Air Cooler Manual

TYR-D air cooler

TYR-T air cooler

Customized air cooler

Index

Preface

In vapour compression refrigeration systems the major components are compressor, evaporator and condenser besides expansion device. Although compressor is called as heart of the system, since it is the only moving component and most expensive one, the stored products do not recognize as to what is the Refrigeration tonnage or what is the type of compressor, condenser etc. The preservation of the product quality is predominantly dependent on the evaporator performance and the high side of the system becomes the back end or supporting part since it is the air cooler which is directly exposed to the stored commodity.

The evaporator or air cooler used for preservation of foods therefore plays the most important part since it circulates the air in the cold room at a temperature and humidity required to be maintained for a particular product to be stored. To maintain uniform temperature in the room and of the product, the air circulation distribution is also very important.

The air cooler selection for each application is unique which would meet the expected requirements. The selection of air cooler for high humidity storages for products such as vegetables, fruits is much different than air cooler selection for low humidity product storages such as onions, seed grains etc. The requirement of blast freezers or precooling requires altogether different approach while selecting the air coolers.

In view of large range of evaporators available in the market having different coil geometries, material of construction, defrosting methods, circuit options, there is no uniformity in technical specifications of air coolers offered by various manufacturers/suppliers. Many important factors are presented in a manner which makes the “real” comparison difficult. Often the selections are based manufacturer software & the comparison of different air coolers offered makes it difficult for the buyer to decide as to which is the right cooler for his requirement.

This guide book is intended to provide refrigeration plant designers, operators, consultants and end users the information needed to make appropriate selection and to serve as a desk reference.

The information presented deals with ammonia air coolers for cold storages, and freezer rooms, blast freezers, processing halls, pre-cooling chambers or any other related end uses for preservation of food.

The document has been prepared and information compiled based on various papers published, text books, ASHRAE volumes, IIAR publications to meet the requirements of engineers, contractors, dealers, actual users etc. without going in to too much theoretical aspects so that it is easy for the reader to absorb the information without missing the important aspects while selecting the air coolers for particular application he has in mind.

The guidebook is prepared by Mr. Ramesh Paranjpey - ASHRAE Fellow Life member and Alfa Laval India team to make the decision makers and other concerned persons aware of the different aspects which need to be considered while comparing air coolers available in the market.

We are sure that the information presented would lead to more appropriate selection of forced circulation air coolers and successful commodity storage application leading to maintain top class product quality without deterioration or weight loss while retaining all the nutritional values.

Refrigeration, Introduction and key definitions

Introduction

Refrigeration is essential and absolute must for food preservation. Heat removal from product via cooling or freezing of product prevents, or slows down microbial and chemical changes in the product. Temperature and humidity as well as air distribution within refrigerated spaces are major considerations in food preservation processes such as post harvest cooling, blast freezing, and product processing or product storage. The timing associated with product temperature pull down, heat removal, is critical to the final condition and market life.

The evaporator is the crucial component within any refrigeration system being responsible for removal of the heat from product and maintaining uniform temperature and air distribution to ensure no stagnant area exist or no appreciable weight loss takes place

Definitions:

1. Forced-Air Circulation Unit Coolers (Unit Coolers). -

A factory-made assembly, including fans for forced air circulation and coil by which heat is transferred from air to refrigerant. These may also be referred to as Air Coolers, Cooling Units, Air Units, unit coolers, product coolers or Evaporators.

2. Liquid Overfeed Unit Cooler. -

A Unit Cooler in which the refrigerant liquid is supplied at a Recirculation Rate greater than one. It can be gravity flooded or forced feed pump circulation system design.

3. Gross Total Cooling Effect (Cooling Capacity).-

The heat absorbed by the refrigerant, W(Btu/h) . This is the sum of the Net Total Cooling Effect and the heat equivalent of the energy required to operate the Unit Cooler. This includes both sensible and latent cooling.

4. Net Total Cooling Effect. -

The refrigeration capacity available for space and product cooling, W (Btu/h). It is equal to the Gross Total Cooling Effect less the heat equivalent of energy required to operate the Unit Cooler. This includes both sensible and latent cooling.

5.Overfeed Ratio. -

The mass ratio of liquid to vapor at the outlet of the Liquid Overfeed Unit Cooler. This may also be referred to as overfeed rate.

6.Recirculation rate-

(Overfeed rate +1) or The mass ratio of liquid circulated to the amount of liquid vaporized

7.Refrigerant Saturation Temperature

Refrigerant temperature at the Unit Cooler inlet or outlet determined either by measuring the temperature at the outlet of the two-phase refrigerant flow, for a Liquid Overfeed Unit Cooler, or by measuring refrigerant pressure and determining the corresponding temperature from reference thermodynamic tables or equations for the refrigerant.

8.Temperature Difference (TD)

The difference between the dry-bulb temperature of the air entering the Unit Cooler and the Refrigerant Saturation Temperature at the unit cooler outlet.

9.Enthalpy Difference (HD).

The difference between the enthalpy of the air entering the Unit Cooler and the calculated enthalpy of saturated air at the Refrigerant Saturation Temperature at the Unit Cooler outlet, J/kg (Btu/lb).

10.Rated Power.

For single phase motors, total fan motor input power, W or kW.

For poly phase motors, individual fan motor output power, kW (hp).

11.Standard Air Conditions. Dry air at 21 ºC (70 ºF) and absolute pressure 101.325kPa(29.92 in Hg). Under these conditions, dry air has a mass density of 1.2 kg/m3 (0.075 lb/ft3).

1.What is Air Cooler-Evaporator:
Air cooling ammonia Industrial evaporators are refrigerant to air heat exchangers widely used in industrial refrigeration applications. They are also known as forced circulation air coolers or air cooling evaporators, or other names as indicated n the definition 1

2.What is the principal of operation:
These are the heat exchangers using tubes that carry ammonia refrigerant inside the tubes. The tubes are externally provided with fins to increase outside heat transfer area. Liquid refrigerant evaporates inside the tubes as it absorbs heat from air flowing over & coming in contact with outside surface of fins and tubes. Individual tubes of heat exchanger are arranged in multiple rows of parallel circuits to achieve increased thermal performance. The external cold fin surface cools the air coming in contact is circulated by fans at a designed velocity.

