogenic bioaerosols, molds, fungi, etc., which are generally in the size range 0.5 to 5 µm. Particles in this size range put most food processing cleanrooms and spaces into ISO Class 6 to 9. Allowable concentrations by particle size for these classes, as measured by use of light scattering (discrete) airborne particle counters (LSAPC), are shown in Table 6 below.

TABLE 6: ISO 14644 CLASSES OF AIR CLEANLINESS BY PARTICLE CONCENTRATION

ISO Class Number	Max allowable concentrations (particles/m3) for particles equal to and greater than the considered sizes, shown below					
	0.1 µm	0.2 µm	0.3 µm	0.5 µm	1 µm	5 µm

6	1,000,000	237,000	102,000	35,200	8,320	293
7	-	-	-	352,000	83,200	2,930
8	-	-	-	3,520,000	832,000	29,300
9	-	-	-	35,200,000	8,320,000	293,000

Part 4 of the standard (Design, Construction, and Startup) provides a method for calculating the supply air flowrate in non-unidirectional airflow (non-UDAF) rooms required to dilute particle contaminants to the concentrations defined by the ISO Class. Cleanrooms with extremely low particle counts designed to meet ISO Class 1-5 typically must have very specialized unidirectional (laminar ceiling to floor) airflow systems. As mentioned above, food processing clean rooms typically fall into ISO Class 6-9 which can be achieved using non-unidirectional airflow, where air is introduced to the room through supply diffusers located on one end of the room and then moves across the length of the room to return registers such that the air supply mixes with room air and dilutes the airborne contaminants. The required steady state supply air flowrate to maintain a specified particle concentration is determined by the rate that particles are emitted in the cleanroom (called the “source strength”) divided the particle concentration minus the deposition rate of larger size particles which will settle and be deposited on room surfaces. The required supply air flowrate is given by Equation [10] below.

$$\dot{V}_{supply} = \frac{S}{\varepsilon \cdot C} - v_D \cdot A \qquad [10]$$

Where

$\dot{V}_{supply} = Supply\ Air\ Flowrate\ to\ the\ Room, m3/s$
$S = Rate\ of\ Particle\ Emission\ in\ Cleanroom\ Air\ (source\ strength), number\ of\ particles/s$
$\varepsilon = Ventilation\ Effectiveness\ (dimensionless)$
$C = Particle\ Concentration\ Limit\ in\ the\ Cleanroom, number\ of\ particles/m3$
$v_D = Deposition\ velocity\ of\ MCPs\ and\ particles\ passing\ through\ air\ and\ onto\ a\ surface, m/s$
$A = Surface\ area\ of\ a\ cleanroom\ where\ deposition\ occurs - usually\ equiv\ to\ the\ floor\ area, m2$

This formula assumes that the number of particles entering the cleanroom or clean zone from the supply air is negligible and can be left out of the formula. When using multi-staged filters with final high efficiency filters, this is a reasonable assumption.

NOTE: While room air changes can be used to make an initial supply airflow estimate (shown in Table 3), the airborne particle concentration is correctly determined by the supply air flowrate and not the number of air changes, defined as the supply air flowrate divided by room volume. In ISO 14664 Part 4 the number of air changes is not used in calculating the supply air flowrate as the number of air changes depends on the volume of the cleanroom. The use of the number of air changes can lead to higher airborne concentrations than expected in small rooms. It can also lead to lower airborne concentrations than necessary in larger rooms, with associated high capital and energy costs. However, if required, the number of air changes can be calculated after the air supply rate has been determined.

The source strength is the rate at which airborne particles are emitted in the cleanroom from people, machinery, movement of equipment and other sources. The maximum source strength (of emitted particles) should be determined as part of the design process. In an operational cleanroom, these can have local variations, and this should be considered when determining the total source strength of the cleanroom.

Source Strength

The particle generation from sources of contamination is expressed as the number of particles dispersed per second. The cumulative number of particles larger or equal to a specific size between 0.1 µm and 5 µm and/or macroparticles larger than 5 µm is considered when determining source strength.

In a cleanroom, the contribution of all sources of a given particle size should be added to determine the source strength *S*, as shown in Equation [11]:

$$S = \Sigma S_i \qquad [11]$$

Where

$S_i = Strength\ of\ Each\ Source\ at\ a\ Specific\ Particle\ Size, number\ of\ particles/s$

NOTE: The source can be at one position (machinery) or can move around (people). In large cleanrooms where a source will not affect the complete cleanroom, the cleanroom can be divided into sections. The calculations would then be applied to each section. Sources do not emit particles at a constant rate, but the rate can vary in time depending on the operations in the cleanroom. When setting requirements, an allowance for excursions in excess of the assumed source strength can be considered.

In practice, the source strength can be difficult to determine if the cleanroom is a new design. In many applications, the contribution from machinery is small compared with that from people. Therefore, the maximum number of people to be found in a cleanroom and the garments they wear are important in determining the source strength. Table 6 below shows measured source strength for people [13] for particles of different sizes as well as microbe-carrying particles (MCPs). The data shown is the average source strength of 55 different personnel when exercising in a dispersion chamber. The beneficial effect of cleanroom garments made from a woven, reusable, polyester fabric with a pore diameter of 28 µm is shown in comparison to normal indoor clothing. It was also reported that gowns (smocks) do little to reduce the source strength, as much of the person's contamination is dispersed from under the gown and into the cleanroom air and, therefore, their source strength can be assumed to be similar to normal indoor clothing. For purposes of estimating source strength for typical food processing cleanrooms, the values for normal indoor clothing in Table 7 are recommended.

TABLE 7: AVERAGE SOURCE STRENGTH FOR PEOPLE EXERCISING IN A TEST CHAMBER

Type of Garments	Ave Source Strength per Person (particles/sec)		
	Particle Size		MCPs
	≥ 0.5 µm	≥ 5 µm	
Normal indoor clothing	35,500	5,500	40
Cleanroom coveralls, hood and full-length boots	17,000	600	3

As mentioned above, the settling and deposition of larger particles onto cleanroom surfaces will reduce the airborne concentration. The surface deposition of particles of ≥0.3 µm and ≥0.5 µm is low and need not be included. However, microbes are carried on skin and clothing particles, and have an average aerodynamic equivalent diameter of about 12 µm, and so are readily deposited onto cleanroom surfaces by gravity. Therefore, if MCPs and large particles ≥5 µm are considered, the reduction by surface deposition should be included, which will require an estimate of the deposition velocity, v_D, used in Equation 10.

Source strength for equipment should be provided by the supplier, or it can be determined by using measurement methods described in ISO 14644 Part 14. Table 8 shows examples of source strength for various types of machines found in pharmaceutical production rooms as an example [13].

TABLE 8: SOURCE STRENGTH FOR VARIOUS MACHINES

Type of machine or equipment	Reference	Source Strength, particles/sec ≥ 0.5 µm
Vial filling machine A	Hejab [8]	33,000
Vial filling machine B	Hejab [8]	500
Blow-fill-seal (BFS) machines	Sundstrom et al [9]	100 – 10,000,000 depending on machine
6-axis robot unmodified	Hnatek [10]	4,000
6-axis robot modified to reduce emissions	Hnatek [10]	0.3

As can be seen in Table 7, source strength can vary dramatically for different types of equipment, and even between different manufacturers of the same type of equipment.

Deposition Velocity and Surface Deposition of Larger Particles and MCPs

The air supply rate required for a non-UDAF cleanroom can be calculated using Equation 10, which incorporates the effect of surface deposition using deposition velocity and the deposition area (usually taken as the room floor area). In the equation, the number of larger particles (≥5 µm) and MCPs which will settle in the room before they are taken up in the return air to the air handler is subtracted from the total supply air flowrate. The Particle Deposition Rate (PDR) is calculated as the airborne particle times the deposition velocity for a given particle size range.

$$PDR = C_D \cdot v_D \qquad [12]$$

Where
$PDR = Particle\ deposition\ rate, No. particles/m2\ /s$
$C_D = Airborne\ particle\ concentration, No. particles\ of\ given\ size/m3$
$v_D = Deposition\ velocity\ for\ particles\ of\ a\ given\ size, m/s$

PDR for particles ≥5 µm and MDR (microbial deposition rate) have been measured and correlated as a function of concentration by various researchers [11][12]. Deposition velocity can be calculated for a given concentration, knowing the PDR or MDR based on equation 12. Deposition velocities for a range of cumulative particle sizes have been reported [11], where it is also reported that the deposition velocity may increase as the airborne concentration decreases. Small particles ≥0.3 µm and ≥0.5 µm were shown to have low deposition velocities of 2.8 x 10-5 m/s and 6.4 x 10-5 m/s, respectively, resulting in very low surface deposition over a range of air cleanliness, hence, it is unnecessary to include their deposition in the calculations. It has been shown [11] that over 80% of airborne particles ≥10 µm are deposited on surfaces in a cleanroom by gravitational settling, and that gravitational settling will be a predominant mechanism down to about 5 µm. Again, particles ≥5 µm and MCPs have greater deposition rates, and so their deposition effects should be included when using equation 10.