3.What are the fields of applications:
The ammonia refrigeration air coolers are widely used to cool and circulate air in cold storages, warehouses and other food processing facilities.

4.What is the Function of Air Cooler:
Heat removal from the product via cooling or freezing of the product prevents, or retards, microbial and chemical changes in the product. In today's world product quality is paramount for the end user and vital to realized revenue. Whether this is related to post harvest cooling, blast freezing, process cooling or product storage many factors will ultimately determine the quality of the product being refrigerated. Temperature and air distribution within refrigerated warehouses are major considerations. The timing associated with product temperature pull down, heat removal, is critical to the final condition, and market life, of the respective product. Temperature deviations of 2°F (1.1K) to 3°F (1.6K) above or below the desired set point are often too great and could be harmful to the product. Air distribution within the facility should be constant throughout and care should be exercised to ensure no stagnant areas exist within the refrigerated space.

5.The evaporator/s is the crucial component within any refrigeration system and performs all these functions. The air coolers are widely used in commercial and industrial refrigeration applications in various segments of food preservation to cool and circulate air in cold storage warehouses.

a. In Fruit and vegetable and other positive temperature cold storages the function of air cooler is to cool the product from ambient temperature to desired temperature and humidity levels in prescribed time limit as also to store the product for a expected duration in the cold storages without deteriorating product quality or weight loss. The cooling unit is not supposed to improve the quality of incoming product but to maintain the same quality throughout its storage life retaining the quality which was there at the time of loading.

b. The unit coolers can be used in processing halls, grading and sorting halls, ante rooms etc.

c. The air coolers are used for negative temperature for storing frozen products such as meat ,fish, vegetables generally at -200C

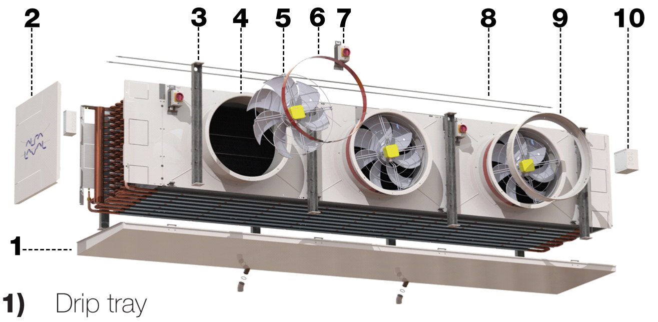
d. The unit coolers are used for Ice cream storage at much lower temperature around -300C

e. The unit coolers are used in food freezing applications such as blast freezers, tunnel freezers, spiral freezers, IQF , and many other products generally at -400C

f. The unit coolers are also used in variety of other applications

Conceptual knowledge of air cooler

These are the heat exchangers using tubes that carry ammonia refrigerant inside the tubes. The tubes are externally provided with fins to increase outside heat transfer area. Liquid refrigerant evaporates inside the tubes as it absorbs heat from air flowing over & coming in contact with outside surface of fins and tubes. Individual tubes of heat exchanger are arranged in multiple rows of parallel circuits to achieve increased thermal performance. The external cold fin surface cools the air coming in contact, which is circulated by fans at a designed velocity.



1) Drip tray
2) End cover (left/right identical)
3) Mounting feet
4) Fan cowl/ring
5) Fan unit complete
6) Fan ring heater
7) Safety/repair switch
8) Electric defrost heater
9) Airsock ring
10) Connection box

Fig. 1

Expectations from a good Air Cooler:

Heat removal from the product via cooling or freezing of the product prevents, or retards, microbial and chemical changes in the product. In today's world product quality is paramount for the end user and vital to realized revenue. Whether this is related to post harvest cooling, blast freezing, process cooling or product storage many factors will ultimately determine the quality of the product being refrigerated. Temperature and air distribution within refrigerated warehouses are major considerations. The timing associated with product temperature pull down, heat removal, is critical to the final condition, and market life, of the respective product. Temperature deviations of 1.1 to 1.6K above or below the desired set point are often too great and could be harmful to the product. Air distribution within the facility should be constant throughout and care should be exercised to ensure no stagnant areas exist within the refrigerated space.

Field of applications of Air cooler :

The air coolers are widely used in commercial and industrial refrigeration applications in various segments of food preservation to cool and circulate air in cold storage warehouses.

The following are few examples of applicator of an Air Cooler

a. In Fruit and vegetable and other positive temperature cold storages the function of air cooler is to cool the product from ambient temperature to desired temperature and humidity levels in prescribed time limit as also to store the product for an expected duration in the cold storages without deteriorating product quality or weight loss. The cooling unit is not supposed to improve the quality of incoming product but to maintain the same quality throughout its storage life retaining the quality during the time of loading.

b. The air coolers can be used in processing halls, grading and sorting halls, ante rooms etc.

c. The air coolers are used for negative temperature for storing frozen products such as meat, fish, processed vegetables generally at -20°C

d. The unit coolers are used for Ice cream storage at much lower temperature around -30°C

e. The unit coolers are used in food freezing applications such as blast freezers, tunnel freezers, spiral freezers, IQF , and many other products generally at -40°C

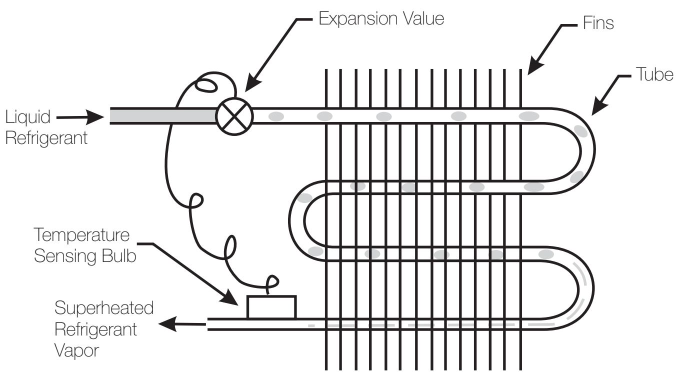
f. The unit coolers are also used in variety of other applications

Types of ammonia refrigeration systems

1.What are the standard methods of supplying ammonia refrigerant to the evaporator

a. Direct Expansion
b. Gravity Flooded
c. Pump circulation-overfeed

2.Describe operation of direct expansion (DX)system:



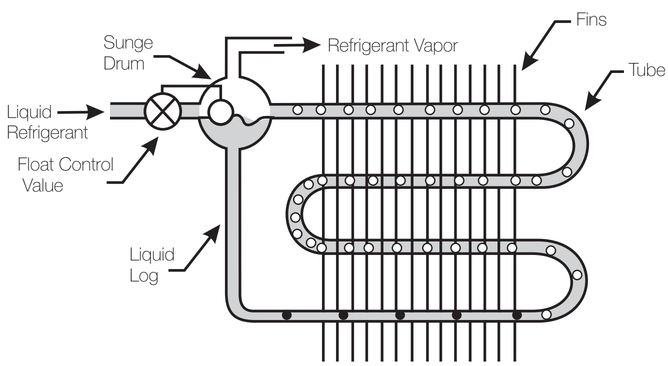
(a) Direct Expansion Air Coil - Fig 1

The high pressure liquid ammonia is expanded in the expansion valve and mixture of low temperature and pressure liquid plus vapour mixture enters the tubes. The mixture is having predominant part of liquid and some quantity of flash gas. In the evaporator tubes the liquid ammonia gradually evaporates as it travels to wards suction header by absorbing heat from air coming in contact with tubes and fins and in turn cools the air. The refrigerant leaving the evaporator is superheated to the extend based on superheat setting of expansion valve which is normally around 5^{0}C. the refrigerant liquid+vapour mixture travels from inlet to outlet when liquid component is reduced and vapour component increases and at the outlet one gets only superheated vapour. The travel of mixture is only in one direction from inlet to outlet and there is no recirculation as the vapour coming out of evaporator directly goes to compressor suction and passes trough entire system before it enters the evaporator again.

The direct expansion ammonia coolers are used generally for positive or above freezing temperature applications for very small requirements such as process, grading/sorting halls etc.

The direct expansion air coolers in ammonia are seldom used due to high latent heat of ammonia and the flow rate to achieve a given refrigerant capacity will be too low with ¾ or 1" tubes. Another important issue is to ensure oil return to compressor with such low ammonia velocities since ammonia and oil being not miscible the oil would tend to remain in the coil endangering the compressor operation. The use of ammonia miscible oils has enabled use of Direct Expansion evaporators in above freezing temperature applications in smaller capacities using tubes less than 5/8" diameter.

3. Describe operation of Gravity Flooded Evaporator systems:



(b) Flooded Air Control - Fig 2

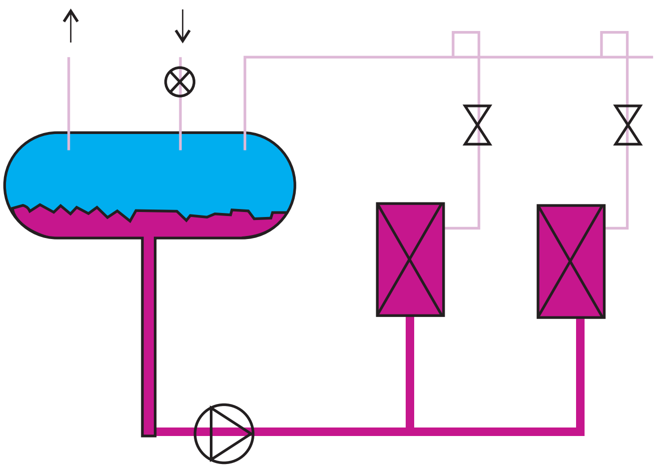
The refrigerant in the evaporator is mostly liquid (flooded) from the beginning to the end of the process. The flooded evaporator provides for recirculation of refrigerant within the evaporator by the addition of a accumulator/surge drum. The liquid refrigerant enters the accumulator drum through the metering device and gravity causes it to flow down to the bottom tube.

The entire coil surface is in contact with wet refrigerant under any load condition. This design produces excellent heat transfer. The vapour produced in the evaporator is separated from liquid in surge drum. The liquid is re-circulated through the evaporator again, while the vapour is sucked by the suction action of compressor.

The flooded evaporator regulates refrigerant flow by a float device, which is designed to maintain a predetermined liquid level in surge drum. The vapour leaving the surge drum is saturated and not superheated as in case of DX evaporator. Flooded evaporators are therefore more efficient as entire coil surface is exposed to wet refrigerant and therefore the heat transfer is better. In dry evaporators part of coil area is wasted in superheating gas and the coil surface is always in contact with part liquid and part gas.

Although these are more efficient compared to DX evaporators, careful design has to be made to ensure proper liquid/vapour separation in surge drum, it's location to ensure that liquid is not carried over to the compressor. The design of surge drum, its various connections and velocity of refrigerant has to be taken care of.
Ammonia refrigeration systems normally work on flooded operation principle.

4. Describe functioning of Pump circulation (overfeed) systems:



Pumped System - Fig 3

In liquid overfeed systems the refrigerant liquid coming out of receiver is expanded to the required pressure/temperature and this liquid is stored in low pressure receiver. It is then pumped in the various operating evaporators, like product coolers, blast freezers or plate freezers. It thus forms an independent low side circuit. The compressor sucks the vapour from this low pressure receiver and the cycle repeats.

The overfeed means much more liquid is fed to evaporator than the liquid actually vaporizes. Excess liquid is called overfeed, which returns to low pressure side accumulator or known as L.P. receiver. Thus the mass flow rate handled by compressor is less than the mass flow rate circulated in the evaporator.

As the number of evaporators increase and as the temperature requirement gets lower and lower, liquid recirculation/overfeed systems are preferred. Normally for more than 3 to 5 evaporators & located considerably away from machine room, liquid recirculation is the best option. Properly designed flooded evaporators and evaporators operating with liquid recirculation operate with equal effectiveness. What is then is the basis for choosing between flooded coils and a liquid recirculation system serving multiple coils?"

The dominant refrigeration system for intermediate to large storage facilities is the liquid overfeed ammonia refrigeration system. This system is suitable for low and medium temperature cold storage."

The use of liquid overfeeds system is therefore advantageous:
1.When there are more than 4 to 6 evaporators of larger capacity in medium or large cold size cold storages.

2.Plant room is located far away from the processing area where evaporators are located involving lengthy refrigerant distribution pipe work.