Tables 9 and 10 below show deposition velocities for particles ≥5 µm and MCPs based on calculations shown in the references mentioned.

TABLE 9: PDR AND DEPOSITION VELOCITIES FOR PARTICLES ≥5 µm

ISO 14644 Class	>=5µm Particle Concentration, No./m3	Particle Deposition Rate, No./m2/s	Deposition Velocity, v_D, m/s
6	293	1.824	0.0062
	500	2.757	0.0055
	1,000	4.711	0.0047
	2,000	8.050	0.0040
7	2,930	10.81	0.0037
	5,000	16.35	0.0033
	10,000	27.93	0.0028
	20,000	47.73	0.0024
8	29,300	64.12	0.0022
	50,000	96.92	0.0019
	100,000	165.6	0.0016
	200,000	283.0	0.0014
9	293,000	380.2	0.0013

TABLE 10: MDR AND DEPOSITION VELOCITIES FOR MCPs

MCP Concentration, No./m3	Microbial Deposition Rate (MDR), No./m2/s	Microbe Carrying Particle (MCP) Deposition Velocity, m/s
1	0.0161	0.0161
10	0.0731	0.0073
20	0.1153	0.0058
30	0.1505	0.0050
40	0.1818	0.0045
50	0.2105	0.0042
100	0.3319	0.0033
200	0.5234	0.0026
300	0.6832	0.0023
500	0.9557	0.0019
1,000	1.5070	0.0015

Ventilation Effectiveness

The effectiveness of the air system in diluting airborne particles and reducing concentration depends on the ventilation effectiveness. This is influenced by the airflow moving between the supply air terminals and return air registers and equipment exhausts, as well as disturbances of the airflow patterns by, for example, machinery, obstructions, or thermal effects.

An experimental study of a non-UDAF cleanroom, and field tests of another 23 non-UDAF cleanrooms, were carried out [14] to determine their air change efficiency (ACE) indexes. The reference explains the method to obtain ACE indexes in cleanrooms, and reports that good air mixing cannot be assumed in non-UDAF cleanrooms. It also contains information on how poor mixing and low ventilation effectiveness can be minimized. The tests showed that when cleanrooms use efficient air supply diffusers and low-level extracts, the supply air will mix effectively with room air, and the ACE index is unlikely to be below 0.7, and may be close to 1; therefore, a ventilation effectiveness value of 0.7 is a reasonable design choice.

Supply Air Flowrate to Ensure Concentration Limit is Rarely Exceeded

The dispersion rates of particles and MCPs from sources will vary in a cleanroom over time and are normally reported as an average number per second and, therefore, the calculated air supply flowrate gives an average airborne concentration (No. particles/m3), where half the counts will be above the average concentration. However, ISO 14644-11 requires the particle concentration not to exceed the class limit, and a cleanroom with an airborne concentration whose limit is exceeded about half the time, would be generally unacceptable. It is almost impossible to design a cleanroom that will never exceed the class limit, as airborne counts conform to a statistical distribution, with counts distributed around the average and a number of outlying high counts. Increasing the calculated supply air flowrate by a factor of 2.7 ensures that 95% of the particle counts fall below the maximum concentration limit for the ISO class selected, and is a reasonable design choice [13][15].

During the air volume flow rate evaluation process, it is recommended that the safety margins are included to address uncertainties associated with source strength data, particle generation and distribution, and ventilation effectiveness. These safety margins should be clearly documented in the design and can then be revisited after the cleanroom becomes operational and the actual airborne concentrations become known.

Supply Air Flowrate Calculation Example

The following is an example of calculated supply air flowrate needed to maintain ISO Class 8 cleanliness with a maximum airborne concentration of 100 MCPs per m3.

Given:
Room Dimensions, WxLxH = 15m x 20m x 6m (49.2’ x 65.6’ x 19.7’)
Type of Room: Poultry Eviscerating
No. workers: 8
Type of garments worn by workers: Normal indoor clothing with cotton smock
Source strength from workers: 8 x 35,500 = 284,000 particles/sec $\geq$ 0.5 µm
Source strength from workers: 8 x 5,500 = 44,000 particles/sec $\geq$ 5 µm
Source strength from workers: 8 x 40 = 320 particles/sec MCPs
Source strength from machinery: 150,000 particles/sec $\geq$ 0.5 µm

NOTE: The total source strength for particles $\geq$ 0.5 µm will be 284,000 + 150,000 = 434,000/sec according to equation [11].

Solution is shown in Table 11 below:

TABLE 11: EXAMPLE CALCULATED SUPPLY AIR FLOWRATE

Airborne Contamination Type	$\geq$ 0.5 µm	$\geq$ 5 µm	MCPs
Maximum concentration particles/m3 – C	3,520,000	29,300	100
Source strength, particles/sec – S	434,000	44,000	320
Ventilation effectiveness - ε	0.7	0.7	0.7
Deposition velocity, m/s - v_D	-	0.0022	0.0033
Floor area, m2 – A	300	300	300
Calculated supply air flowrate (minus surface deposition), m3/s	0.18	1.49	3.58
Adjusted supply air flowrate (x 2.7), m3/s	0.48	4.01	9.67
Adjusted supply air flowrate (x 2.7), scfm	1,017	8,497	20,490
Room volume, m3	1800	1800	1800
Air change rate, changes/h	0.96	8.0	19.3

In the example above, the supply air flowrate required to maintain a maximum MCP concentration of 100 particles/m3 is significantly higher than the supply air flowrates required for either of the other two particle sizes, and in this case is the appropriate air flowrate to use for design of the air handler.

NOTE: When evaluating the supply air flowrate, any specified recovery rate or recovery time that has been given in the requirements will also have to be considered. If a higher supply air flowrate is required to satisfy these requirements, then the higher value should be used. More information on recovery rate and recovery time calculations can be found elsewhere (ISO 14644-4:2024 Annex B).

Calculated Supply Air Flowrate vs Air Change Rate

There is a precedent of using simplified rule-of-thumb values for air change rates published in supplier literature, or guidance documents from various government or industry sources, or based on past experience. This approach may ignore critical variables that could significantly affect the room particle concentration in terms of air change rate requirements. Such variables include room internal particle size and generation rate, particle surface deposition, particle entry through filtered supply air, particle exit through return and exhaust air, air leakage (particle loss or gain) under pressurization or depressurization, layout of processes, and locations or supply, return, and exhaust registers. Intuitively, for example, activities that generate higher levels of particle concentration would need a higher air change rate to dilute particle concentration than those that generate lower levels of particle concentration, but using a simplified approach may mask or miss such differences.

While a rigorous approach, as shown in the example above, to calculating air changes based on rates of contaminate generation and the resulting particle concentrations in a room is desirable, this process is complex and not all of the required information or estimation methods (i.e., CFD) may be available to the designer. Table 3 (with notes) in the preceding section, therefore, gives an initial estimate based on standard practice values for air changes in various types of processing cleanrooms.

Room Air Flowrate Based on Heat Load and Temperature Rise

Room airflow should also be calculated based on an allowable temperature rise by rearranging the basic equations listed above and knowing the room load and SHR, as follows:

$$Required\ Room\ Airflow, SCFM = \frac{\dot{q}_{sens}}{60 \cdot \rho_{sa} \cdot C_{p\ air} \cdot \Delta T_{air}} \qquad [13]$$

The same example shown in Table 11 is used to illustrate.

Given:
Room Dimensions, WxLxH = 15m x 20m x 6m (49.2’ x 65.6’ x 19.7’)
Type of Room: Poultry Eviscerating
Room Load (Table 3): 100 sq ft/TR
Assumed SHR = 0.67
Allowable Room Air Temperature Rise = 10°F
Assumed Air Specific Heat = 0.243 Btu/lbm°F

Calculated Room Air Flowrate Based on Allowable Room Air Temperature Change:

$$Room\ Sensible\ Load, \dot{q}_{sens} = \frac{49.2ft \cdot 65.6ft}{100sq\ ft/TR} \cdot 12{,}000\ Btu/h/TR \cdot 0.67 = 259{,}493\ Btu/h$$

$$Required\ SCFM = \frac{\dot{q}_{sens}}{60 \cdot \rho_{sa} \cdot C_{p\ air} \cdot \Delta T_{air}} = \frac{259{,}493}{60 \cdot 0.075 \cdot 0.243 \cdot 10} = 23{,}730\ SCFM$$

In this example, the calculated room air flowrate is based on the allowable room air temperature rise of 10°F and equals 23,730 SCFM. Bear in mind that this air flowrate is the average room air flowrate, not the supply air flowrate introduced to the room by the air handler. The final room supply air flowrate for the equipment design must be based on the average room air flowrate plus the airflow required for room pressurization plus any process exhaust from equipment in the room. Room pressurization is covered in more detail below.