3.Special evaporators like spiral freezers/IQF or plate freezers are involved.

4.The requirement is for medium or low temperature commodity storage.

5.Hot gas defrost systems to be used

Air cooler design fundamentals

1.Define what is Primary surface and Secondary surface:

Evaporator coils are constructed using tubes and fins bonded to the tubes. Fins are referred to as 'secondary surfaceA_0' and tubes are referred as 'primary surface A_i'

2.Which surface is more effective:

For purpose of calculating evaporator performance, primary surface is more effective since the liquid ammonia at lowest temperature is inside the tube and heat transfer rate is highest because of higher 'TD' between air and refrigerant. The primary surface is therefore considered as 100% effective in its contribution to its contribution to the total heat transfer surface area.

The secondary surface has a surface effectiveness less than 100% due to the change in surface temperature from the root to the tip of the fin

How the effective heat transfer area required is then calculated?

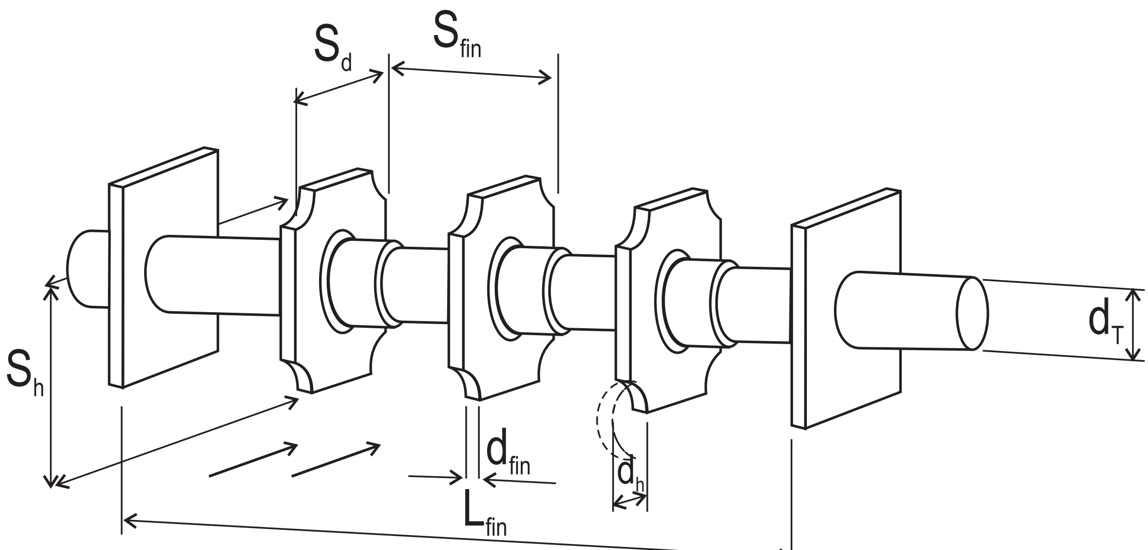


Fig 4

Where:

S_h	is	Fin height pert tube	(Unit)
S_d	is	Fin depth per tube	(mm)
S_{fin}	is	Fin Spacing	(mm)
r	is	ratio of enhanced fin surface area to flat, claimed by applicant (to be supported by drawing and calculations if grater than 1)	
d_T	is	Tube outside diameter	(mm)
d_h	is	Diameter of other holes in fin e.g. heater holes	(mm)
L_{fin}	is	Finned length	(m)
n	is	Number of other holes (d_h) per tube holes(d_T)	

The Surface area per tube, per meter - finned length is: $SA=(2/S_{fin}/1000) \times r \times (S_h \times S_d - \pi/4 \times (d_T^2 + n \times d_h^2)) + \pi \times d_T/1000$

The Total Surface Area(m^2)

$TSA = SA \times L_{fin} \times$ Number of Tubes

There are two important equations which cover the selection

a. The standard equation for heat flow is

$Q=(A_0 \times U \times LMTD)$

Where 'Q' = Total heat transfer rate

'U'= overall Heat Transfer Co-efficient

'LMTD' or ΔT = logarithmic Mean Temperature Difference

This equation helps in determining the required heat transfer area for the cooling coil and once selected/manufactured and supplied cannot be altered at site.

b. The second equation is

Qsensible = m x Cp (T air entering-T air leaving)

Or

$$qs = \frac{m3}{hr} \times \rho \text{ (air density } \frac{kg}{m3}\text{) x Cp (T air entering-T air leaving)}$$

Or

$$q\ total=\frac{m3}{hr} \times \rho \text{ (air density } \frac{kg}{m3}\text{) x Cp (h1 air entering-h2 air leaving)}$$

h1= Enthalpy of air entering coil

h2= enthalpy of air leaving coil

This equation allows us to select required air quantity to be circulated in order to maintain uniform temperature inside the cold room. The air quantity can be varied by changing the fan speed or switching off one or more fans and this would alter the air cooler performance and the conditions in the cold room.

Factors under manufacturer's or designer's control

The first equation has two parts, the first being 'U' value or overall heat transfer coefficient and it is under the coil designer's control. The coil designer has control over following parameters

1. Tube and fin material of construction
2. Fin spacing
3. Fin pattern-staggered or parallel
4. Fin type –plain plate or any other pattern to increase fin area
5. Fin thickness
6. Tube diameter and thickness
7. Circuiting
8. Air and refrigerant flow pattern
9. Coil Geometry

Factors under Application Engineer's control

Whereas the application engineer has control over the coil surface area 'A' to be selected and TD, i.e. room temperature and evaporating temperature. An application engineer, while selecting the air coolers uses different combinations of ΔT and A_0 to arrive at cost effective and appropriate solution of air cooler to suit particular commodity storage requirements

1. Air quantity
2. Cold room temperature
3. Air inlet temperature to coil
4. Air out let temperature from coil
5. Whether blow through or draw through design
6. Saturated evaporating temperature at coil outlet
7. Coil pressure drop
8. Air side pressure drop
9. Air throw
10. Fan characteristic-static v/s head at operating temperature
11. Fan power consumption at operating parameters and installed motor power

We shall now analyze the first equation in more details from the manufacturer's perspective

$Q=(A_0 \times U \times \Delta T)$

Q=Total sensible heat to be transferred per unit time (kW)

'U' is the overall heat transfer coefficient and is mostly the characteristic of coil based on its design and application engineer has a limited role in its selection.

Cross sectional area of typical fin-

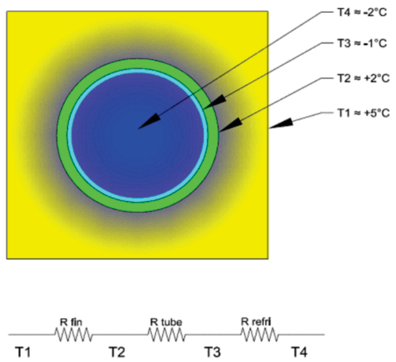


Fig 5

In the evaporator, the heat flows in series through three thermal resistances.

1. From the room air to the outside surface of the evaporator or fins in case of finned coolers-R_f
2. From the outside surface to the inside surface of the evaporator-R_m
3. From the inside surface of the evaporator to the refrigerant-R_r

   The heat flow can be compared to electrical current passing through series of resistances as shown in the fig 3.

R_f- Resistance to heat flow on air side and is dependent on air velocity, fin configuration and type of fins as well as its spacing.

η- Fin efficiency- If the entire fin were at the same temperature as the tube, the fin effectiveness would be

100%. However it is not the case and the temperature increases progressively as we move further away from the tube. All the air side area is therefore not 100% effective/efficient. The effectiveness of the fin is given by the symbol '**η**' and the value varies from 0.3 to 0.7 for commercial coils depending upon fin material and fin thickness and on type of fin.

R_m – Resistance to heat flow due to metal conductivity and thickness of the tube = x/k

R_r- Resistance to heat flow on the refrigerant side and depends upon refrigerant used, its velocity and tube diameter.

The overall heat transfer coefficient 'U' is sum of the reciprocal of all resistances and depends upon (a) the heat exchanger configuration (b) the ratio of primary to secondary surface area(c) the air velocity, temperature, density (d) properties of refrigerant and its velocity (e) tube and fin material (f) and fouling factors on either side.

$$U=\frac{1}{(R_f + R_m + R_T)}$$

The Heat flow can also be described as under

1 $Q= A_0 \times U \times (T_1 - T_4)$
2 $Q= h_f \times \eta \times A_0 \times (T_1 - T_2)$
3 $Q= \frac{k}{x} \times A_m \times (T_2 - T_3)$
4 $Q= h_r \times A_i \times (T_3 - T_4)$

Where
hf = is the heat transfer coefficient on the air side = $\mathbf{\frac{1}{R_f}}$ **(W m^{-2} k^{-1})**
hr= is the heat transfer coefficient on refrigerant side = $\mathbf{\frac{1}{R_r}}$ **(W m^{-2} k^{-1})**
hm = is the heat transfer coefficient of the tube = $\mathbf{\frac{1}{R_m}}$ **(W m^{-2} k^{-1})**
k= is the conductivity of the material **(W m^{-1} K^{-1})**
x= is the thickness of tube **-m**
A_0= is the outside or fin area which is the tube external area in case of plain tubes or **Secondary area-m^2**
A_i= is the inside area of the evaporator through which the refrigerant flows or **Primary area-m^2**

Equation 1 can be rearranged as - $(T_1 - T_4)= \frac{Q}{A_0 \times U}$

$(T_1 - T_4)$ can also be expressed as $[(T_1-T_2) + (T_2-T_3) + (T_3-T_4)]$

Substituting $(T_1 - T_4)$, (T_1-T_2), (T_2-T_3) and (T_3-T_4)
from equations 1, 2, 3 and 4 respectively,

$$\therefore \frac{Q}{A_0 \times U} = \frac{Q}{h_f \times \eta \times A_0} + \frac{Q}{k_x \times A_m} + \frac{Q}{h_r \times A_i}$$

$$\therefore \frac{1}{A_0 \times U} = \frac{1}{h_f \times \eta \times A_0} + \frac{1}{\frac{k}{x} \times A_m} + \frac{1}{h_r \times A_i}$$

$$\therefore \frac{1}{U} = \frac{1}{h_f \times \eta} + \frac{x}{k} + \frac{A_0}{A_m} + \frac{A_0}{A_i} \times \frac{1}{h_r}$$

$$\therefore \frac{1}{U} = \frac{R_f}{\eta} + R_m \times \frac{A_0}{A_m} + R_r \times \frac{A_0}{A_i}$$

The heat transfer coefficient **'U'** should be as high as possible, to make the evaporator compact and efficient, which means sum of the resistances mentioned above should be as small as possible

1. Looking at individual resistances, it means **R_f** should be as low as possible and fin efficiency **n** should also be as high as possible. As the fin area increases fin efficiency reduces since the area away from primary tube area is not as effective as area near the tube as mentioned above.

2. R_r should be as low as possible
3. A_0/ A_i should be as low as possible.(Less external secondary fin area A_0 & more internal primary area A_i)

The above equations lead to very important conclusion that the coil which has more primary surface area and less secondary surface area will have highest heat transfer efficiency.

Definitions of Terms applied to air coil

a. Coil circuit: the route of the refrigerant from the time it enters a tube until it leaves
b. Coil Depth: No of rows of tubes the air passes from entrance to exit of the coil
c. Face area: The cross-sectional area through which the air passes as it enters the coil, alternately it is also the fin height x fin width
d. Face Velocity: The volume rate of air flow divided by the face area
e. Header: A common pipe from which all circuits are supplied refrigerant. Also a common pipe which gathers all refrigerant leaving all the tube circuits.
f. Pass: The flow of refrigerant through one straight section of the circuit
g. Prime Surface: The air side area of the tubes that is in contact with air
h. Secondary surface: The surface area of both sides of the fins in contact with the air.
i.Return bends: or U bends are short sections of curved pipe to direct the refrigerant leaving one pass to the entrance of the next pass.