Room Pressurization

Controlling air pressures in a cleanroom is an important part of effective contamination control, providing resistance to infiltration of external sources of contaminants. In non-pressurized spaces, or spaces with air pressures lower than that of the surrounding environment, nearby particulate contaminants enter the cleanroom by infiltration through doors, cracks, pass-throughs, and other penetrations for pipes, ducts, etc. Pressurization resists this infiltration of unfiltered external sources of contaminants. A cleanroom with the most stringent cleanliness requirements should have the highest air pressure relative to its adjacent rooms, with decreasing room pressures corresponding to decreasing cleanliness levels. In a facility with multiple cleanrooms operating with varying pressures, best control of room pressurization will be achieved with individual air handlers serving each space.

Positive pressurization is achieved by designing the air handling system to deliver a supply air flowrate which is greater than the sum of return and exhaust air flowrates, plus air leakage through openings and cracks around door frames, etc. ASHRAE (HVAC Applications Handbook) states that a static (door closed) pressure differential between rooms, regardless of cleanliness class, of 0.04 in of water will minimize particle migration.

ISO 14644-4:2020 Annex B.2.2.2 states that “Depending on the criticality of the segregation, pressure differentials between rooms should typically range from 7.5 Pa to 15 Pa (0.03 to 0.06 in of water). However, in multiple connected rooms with different cleanliness requirements, it might be necessary to design for smaller pressure steps to avoid excessive pressures in the highest-pressure room in the cascade. An increase in pressure differentials creates an increase in air velocity through any gaps. Pressure differentials in the range 7.5 to 15 Pa (0.03 to 0.06 in of water) results in velocity through gaps in the order of 3.5 to 5.0 m/s (690 to 980 fpm). Precautions should be taken to ensure accurate measurement of separating airflow or pressure and to investigate the stability of the installation. Excessive pressure differences can be problematic because they can stress the structure of enclosures and make door opening difficult, for example. In situations where the airflow volume through leakage paths is low due to high pressure integrity of the barrier (a more airtight enclosure), maintaining pressure stability can be more difficult, because in this instance small changes in the volume of supply and return or exhaust air can lead to large changes in the pressure differentials, unless there is precise control of airflow volumes.”

The correct amount of supply air introduced by the air handler to the room to maintain the pressurization of 0.04 in of water needed to prevent contamination of the room by infiltration will be equal to the return (room) air flowrate plus the exhaust air flowrate plus the area of all openings and cracks times an air leakage velocity of approx. 800 fpm.

$$\dot{V}_{supply} = \dot{V}_{return} + \Sigma\dot{V}_{exhaust} + \Sigma A_{open} \cdot v_{leak} \qquad [14]$$

Room pressurization is provided by the outdoor air introduced to the room by the hygienic air handler. The amount of outdoor air can be expressed as a percentage of the total supply airflow, or as scfm.

Generally speaking, the only way to maintain a differential pressure 0.04 inches of water in the processing room is to design the room with closed doorways. In other words, it is not practical or possible, to design a pressurized room with doorways which are continuously kept open. Depending on the cleanliness of the adjacent hallways or rooms, an airlock or vestibule may be required to maintain room pressure and required cleanliness levels.

As mentioned above, leakage air flowrate to maintain adequate room pressure can be estimated as the area of all room openings, including the cracks around door frames as well as door undercuts, times 800 fpm air velocity. An alternate method is to estimate the leakage rate of 5% for "tight" rooms with minimal doorway activity, ranging up to 15% for rooms with high doorway activity. Note that any process exhaust will add to this outdoor air percentage. The following example illustrates the calculated air flowrate required to maintain 800 fpm velocity through room openings in terms of scfm and % of supply air.

Given:
Room Dimensions, WxLxH = 15m x 20m x 6m (49.2' x 65.6' x 19.7')
Type of Room: Poultry Eviscerating
Number and Size of Rapid Rollup Doorways (with airlock), (Qty) x W x H = (1) x 8' x 8'
Number and Size of Man Doors, (Qty) x W x H = (2) x 3' x 7'
Assumed Doorframe Crack Width = 1/8"
Assumed Door Undercut = ½"
Process Exhaust Air Flowrate = 1,000 scfm

Calculated Supply Airflow Based on Air Openings and 800 fpm.

$$Room\ Volume = 49.2 \cdot 65.6 \cdot 19.7 = 65{,}582\ ft3$$
$$Calculated\ Room\ Airflow\ based\ on\ 10°\text{F}\ temperature\ rise = 23{,}730\ scfm\ (see\ example\ above)$$
$$Rollup\ Doorway\ Opening\ Area = \big((8+8+8)\cdot 12 \cdot 1/8 + 8 \cdot 12 \cdot 1/2\big) = 84\ in2 = 0.58\ ft2$$
$$Man\ Door\ Opening\ Area = 2 \cdot \big((7+3+7)\cdot 12 \cdot 1/8 + 3 \cdot 12 \cdot 1/2\big) = 87\ in2 = 0.60\ ft2$$
$$Total\ Opening\ Area = 0.58 + 0.60 = 1.18\ ft2$$
$$Pressurization\ Air\ Flowrate = 1.18 \cdot 800 = 944\ scfm$$
$$AHU\ Supply\ Air\ Flowrate = 23{,}730 + 1{,}000 + 944 = 25{,}674\ scfm$$
$$Outdoor\ Air\ Flowrate\ as\ \%\ of\ Supply\ Air = (1{,}000 + 944)/25{,}674 \cdot 100 = 7.6\%$$
$$Number\ of\ Air\ Changes\ per\ hour = 25{,}674 \cdot 60/65{,}582 = 23.5$$

Example Summary:
- Supply air flowrate required to maintain ISO Class 8 cleanliness and MCP concentration of 100/m3 = 20,490 scfm (Table 11)
- Supply air flowrate required to maintain room air temperature rise of 10°F at a room load of 100 sq ft/TR and SHR=0.67, with 1,000 scfm exhaust and room pressurization of 0.04 in water = 25,674 scfm
- Conclusion: Size air handling unit for 25,674 scfm supply air flowrate.

Mixed vs Unmixed Room Airflow and Average Room Temperature

The temperature of the air supplied to the room will rise in proportion to the sensible heat load and the room air flowrate according to equations [5] and [13] above. Determining the average room temperature will depend on how much mixing of the supply air is assumed to take place in the room. If it is assumed that the air is fully mixed, this implies that the supply air immediately absorbs all of the room heat load such that the return air temperature will be equal to the average room temperature. If it is assumed that the air is unmixed, this implies that the air temperature will rise at a constant rate as it moves through the

room from the supply to the return air openings. The average room temperature in the unmixed case then, will be equal to the average of the supply and return air temperatures based on the calculated air temperature rise. For a given room heat load, the fully mixed airflow assumption will result in a lower required supply air temperature than the unmixed airflow assumption. In terms of equipment sizing, the fully mixed airflow assumption (return air temperature setpoint) will result in lower evaporator suction temperatures and large cooling coils compared to the unmixed airflow assumption. *In other words, an assumption of fully mixed airflow will result in the more conservative equipment selection and calculated refrigeration load.*

Colmac Hygenair software allows equipment to be selected based on either fully mixed or unmixed airflow assumptions. This is accomplished by allowing the user to size the air handler based on a "return air" temperature setpoint or based on a "room average" temperature setpoint. There are help screens in the software to assist the user in selecting which selection method is appropriate for their application.