We shall now look at and analyze factors which are under the control of manufacturer/equipment designer

1. Material of construction for tubes and fins

Materials of construction for coil commonly considered for ammonia coolers are

a. Hot dip Galvanized Steel (Stl/Zn)
b. Stainless tubes with Aluminum fins (SS/Al)

Decision to select material depends on

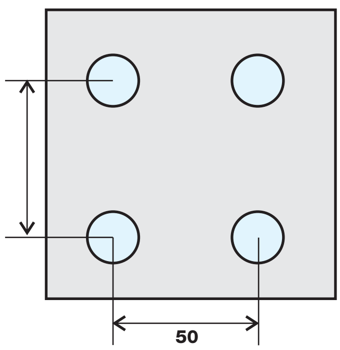
a. Performance
b. Cost
c. Life of equipment-corrosion resistance
d. Weight
e. Ease of manufacture and welding
f. Ease of Site welding and repairs if required
g. Strength
h. Reliability

The generally preferred industrial standard currently used is Stainless steel tubes and aluminum fins
Some manufacturers have also introduced aluminum tubes and aluminum fins for ammonia applications

2. Tube

The heat transfer resistance on air side is say 0.11305 m20C/W and on refrigerant side is 0.000833 m20C/W which means air side resistance is 20 times that on refrigerant side. It is important to note that the heat transfer coefficient on ammonia side is very high compared to air side and hence internally finned tubes are hardly used in ammonia applications and therefore there is no need to increase internal tube area by using grooved tubes.(Stoecker -175 page)
Also the conductivity of material has comparatively less contribution in overall 'U' value or heat transfer coefficient.

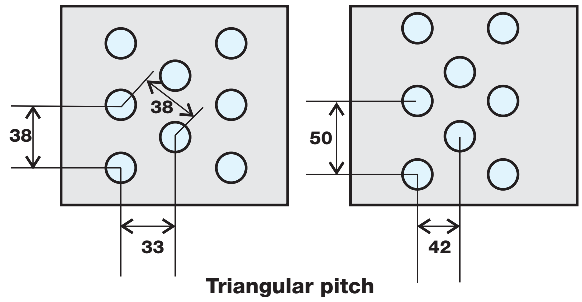
3.Fin Pattern-Square V/s Triangular



Square pitch
Fig 6

Advantages of Square fin Pattern

a. The cooler manufacture becomes simpler and sturdy
b. The frost build up is less and infrequent
c. Easy and quicker defrosting
d. Air resistance is lower
e. Motor power requirement is lower
f. Longer cooling periods since infrequent defrosting
g. Higher fin temperatures result in less dehydration, better product quality

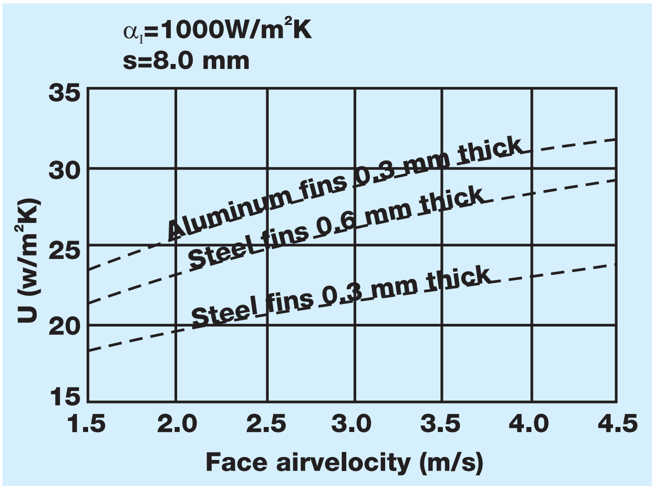


Triangular pitch
Fig 7

Advantages of Triangular fin Pattern

a. Better air side heat transfer coefficient
b. The cooler manufacture becomes compact
c. More turbulence across coil, lower fin temperature
d. Higher secondary surface area for similar configuration

Both the fin thickness and fin material count. The heat should flow from the fin to the tube. The bigger the heat load, the bigger the heat resistance. So, the thinner the fin the greater the resistance will be. This effect combined with the temperature flow in the fin is called fin efficiency. To get an idea of these effects, we would like refer to the diagram below.



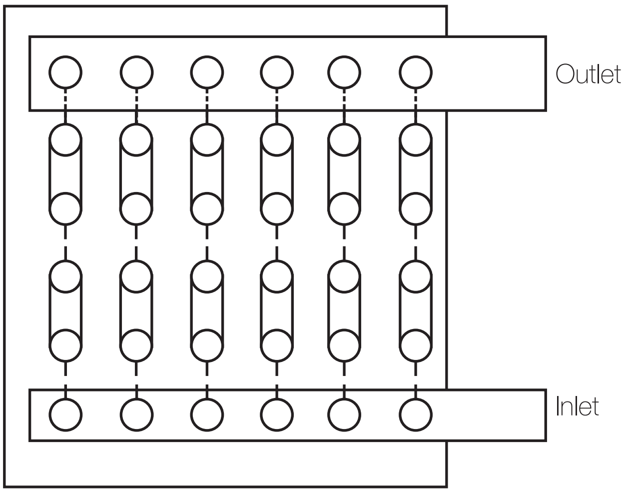
Overall heat transfer coefficient with different fin type.
Fig 8

COIL CIRCUITING:

Coil circuiting is important to the performance of the evaporator. Each coil is circuited to maximize the performance within a given set of operating parameters. Coil circuiting is designed to control the pressure drop within the coil, velocities of refrigerant and to ensure maximum wetting of coil internal tube surface.
The medium temperature or high coils have fewer circuits and longer lengths
The low temperature coils have on the other hand multiple circuits and shorter lengths.
This is due to the fact that at -40^{0}C ammonia produces 1.55

m^3/kg vapour where as at --5^{0}C it produces only 0.347 m^3/kg of vapour which is nearly 4.5 times less compared to -40^{0}C. Hence while designing low temperature coolers the circuiting is done to accommodate this extra vapour volume and hence the circuits are with smaller length and multiple circuits.

The evaporators are circuited having air flowing over the coil either with vertical or cross feed designs



Vertically Fed Coils

Fig 9

In vertically fed coils refrigerant flows from the bottom to the top or from top to the bottom. Cross over piping is utilized to balance refrigerant pressure drop and circuit load.

Vertically circuited coils in gravity flooded applications or liquid re-circulated applications do not use orifices to meter refrigerant flow to each circuit, so the circuit is unrestricted.