SCFM vs ACFM

While SCFM is convenient for calculating rates of heat transfer, moisture removal, etc., actual airflow, or "ACFM" must be used when selecting fans since fans are volumetric devices. Actual airflow can be easily calculated knowing SCFM by using the following equation:

$$ACFM = SCFM \cdot \frac{\rho_{sa}}{\rho_{actual}} = SCFM \cdot \frac{0.075}{\rho_{actual}} \quad [15]$$

Where:
$ACFM = Actual\ Volumetric\ Flowrate\ of\ Air, cubic\ feet\ per\ minute$
$SCFM = Standard\ Volumetric\ Flowrate\ of\ Air, cubic\ feet\ per\ minute$
$\rho_{sa} = Density\ of\ Standard\ Air\ (Air\ at\ 70°F\ and\ Sea\ Level) = 0.075\ lbm\ dry\ air/ft3$
$\rho_{actual} = Density\ of\ Air\ at\ Actual\ Dry\ Bulb\ Temp, Humidity, and\ Elevation, lbm\ dry\ air/ft3$

External Static Pressure

Colmac HygenAir™ software accurately calculates all of the air pressure drops within the air handler unit to correctly size the supply fan and motor. The air pressure drop through supply and return air ductwork, however, will be different for each project and will depend on the type and size of the ductwork and on the type of supply and return diffusers used. This external pressure drop must be calculated and entered in the HygenAir™ software to correctly size the unit supply fan and motor.

Effect of Elevation on Airflow

As mentioned above, fans are selected, and fan power is calculated based on actual airflow (ACFM) rather than standard airflow (SCFM). The density of air decreases as elevation increases, which results in reduced air mass flow rate for a given fan size and speed. This results in a corresponding reduction in heat transfer capacity according to equations [5] and [6]. In other words, as air density goes down, the ACFM must increase to maintain a required mass flowrate of air. It is important, therefore, to know the elevation of the jobsite and to enter that elevation into the HygenAir™ software.

XIII. Bypass Air Defrost

Certain types of food processing cleanrooms may be operated at temperatures as low as 38°F to manage pathogens (reduce risk of contamination) and improve food safety. In order to reach air temperatures this low, the cooling coils in air handling equipment will have to be operated with evaporating (or brine) temperatures well below freezing. In order to operate continuously at these low temperatures, cooling

coils will need to be defrosted periodically to melt accumulated frost, collect it in drainpans, and drain it away via sanitary drain piping.

The lowest evaporating temperature at which cooling coils can be operated without frosting (and requiring a defrost cycle) is 30°F. This translates to a minimum room temperature without defrosting of between 45 and 50°F. Below these temperatures, the cooling coils must be actively defrosted. Colmac Coil has developed a sequential coil defrosting system which allows continuous cooling operation during the process mode at low room temperatures. Three cooling coils are required for sequential defrosting, piped as shown in Figures 1, 2, and 3. The sequential defrosting controls within the Colmac HygenAir™ control panel keeps two of the coils active at the design evaporating temperature by fully opening the motorized back pressure regulating valves mounted at the outlet of each coil. The motorized back pressure regulating valve on the third coil is modulated closed to maintain an evaporating temperature during defrost of 38°F. The defrost control automatically regulates the motorized back pressure regulating valves to keep two of the three coils at design evaporating temperature while sequentially defrosting the third. This approach continually provides the room with the required cooling and airflow without the need for additional hot gas piping and controls to defrost coils.

XIV. Controlling Humidity

Room air humidity has been tied to the viability of various pathogens and biological contaminants at room temperatures [16]. Humidity has various complex, but nonetheless decisive, effects on both the susceptibility of humans to respiratory infection but also on the rate of growth of pathogenic and other undesirable organisms. Very low relative humidity causes dryness of the skin and mucous membranes, which may lead to chapping and irritation of the throat and other sensitive areas leading to infection. Low relative humidity also can result in excessive static electricity which can lead to processing problems with powdered food products or ingredients. High relative humidity can lead to the accelerated growth of various pathogens, molds, and fungi. Controlling humidity within the cleanroom can minimize the growth of pathogens and promote worker comfort and health. High humidity in cleanrooms can also result in condensation on cold surfaces where pathogens and biological contaminants can grow directly or can contaminate food products from dripping condensate.

The ideal room relative humidity is therefore a critically important criteria to consider and establish when designing air handling equipment and environmental controls in the cleanroom.

Effect of Humidity on Biological Contaminants

The relative humidity of the air as well as surface condensation provides a favorable medium for survival and growth of biological contaminants such as bacteria, viruses, molds, and fungi.

Bacteria

Bacteria almost always contaminate wet surfaces, such as cooling coils and drainpans within air handling equipment or water left standing on floors or other flat surfaces in the cleanroom. If these wet surfaces remain uncleaned or untreated, bacteria may leave the surfaces and enter the airstream in aerosol form. Examples of these types of bacteria include Staphylococcus aureus, Pseudomonas aeruginosa, Enterobacter, and Acinobacter. Legionella is commonly found in cooling coil drainpans which are incorrectly designed such that water does not drain quickly and completely. Upon becoming airborne these bacteria will be affected by the humidity of the airstream. Several species of bacteria including Escherichia coli, Aercbacter serogenes, and Mycoplasma gallisepticum prefer relative humidity below 40%. Other species of bacteria including Serratia marcescens and E. coli prefer relative humidity above 40%. Mycoplasma laidlawii prefers relative humidity at either the high or low end of the scale, above 75%, and below 25%.

The combination of bacteria that prefer high relative humidity, those that prefer low relative humidity, and those that prefer either low or high humidity produces a mid-range of humidity between 30% and 60% in which bacterial populations are minimized.

Viruses

Vaccinia virus (cowpox), Equine encephalitis virus, influenza virus, para influenza virus, and other myxoviruses (including measles) survive better in aerosols of low relative humidity (less than 50%). Polio virus and herpes virus remain viable longer in relative humidity exceeding 50%. Adeno virus, the cause of some acute respiratory infections, prefers relative humidity between 70% and 80%.

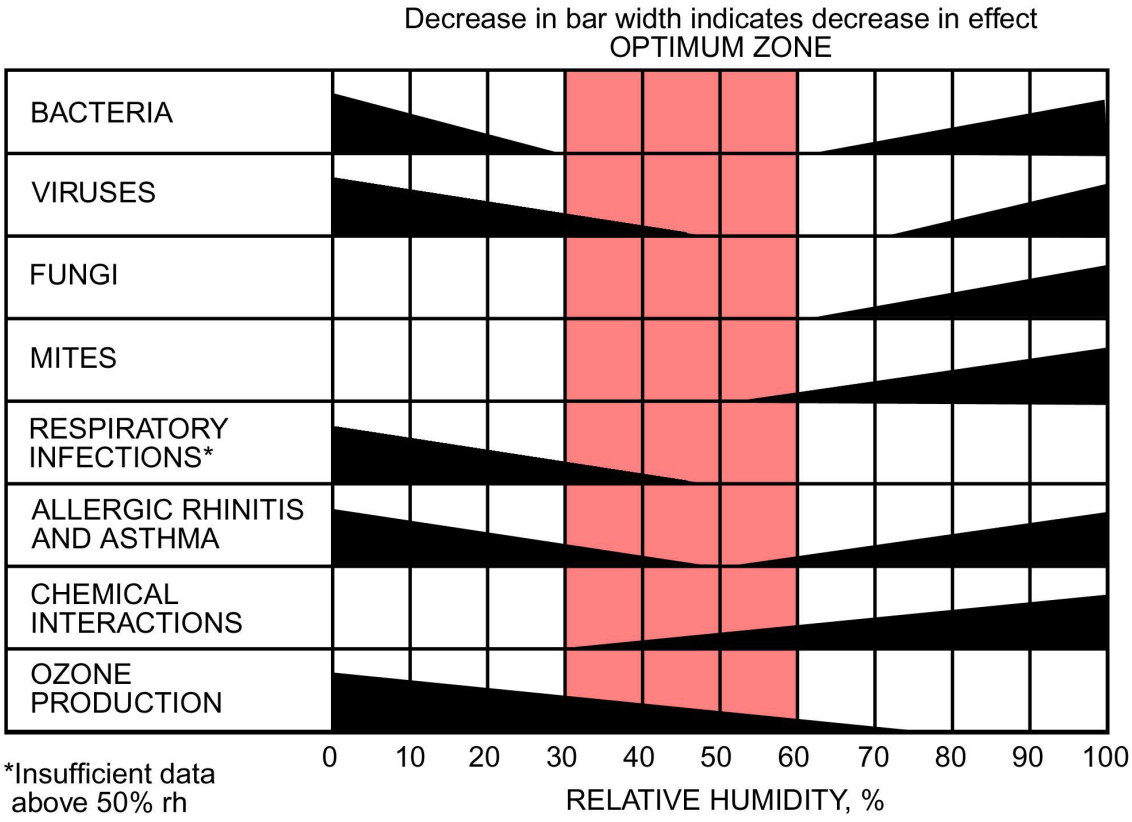
The combination of viruses that prefer high relative humidity and those that prefer low relative humidity produces a mid-range of humidity between 50% and 70% in which the viral population is minimized.

Fungi

Any wet organic material may support the growth of fungi. Damp walls, leather, cotton, paper, fireproofing materials, insulation, and food particles have all been shown to be sources of contamination. While harborage spaces such as these can be managed by good cleaning and sanitation practices, there is a correlation between relative humidity and fungal growth. The maximum growth of fungus occurs above 95% relative humidity and almost ceases below 80%.

Figure 11 below summarizes the research of Stirling [16] and includes other effects of humidity on human health, such as allergies and asthma, chemical interactions, and ozone production.

FIGURE 11: EFFECT OF RELATIVE HUMIDITY ON PATHOGENS AND CONTAMINANTS



Room Humidity and Condensation

Condensation will occur whenever a) humid air come into contact with surfaces which are at a temperature below the dew point temperature of the air, or b) two airstreams mix in such a way that the humidity of the resulting mixed air condition is greater than 100%, in other words, the air is supersaturated with moisture. In this case fog (airborne liquid droplets) will form.

Tables 12 and 13 below show the dew point temperature of room air at various dry bulb temperatures and relative humidities (sea level).

TABLE 12: DEW POINT TEMPERATURE VS RELATIVE HUMIDITY (SI)

	Dew Point Temperature, °C			
Ave Room	Room Relative Humidity, %			
Temp, °C DB	50	60	70	80
3	-5.7	-3.5	-1.7	-0.1
4	-4.9	-2.7	-0.9	0.9
5	-4.0	-1.9	0.0	1.8
6	-3.2	-1.0	1.0	2.8
8	-1.6	0.7	2.9	4.8
10	0.1	2.6	4.8	6.7
15	4.7	7.3	9.6	11.6
20	9.3	12.0	14.4	16.4
25	13.9	16.7	19.2	21.3

TABLE 13: DEW POINT TEMPERATURE VS RELATIVE HUMIDITY (IP)

	Dew Point Temperature, °F			
Ave Room	Room Relative Humidity, %			
Temp, °F DB	50	60	70	80
38	22.3	26.1	29.4	32.4
40	23.9	27.8	31.1	34.3
45	28.0	32.0	35.8	39.2
50	32.1	36.7	40.6	44.1
55	36.7	41.4	45.4	49.0
60	41.3	46.1	50.2	53.8
65	45.9	50.8	55.0	58.7
70	50.5	55.5	59.8	63.6
75	55.1	60.2	64.6	68.4
80	59.7	64.9	69.3	73.3

Any surface in the room which is at a temperature below the dew point temperature shown in the table will condense moisture. Cold surfaces could include pipes carrying refrigerant of cold food products, refrigerated equipment, the food product itself. Insulating pipework with adequate insulation and a robust vapor barrier installed on the outside (the warm side) of the insulation will prevent condensation in most cases. Condensation on processing equipment and cold food products may be unavoidable and can only be handled by installation of drip trays and pans. Depending on the degree of air mixing in the room, condensation may form on the supply air diffuser box. This is due to the supply air always being introduced at a lower temperature than the room air. This possibility of condensed moisture forming on the supply air diffuser box can be mitigated in a couple of ways. Specifying the diffuser box with an insulated full coverage drain pan and an actively heated insulation cover is the most positive way to prevent condensation. Designing the supply air flowrate for a maximum air temperature rise in the room of 8 °F (4.4 °K) and using active reheat to maintain a maximum room relative humidity of 70% will also prevent condensation on the surface of the supply air diffuser box and/or ductwork.

Another recommended practice with regard to the introduction of supply air to the room is to direct the air across ceilings or walls in such a way that the surfaces are “swept” by the drier supply air.

Effect of Humidity on Filtration Media

Relative humidities greater than 80% will increase the airflow resistance through the micro-glass paper media commonly used in final filters. Relative humidity levels above 96% can cause rapid microbial growth fed by the dirt pack within the filter media in final filters. This growth from high humidities can actually grow through HEPA filter media. Synthetic filter media will handle moisture slightly better than micro-glass, but performance is still degraded by 2 to 3 MERV ratings [19].

Room Relative Humidity: Conclusions

From the above discussion, it is apparent that designing air handling equipment to maintain room relative humidity within a range of 50 to 70% produces a number of benefits, including:

- Minimizing the growth and/or viability of pathogens and biological contaminants
- Reducing or eliminating condensation on surfaces
- Maximizing filter performance and avoiding premature filter failures

While it is possible to achieve room relative humidities in the range of 50 to 60% with mechanical refrigeration, most of the time desiccant dehumidification equipment is used to reach these low humidities. Desiccant dehumidification equipment is typically very expensive and energy intensive to operate. Relative humidities in the range of 70 to 80% can almost always be achieved using mechanical refrigeration and reheat. The recommended general design criteria for food processing cleanrooms, therefore, is to control room relative humidity at 70% using the required refrigeration and reheat. Colmac HygenAir™ software makes these complex design calculations simple and accurate knowing room load, climate conditions, and room configuration.

Controlling Humidity During Sanitation and Cleanup Mode

Mixed air HygenAir™ Air Handling units are designed to provide 100% outdoor air during room sanitation and cleanup modes to rapidly and continuously exhaust warm humid air and fog formed from hot wash down water and chemicals. During winter months, heating is typically provided by direct or indirect gas-fired heating of the outdoor air.

Another important function of the air handling unit is to continue operating with 100% outdoor air after washdown operations have ceased until the room relative humidity reaches 80%. Once this relative

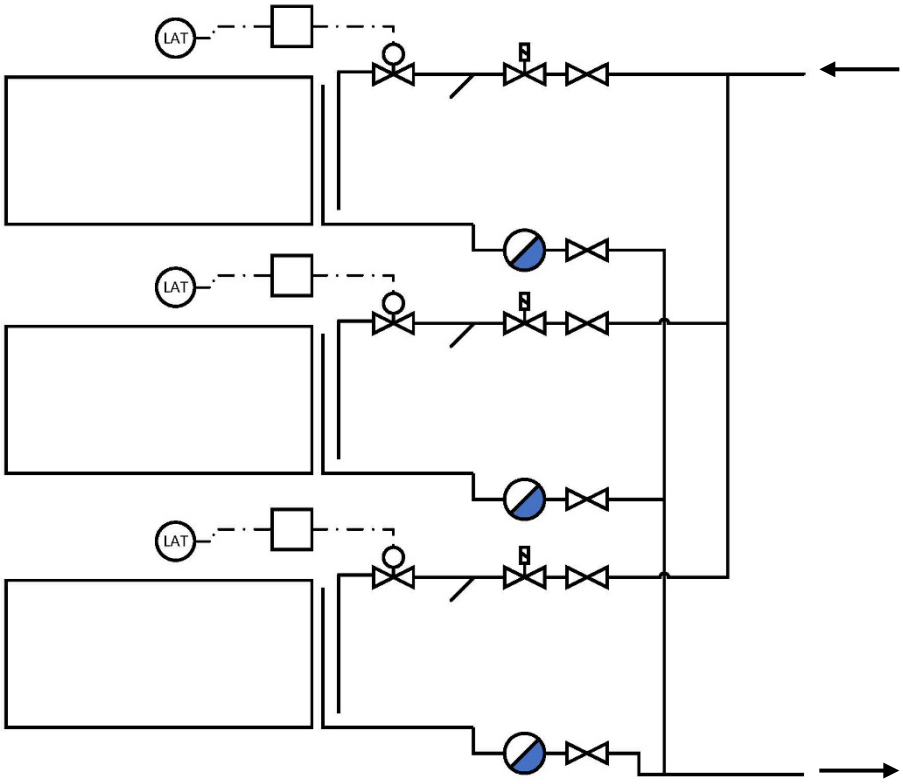
humidity is achieved, outdoor and return air can return to their process mode settings and refrigeration with reheat can restart to continue to dehumidify (“dry out”) the room to the room relative humidity setpoint.

Reheat

Control of relative humidity can effectively be accomplished within the hygienic air handler by using the refrigeration system to first cool and dehumidify the airstream to the design dew point temperature, and then “reheating” to the room dry bulb temperature setpoint. Reheating the airstream can be done using an available heating medium, usually warm glycol or condensing refrigerant gas. In the case of a direct refrigeration system, the air is cooled and dehumidified through an evaporator coil. The airstream is then reheated using the heat from the high side of the system, available in the form of hot gas from the discharge of the compressor, condensed back to a liquid in the reheat coil. By installing a liquid drainer on the outlet of each reheat coil, the leaving air temperature from the coil can effectively be controlled via a motorized control valve on the coil inlet which regulates the flow and condensing pressure (and temperature) of the refrigerant entering the coil.

Figure 12 below shows the condensing reheat coil control valve arrangement described above.