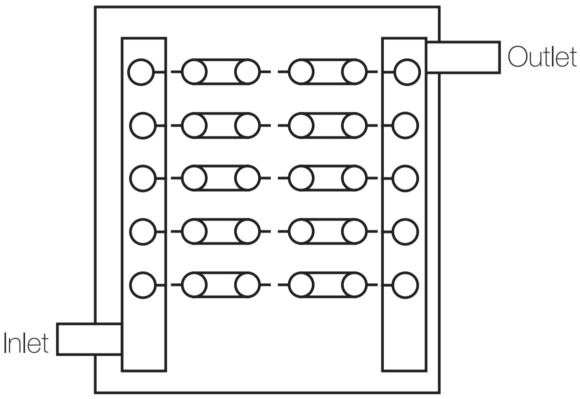


Fig 10

Cross-fed Coils

In cross fed coils refrigerant flow is either parallel or counter to the air flow. Coil pressure drop is controlled by the number and length of circuits. A cross fed coil has an orifice to meter the refrigerant feed to each circuit and will graduate the size of orifice to compensate for the static head exerted by liquid within the coil header. The size of orifices will vary based upon feed temperature and capacity. Generally the lower circuit has smaller orifice where as the upper circuits have larger orifices of 1.6 to 2mm hole diameter. The lower orifice is also eccentric to ensure no accumulation of oil in lower tubes and easy oil draining.

Cross fed coil are generally preferred in glycol systems or dx systems.

Having discussed the coil design & manufacture features we shall now discuss the parameters on which system designer or application engineer has to pay attention

The most important is the TD selection

The 2nd equation $Q_s = {}_{m.} C_p$(T air entering-T air leaving) can be written as

$Q_s = {}_{m.} C_{p.} \mathcal{E}$.(Ta.ent-T evaporating) or $Q_s = {}_{m.} C_{p.} \mathcal{E}$.(TD1) Where '$\mathcal{E}$' Is coil effectiveness=(Ta.ent-T a.lvg)/(Ta.ent-T evap.)

Heat exchanger effectiveness '$\mathcal{E}$' is defined as the ratio of the actual amount of heat that could be transferred to the maximum possible heat that could be transferred with an infinite area

Tentering =is dry bulb temperature entering the coil
Tleaving=is the dry bulb temperature leaving the coil
Tevaporating = is the average refrigerant evaporating temperature in the coil

We have discussed earlier the of fin temperature increases at positions progressively further removed from the tube and the fin effectiveness is therefore dependant on such factors as the choice of material, fin thickness, distance between the fins, air velocity etc. and varies between 0.3 to 0.7 based on these factors.

The coil effectiveness '$\mathcal{E}$' can be considered as constant for a given coil in refrigeration applications

Temperature Difference and relation to air volume

It is also important to keep in mind that in a cold room, the change in the temperature of air (reduction) as it passes through the evaporator coils has to be equal to the change in the temperature of air (increase) as it circulates throughout the room.

It means if the temperature gradient inside well designed cold room has to be limited to max 2 deg C then the coil temperature drop also gets limited roughly to 2deg C

The quantity of air circulating determines the temperature gradient maintained in the room and is clear from the equation

$$Qs = \frac{m3}{hr} \text{ xp(air den.} \frac{kg}{m3} \text{ xCp(T air ent - T air leav)}$$

Air cooler selection basis – DT1 or DTM?

DT 1= Temperature Difference' is the difference between air entering temperature to cooler or return air temperature or cold room temperature and the saturated evaporating temperature at the cooler outlet ($T_{a.ent}$-T_{evap}.)

Many other selection procedures use DT2, DTM, LMTD as temperature differences for rating the coolers

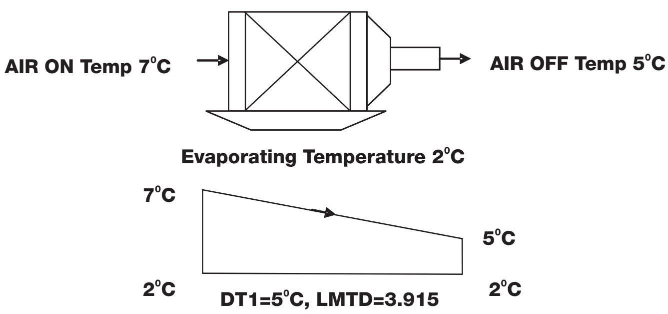
DT2=can be defined as Temperature difference Air leaving temperature from coil minus evaporating temperature ($T_{a.lvg}$-T_{evap})

DTM= Can be defined as mean temperature between coil inlet and coil outlet temperatures ($T_{a.ent}$-T_{evap}) + ($T_{a.lvg}$-T_{evap}.)/2

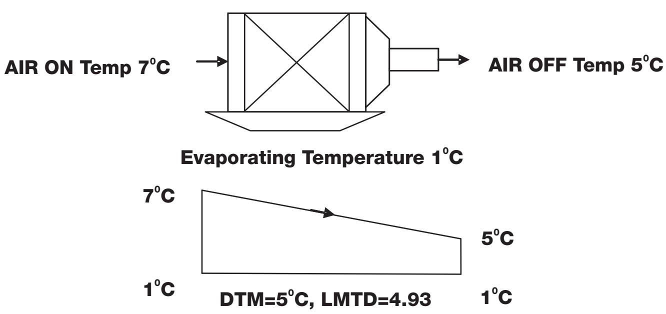
LMTD= Log mean temperature difference=

$$(T_{a.ent}-T_{evap}) - (T_{a.lvg}-T_{evap.})/ Log_e \frac{(Ta.ent-Tevap)}{(T_{a.lvg\text{-}Tevap})}$$

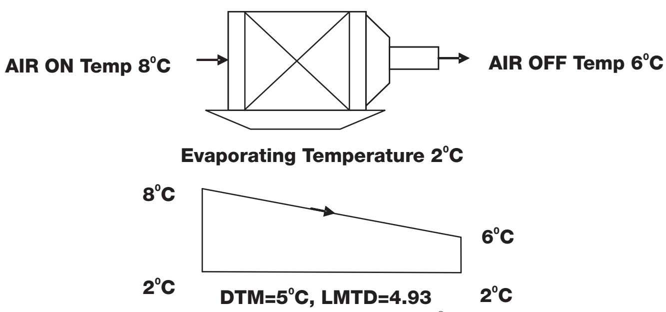
CASE 1& 2



CASE 3



CASE 4



Case 1- Assuming air inlet to coil as 7° C-Air outlet from coil as 5° C and saturated evaporating temperature of refrigerant in coil as 2° C –the DT1 becomes 5° C and LMTD is 3.915