FIGURE 12: REHEAT (CONDENSING) COIL CONTROL VALVE GROUP



Hot Gas Reheat Example:
Given:
Room Temperature Setpoint: 40°F
Room Relative Humidity Setpoint: 70%

Solution:
The refrigeration system and evaporator coils must be designed to deliver a leaving air dry bulb temperature from the evaporator coils that is at least as low as the dew point temperature shown in Table 13 for a 40°F room temperature and 70% relative humidity, that is, 31.1°F. The outlet pressure regulating valves (see Figures 1-3) on each of the evaporator coils will be controlled based on a signal from a room relative humidity sensor to maintain the 70% rh setpoint. The controller will open the pressure regulating valves (lowering the evaporating pressure and temperature) on increasing room relative humidity. The hot gas inlet regulating valves on each reheat coil will then be controlled to maintain the room dry bulb temperature setpoint of 40°F, opening on falling temperature (raising the condensing pressure and temperature).

XV. Operating Modes

HygenAir™ Hygienic Air Handling Units are designed to operate in several modes specific to food processing cleanrooms, including Process, Sanitation and Cleanup, Dry-out, and Economizer modes.

Process Mode

During periods when normal production of food products is taking place, outdoor air is mixed in the correct proportion with recirculating return air from the room by means of opposed blade balancing dampers, to provide pressurization and prevent infiltration of contaminated air from other parts of the facility. This mixed air is first pre-filtered to remove any dust, dirt, or contaminates from the outdoor air, normally with MERV 8 or 9 filters. The pre-filtered air passes through the supply fans and enters the cooling coils where it is cooled and dehumidified. The cooled air is then reheated to bring the supply air to the correct temperature and humidity before it passes through the final filters to reach the required cleanliness with filters having MERV rating ranging from 13 to HEPA filtration. The conditioned and filtered air then passes through insulated, cleanable ductwork into the room.

Cleanup/Sanitation Mode

Food processing cleanrooms are periodically cleaned and sanitized with heated water (as hot at 160°F). This process produces large amounts of humid air and fog in the room which is removed from the space by operating the air handler in this mode. During this mode, outdoor and exhaust air dampers are fully opened, and return air dampers are fully closed. The outdoor air is heated by the gas-fired heater (direct or indirect-fired), as necessary to maintain the room air temperature during the cleanup and sanitation cycle at some setpoint, usually around 80°F. The exhaust fans are energized and speed is regulated based on a room differential pressure sensor signal to maintain the room at the pressurization setpoint.

Dry-out Mode

At the end of the cleanup/sanitation cycle the room needs to be “dried out”, that is, as much free water on floors, walls and ceilings, and equipment needs to be evaporated and exhausted as possible before returning to process mode. This can be accomplished by returning the dampers to the normal process mode positions to achieve the normal room pressurization setpoint. The room temperature set point, however, is set to a relatively high temperature (75 to 80°F) and the lowest humidity possible. This will drive all of the evaporator coil outlet pressure regulating valves fully open to achieve the lowest evaporating temperature (maximum dehumidifying effect) possible, while the reheat coil supply pressure regulating valves will drive open to maintain the dry-out mode room temperature setpoint. At the end of the dry-out mode, the room temperature and humidity setpoints are returned to the process mode values.

Economizing Mode

During periods of low outdoor ambient temperatures, this mode allows more than the normal process mode amount of outdoor air to enter the air handler. This reduces the load on the refrigeration system and when properly applied and controlled, can save a significant amount of energy. Note that economizing only makes sense in northern climates during winter months. The economizer setpoint will, of course, depend on the design room temperature – the colder the room temperature the lower the economizer setpoint. Therefore, economizing is not practical or economical in southern states with warmer winter temperatures.

During economizer mode, the outdoor dampers modulate open, and the return air dampers modulate closed by offsetting percentages. The exhaust fans are energized, and exhaust air dampers modulate open as exhaust fan speed increases to maintain the room pressurization setpoint. Economizer control, needless to say, is complex and is offered as an optional feature on HygenAir™ Air Handlers.

XVI. Part Load Operation

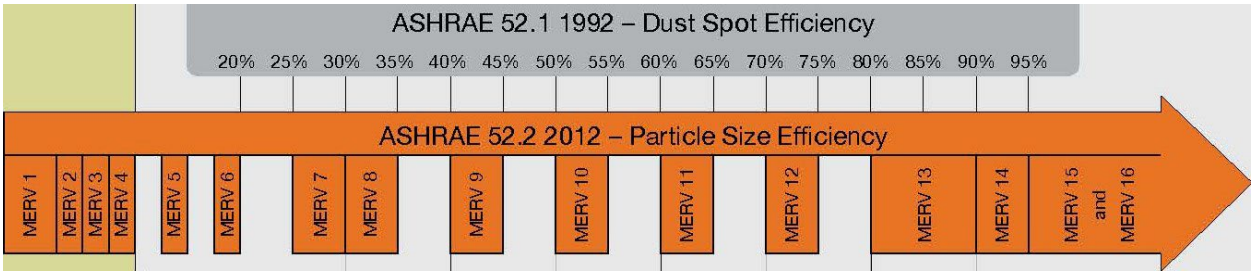
Normally, cleanroom air handling equipment is selected based on summertime full load operation. It is important, however, to anticipate how the air handler and refrigeration system will respond to off-design and part-load conditions. Colmac HygenAir™ selection software allows the user to make rating performance calculations for these off-design conditions. This is a recommended step in the design process.

XVII. Filtration

As mentioned above, one of the critical functions of the hygienic air handler is to deliver conditioned air to the cleanroom which is essentially free of particulate contaminants. Filtration is accomplished using a pre-filter section to capture larger particles in the outdoor and return air, an optional intermediate filter section downstream of the pre-filters to capture smaller particles. This intermediate filter section works to provide cleaner air to the supply fan and cooling coils, and also extend the life of the final filters. The final filter section downstream of the cooling and reheat coils then acts to remove the smallest particles from the supply airstream prior to its introduction to the cleanroom.

Filters sold in North America are tested and rated according to ASHRAE Standard 52.2-2017. The method of testing defined by the standard measures the performance of filters in removing particles of specific diameters as the filter becomes loaded by standardized loading dust fed at intervals to simulate accumulation of particles during service life. The standard also defines a method for counting airborne particles of 0.30 to 10µm in diameter upstream and downstream of the filter in order to calculate removal efficiency by particle size. The data measured by testing is then used by the filter manufacturers to establish and report performance as a minimum efficiency reporting value (MERV) rating. ASHRAE 52.2 replaces the previous ASHRAE Standard 52.1-1992, which was based on measuring dust-removal performance, first by the percentage of the weight of a defined synthetic dust captured by the filter (weight arrestance), and second by comparing the blackening of targets both upstream and downstream of the filter using ambient atmospheric dust (dust-spot efficiency). More detailed information on Standard 52.2 and MERV ratings can be found in reference [20]. Figure 13 below gives an approximate comparison between ASHRAE 52.2 and 52.1 filter ratings.

FIGURE 13: ASHRAE STANDARD 52.2 AND 52.1 RATING COMPARISON



Filters found in Colmac HygenAir™ selection software are shown based on MERV ratings.

Filter MERV ratings must be defined by the food processor based on the type of product being produced in the cleanroom, outdoor environmental conditions, particulate size and source strength, etc., and on their own hazard analysis and food safety plan. As a general reference, however, Table 14 below shows various types of contaminants by particle size along with a suggested corresponding MERV filter rating [21].

TABLE 14: MERV RATING VS CONTAMINATE PARTICLE SIZE

MERV Rating	Min Particle Size	Type of Contaminant
17-20	<0.3 µm	Viruses, carbon dust, sea salt, smoke
13-16	0.3 – 1.0 µm	Bacteria, droplet nuclei (human sneeze), cooking oil, most smoke and insecticide dust, most face powder, most paint pigments
9-12	1.0 – 3.0 µm	Legionella, humidifier dust, lead dust, milled flour auto emission particulates, nebulizer droplets
5-8	3.0 – 10.0 µm	Mold, spores, dust mite debris, cat and dog dander, hair spray, fabric protector, dusting aids, pudding mix
1-4	>10.0 µm	Human hair, pollen, dust mites, cockroach debris, sanding dust, spray paint dust, textile fibers

Inadequately filtered supply air associated with air handlers can be one of several sources of pathogens and spoilage organisms in the plant environment [22]. Other sources of contamination of the processing environment exist and may include soiled work clothes, street shows, roof leaks, inadequately trapped floor drains, drain backups and negative air pressure drawing in air from contaminated outside areas. However, with regard to airborne particles, it is believed that liquid or solid particles of principal concern are between 5 and 20 µm. Particles smaller than 5 µm are likely to remain suspended in the air of a facility for an extended time (long enough to reach the air handling unit), whereas particles larger than 20 µm are likely to settle quickly and be deposited on surfaces in the room where they can be removed by adequate cleaning and sanitation techniques. In most cases, pre-filters with MERV ratings of 8 or 9 sufficiently capture contaminants found in the outdoor and return air entering the air handler. A more restrictive final filter is needed downstream of the cooling and reheat coils since the air handling unit can be a growth niche for pathogens, especially on the wet surfaces of cooling coils. For the final filters HEPA filtration is desirable, however if this is impractical or unnecessary, the final filters should be designed to capture particles 5 µm or larger at a minimum (MERV 12 to 14).

Replacement frequency of filters will depend on a number of factors, and will vary for the pre-, intermediate, and final filters. This replacement frequency must be established by the food processor according to their hazard analysis, food safety plan, and supply air cleanliness requirements. As the filters load up with contaminants, air pressure drop across the filters will increase. This differential pressure drop

can easily be measured and used to signal operators to replace filters at an appropriate setpoint. This method of determining replacement frequency for pre- and intermediate filters is recommended. Replacement frequency of final filters can also be determined by differential pressure measurement, but some minimum replacement frequency should also be considered. Regardless of replacement frequency, filters should be inspected visually on a regular basis to confirm that filters are intact and have not failed structurally such that air is bypassing the filter section.

XVIII. Pathogen Control Strategies Within Air Handlers

The interior surfaces of Colmac HygenAir™ Air Handlers have been designed to be fully cleanable with no hollow bodies or flat surfaces where water can accumulate. Materials of construction have been selected to resist corrosion when exposed to commonly used cleaning and sanitizing chemicals.

However, the wet surfaces of cooling coils can become a niche for pathogens if not accounted for in the design of the air handler. High populations of microorganisms have been found on cooling coils and in drainpans with standing water in air handlers [23]. Cooling coils unavoidably condense moisture from the airstream during normal operation. Most cooling coils found in hygienic air handlers have many rows deep (as many as 16) in the direction of airflow and are inherently challenging to thoroughly clean and sanitize either manually or with clean-in-place (CIP) nozzles. Hence, if these wet surfaces are not (or cannot be) thoroughly cleaned and sanitized, pathogens and other biological contaminants (mold and yeast) will grow. Coil face velocity must therefore be limited to 500 to 600 fpm to avoid carryover of water droplets from the coil fins into the airstream. A properly designed flashing must also be installed on the downstream side of the coil section to prevent any air from bypassing around the coil fins or through the drainpan. Air velocities across wet surfaces such as coil fins or drainpans in excess of 780 fpm (4 m/s) will generate fine droplets which could aspirate contaminants and carry them into the airstream. While the average coil face velocity may be designed for 500 fpm, for example, bypassing air through poorly designed flashing and dead plates can easily reach velocities in excess of 1,000 fpm. The pathogens or other biological contaminants (yeast and mold) aspirated by this high velocity bypass air will be deposited on the surface of the final filters where it can grow and ultimately cause failure of the filter media and contamination of the supply air.

One method of killing and/or inactivating pathogens is by specifying Colmac Anti-microbial fin stock on cooling coils. Details on this important optional feature are shown in the “Anti-microbial Fin Alloy” section above. This is a completely passive technique for eliminating pathogens on coil fin surfaces and so does not require active control or power consumption for the life of the equipment.

Two other active means of controlling pathogens and biological contaminants on cooling coil surfaces are available as optional features:

1. UV-C lighting
2. Airstream ionization

UV-C Lighting

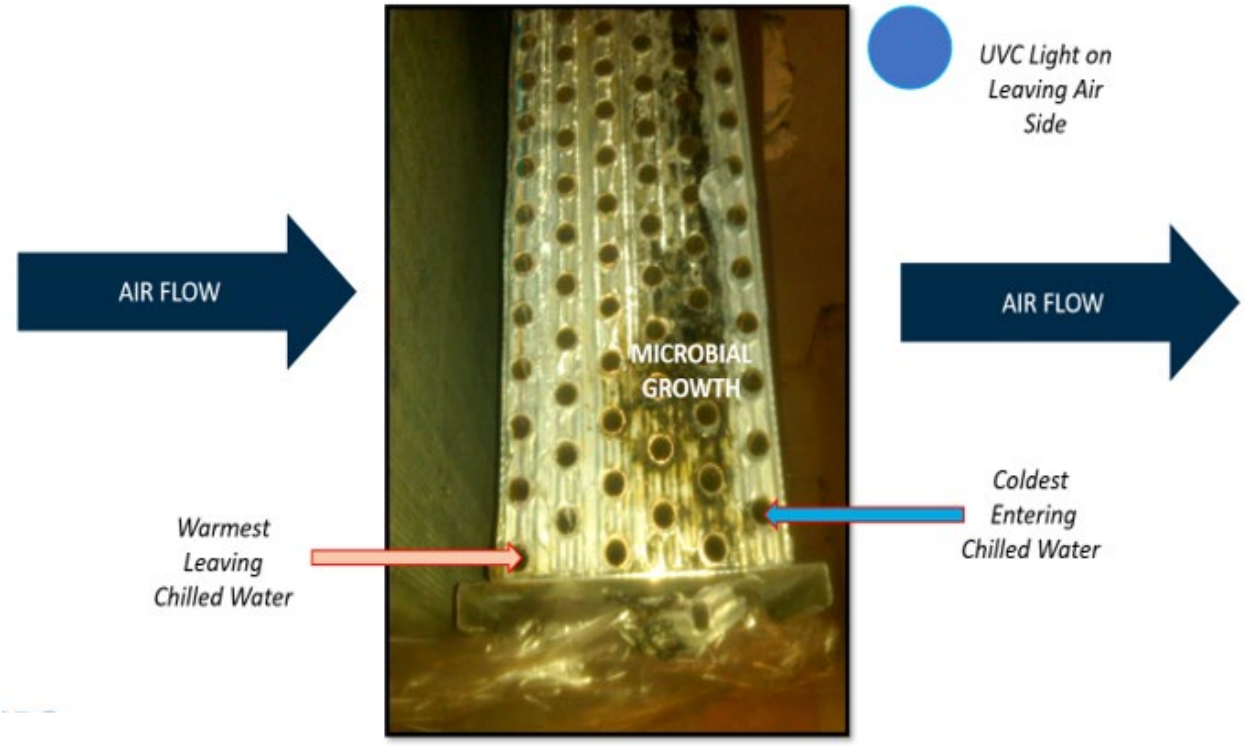
This method of active disinfection has been available for some time. While this technology may be effective in some cases (direct exposure to flat surfaces), it is of very limited effectiveness when applied to finned coils having a depth in the direction of airflow of more than 1 row. Other disadvantages of this technology include inherent safety risks due to exposure of service personnel to UV light, risk of breakage during changeout and maintenance, and excessive power consumption. Figure 14 below shows a cooling coil taken from an air handler fitted with UV-C lights on the leaving air side of the coil. In this case the UV-C lights did not prevent microbial growth in the interior rows of the coil.

Airstream Ionization

With this technology, ionization emitters are mounted across the width of the coils on the air entering side spaced roughly every 60” in the vertical dimension. The emitters “blanket” the airstream entering the cooling coils with ionized air which contacts coil surfaces and very effectively kills and inactivates pathogens, stopping microbiological growth through the entire depth of the coil. Interestingly, ionization of the airstream can also control certain gaseous contaminants such as odors. Comparing airstream ionization with UV-C lighting, ionization offers the following benefits:

- Lower first cost
- Lower operating cost
- Effective kill and inactivation of microbiological organisms throughout the coil depth
- Zero health and safety risk to maintenance personnel
- Lower maintenance cost
- No replacement cost (if maintained properly)
- Low risk of breakage during normal handling

FIGURE 14: BIOLOGICAL GROWTH ON COOLING COIL EQUIPPED WITH UV-C LIGHTS



Proper design and maintenance of drain traps on the air handler drainpans is also critical to avoid plugging and creating areas of standing water. Preventing insects and vermin from entering the air handler through drains is also a critically important consideration which must be incorporated in the design of the drain piping system.

Ductwork, especially on the supply side of the air handler, must be maintained and kept free of penetration that may allow entry of contaminants and moisture. Ductwork must be insulated adequately to

prevent condensation on the interior surfaces, and doors spaced conveniently for inspection and cleaning must be included in the design.

XIX. References

[1] ASHRAE (2017). Fundamentals Handbook, Chapter 1 Psychrometrics. ASHRAE, Atlanta, GA 30329 USA.

[2] ASHRAE (2017). Fundamentals Handbook, Chapter 11 Air Contaminants. ASHRAE, Atlanta, GA 30329 USA.

[3] ASHRAE (2017). Fundamentals Handbook, Chapter 18 Nonresidential Cooling and Heating Load Calculations. ASHRAE, Atlanta, GA 30329 USA.

[4] ASHRAE (2018). Refrigeration Handbook, Chapter 24 Refrigerated-Facility Loads. ASHRAE, Atlanta, GA 30329 USA.

[5] Colmac Coil (2018). Refrigeration Load Simulator Software. Colmac Coil Manufacturing, Inc. Colville, WA USA

[6] Nelson B. (2016). DX Ammonia Piping Handbook, 4th Edition. Colmac Coil Manufacturing, Inc. Colville, WA USA

[7] Brody A. (2011). “Cleanroom Technologies in Food Processing and Packaging”. Food Technology Magazine, July 2011, Vol 65, No. 7. Institute for Food Technologists, Chicago, IL 60607 USA.

[8] Hejab M. (1992). “Prediction of airborne contamination in conventionally ventilated cleanrooms”. PhD Thesis, University of Glasgow, Scotland.

[9] Sundstrom S, Ljungqvist B, Reinmuller B. (2007). “Some observations on airborne particles in blow-fill-seal filling rooms”. PDA Journal of Pharmaceutical Science and Technology 61(3): pp147-153.

[10] Hnatek ER. (1995). “Integrated Circuit Quality and Reliability, Second Edition”. CRC Press, Boca Raton, FL, USA.

[11] Whyte W, Agricola K and Derks M. (2016). “Airborne particle deposition in cleanrooms: relationship between deposition rate and airborne concentration”. *Clean air and Containment review; 2016* Issue **25**:4–10. EC Pharma Ltd (Euromed Communications), Passfield Business Centre, Lynchborough Road, Passfield, Hampshire, UK

[12] Whyte W, Eaton T. (2016). “Deposition velocities of airborne microbe-carrying particles”. European Journal of Parenteral & Pharmaceutical Sciences 2016; 21(2): pp 45-49. Pharmaceutical and Healthcare Sciences Society, Swindon, Wilshire SN2 7RR UK

[13] Whyte W, Eaton T, Whyte WM, Lenegan N, Ward S, Agricola K. (2017). “Calculation of air supply rates and concentrations of airborne contamination in non-UDAF cleanrooms”. European Journal of Parenteral & Pharmaceutical Sciences 2017; 22(4): pp 126-138. Pharmaceutical and Healthcare Sciences Society, Swindon, Wilshire SN2 7RR UK

[14] Whyte W, Ward S, Whyte WM, Eaton T. (2014). “Decay of airborne contamination and ventilation effectiveness of cleanrooms”. International Journal of Ventilation, 13(3); pp. 1-10. INVIE vzw – the International Network for Information on Ventilation and Energy Performance, Ghent, Belgium.

[15] Whyte W, Whyte WM, Eaton T and Lenegan N. (2014). “Calculation of air supply rates for non-unidirectional airflow cleanrooms”. *EuropeanJournal of Parenteral and Pharmaceutical Sciences* 2014;**19**(4):121–129. Pharmaceutical and Healthcare Sciences Society, Swindon, Wilshire SN2 7RR UK

[16] Sterling, E.M., A. Arundel, and T.D. Sterling. (1985). “Criteria for human exposure to humidity in occupied buildings”. *ASHRAE Transactions* 91(1B):611-622. ASHRAE, Atlanta, GA USA.

[17] Michels H.T., Wilks S.A., Keevil C.W. (2004). “Effects of Copper Alloy Surfaces on the Viability of Bacterium, E. coli O157:H7”. Hygienic Coatings & Surfaces, Conference Proceedings 26-28 Jan, 2004. The Paint Research Association. Teddington, TW11 8LD UK. ISBN 0-9543164-5-2

[18] Warnes S.L., Little Z.R., Keevil C.W. (2015). “Human Coronavirus 229E Remains Infectious on Common Touch Surface Materials”. mBio Journal, Nov/Dec 2015 Vol 6;Issue 6. American Society for Microbiology, Washington DC USA

[19] Personal correspondence with AAF/Flanders factory engineers. (2023).

[20] AAF/Flanders (2020). “Air Filtration Reference & Product Guide”. American Air Filter Company, Inc. a member of the DAIKIN Group. 9920 Corporate Campus Drive, Suite 2200, Louisville, KY 40223-5690.

[21] Stier R. (2015). “Handling the Air in Your Plant”. Food Engineering Magazine, Nov 2015. BNP Media, Troy, MI 48084 USA

[22] Kornacki J.L. (2014). “Airborne Contamination: A Microbiologist’s Perspective”. Food Safety Magazine, Jun 2014. BNP Media, Troy, MI 48084 USA

[23] Hugenholtz P., Fuerst J.A. (1992). “Heterotrophic Bacteria in an Air-Handling System”. Applied and Environmental Microbiology, Dec 1992, p. 3914-3920. American Society for Microbiology, Washington DC 20036 USA

[24] Nelson B. (2022). “Low-charge DX Ammonia Industrial Plant Design”. Technical Bulletin dated 26 Sep 2022. Bruce V. Nelson Engineering, LLC. Colville, WA 99114 USA

[25] King Company (2002), “King Food Plant HVAC Selection Guidelines” Nov 2002, King Company, Owatonna, MN 55060 USA

XX. APPENDIX A: OVERVIEW OF US FOOD FACILITY REGULATIONS

FDA has regulations specifying **Good Manufacturing Practices** (GMPs) for the food processors that FDA is responsible for regulating (*see* **21 CFR Part 110** and **21 CFR Part 117 Subpart B**). GMPs address a variety of topics.

USDA has some comparable regulations for egg processors; these **Operating Procedures** (**9 CFR §§590.500 – 590.575**) address topics similar to FDA’s GMPs, but are more specific because the egg operating procedures address a single food industry.

USDA also has **Sanitary Standard Operating Procedures** (SSOPs) for the meat and poultry industries (*see* **9 CFR Part 416**). Again, many of the topics addressed in the SSOPs are similar to topics addressed in FDA's GMPs.

In addition, FDA has some detailed expectations for the juice and seafood industries to augment the GMPs (e.g., **21 CFR §120.6** and **21 CFR §123.11**); FDA also refers to these additional requirements for juice and seafood as SSOPs – *a source of possible confusion*.

FDA refined its GMPs (in late 2015). These revisions (codified at **21 CFR Part 117 Subpart B**; but 21 CFR Part 110 remains), of course, do not apply to egg products, meat and poultry processing which are regulated by USDA. Restated, USDA is not updating the operating procedures for egg processing or the SSOPs for meat and poultry processing at this time (early 2016).

FDA has required juice and seafood processors to develop and implement **HACCP** plans (**21 CFR Part 120** and **21 CFR Part 123**, respectively); these plans are underpinned by GMPs (see 21 CFR §120.5 and 21 CFR §123.5) and continue to be underpinned by the newest GMPs (**21 CFR Part 117 Subpart B**).

Similarly, USDA requires meat and poultry processors to have HACCP plans (**9 CFR Part 417**); although not explicitly stated in the USDA regulations, one would expect a relationship between the processor's SSOPs and its HACCP plan.

In late 2010/early 2011, Congress enacted the FDA Food Safety Modernization Act to require all FDA food processors (except seafood and juice processors) to develop and implement a **Food Safety Plan** (**21 U.S.C. §350g**), occasionally described as the "next generation of HACCP."

- Prior to the Food Safety Modernization Act, food processors overseen by FDA (except seafood and juice processors) were not required to have a HACCP plan.
- Congress allows seafood and juice processors to continue to apply existing HACCP regulations and plans, rather than require those businesses to revise their practices to align with the requirements of a Food Safety Plan.
- The Food Safety Plan of the Food Modernization Act has no impact on USDA, or the HACCP requirements that USDA has in place for meat and poultry processors.

FDA regulations for the development and implementation of Food Safety Plans are codified at **21 CFR Part 117 Subpart C**.

The primary components of 1) FDA HACCP plans for juice and seafood processors, 2) USDA HACCP plans for meat and poultry processors, and 3) the Food Safety Plans for other food processors subject to FDA oversight are similar. There are also some significant differences that warrant mention.

Detailed information on the specific requirements of these regulations is outside the scope of this document, and so the reader is referred to the references and food safety professionals for guidance.

For more information contact Colmac Coil Manufacturing, Inc.

PO Box 571 | Colville, WA 99114
1.800.845.6778 | 509.684.2595
info@colmaccoil.com | www.colmaccoil.com