Case 2 -if for the same conditions DT2 becomes 3°C

Case 3 (DTM)- If the cooler selection is made based on DTM method, with the same room temperature of 7° C, the air inlet temperature to coil would be 7° C , coil leaving temperature as 5° C and evaporating temperature would be 1° C instead 2° C and then DTM would be 5° C. This means the DT1 is actually 6° C and it means cooler area is 8.33% less compared to selection based on DT1 method, besides compressor is operating at 1° C evaporating temperature instead at 2° c thus consuming more power. The COP falls approximately 2.0 to 3.6 % for every 1° C reduction in suction temperature (stocker 1998) for ammonia refrigerant. Also LMTD would be 4.93

CASE4: Alternately if the evaporating temperature of 2° C is to be maintained to avoid additional compressor energy penalty then in order to have DTM as 5° C, the air inlet temperature should be 8° C and air outlet temperature would be 6° C. In this case the room temperature would be maintained at 8° C instead desired 7° C and LMTD would be 4.93

It is important to note that it is LMTD which finally determines the heat transfer area for the coil and manufacturer will allocate coil surface accordingly.

The conclusion from the above example is in DTM method the LMTD is higher by 20 to 25%. This means coil would be smaller and therefore less expensive and comparison of two coolers from different manufacturers for same duty conditions one based on DT1 and second based on DTM method is incorrect.

Also if the coolers are operating at subzero conditions then smaller surface means more frost deposits and defrost would be more frequent.

Average Room Temperature and Ratings

Control of refrigeration system is normally accomplished by maintaining room temperature, by cycling compressors and coolers by on/off operation.

When temperature is rising the equipment is switched on and when temperature is falling below set point the equipment is switched off.

Location of sensor relative to location of cooler is therefore important

Normally the coolers are located at ceiling level. The air coming on at the coil inlet is warmest and operates with highest DT1.

The floor mounted coolers on the other hand would be subjected to coldest air in the room with smallest DT1

Dt1 ratings are conservative method and recommended whenever the coil air on temperature to the coil is less than the maximum found in the room.

Conclusion:

1. The DTM rating method assumes artificially high temperature difference
2. Evaporators selected using DTM ratings will have less surface area and cost less than evaporators selected using DT1 selection method
3. Since DTM ratings result in undersized evaporator selections, the operating system suction temperature will be lower than expected. This results in greater compressor power consumption compared to evaporators selected using DT1 ratings

Air Cooler design and selection guidelines

The selection of TD necessary to obtain unit cooler performance varies with the application.
(Ref: ASHRAE Refrigeration Volume 2010 page 14.5)

Application	Required RH	DT1 recommended
Very High Relative Humidity	95%	2.8 to 4 K
High Relative Humidity in cold room	Above 90%	4 to 5 K
High Relative Humidity	85%	6 to 7 K
Medium Relative Humidity	75%	7 to 9 K
Relatively low Humidity -packaged products	60%	11 to 16 K

Sensible & Latent loads

We have so far looked at air cooler performance with sensible load only which means sensible cooling capacity of the cooler.

Whenever cooling coil surface operates at temperatures lower than dew point temperature of the air being cooled, water vapour in the air would condense and drained from the coil surface or if the temperature is below 0^{0}C frost would be deposited on the coil. This cooling effect associated with water vapour is called as latent heat removal or dehumidification.

The sum of sensible load plus latent load is called as total load. The coil should be capable of dealing with both sensible and latent heat load.

The ratio of sensible cooling load divided by total load is called as SHR and defines the slope of the air process line on psychrometric chart.

SHR = Sensible cooling load / (sensible + latent cooling load)

Or SHR= Sensible cooling load/Total cooling load

Sensible cooling load contributing factors are

1. Product cooling/freezing load
2. Transmission load through walls and ceiling
3. Air cooler electric motors
4. Forklifts
5. Lighting
6. Cooling of infiltration air
7. People

The latent heat contributors are

1. Infiltration air
2. Respiration of food products like fruits and vegetables
3. Surface moisture on product
4. Packaging material and other objects entering space
5. Human respiration
6. Humidification equipment used such as foggers for above freezing applications

Relative humidity of space can be predicted by plotting the air process line on psychrometric chart. The end point would be on saturation line at predicted coil surface temperature (ADP), and SHR line drawn from this point to room conditions. The intersection of this line with vertical line drawn from entering dry bulb temperature indicated the relative humidity of the air entering the coil.

The mass transfer process is thermally more efficient and extremely high.

The capacity of evaporator coil therefore increases when both sensible and latent heat cooling is taking place and can be expressed as
Total cooling capacity= Sensible cooling capacity/SHR

Selecting air cooler with rating based on SHR less than 1.0 would result in undersized air coolers especially where high relative humidity is required to be maintained.

Conclusion:

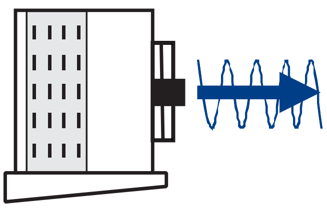
1. American air cooler manufacturers published ratings with SHR=1.0(all sensible) and DT1

2. This results in very conservative selection
3. The European manufacturers include latent heat SHR factors ranging between 0.85 to 0.95
4. Some of them also publish ratings with DT1 or DTM or both
5. Misapplication of DTM and or total cooling ratings (sensible load +latent load) can result in undersized air coolers resulting in failure of the refrigeration system to perform to expected energy efficiency levels and expected cooling performance
6. As mentioned above if the evaporators operating under wet and frosted condition are normally based on 100% sensible load.
7. For rooms operating above freezing, the moisture in the room is latent load and presence of heavy latent load can diminish the ability of the evaporator to handle sensible load.
8. For example a100kW coil selected on 50C TD is operating in wet condition. When room moisture content increases to 75% relative humidity, the coil performance also increases to say 120 kW and 20% of the performance is to handle latent load. The sensible capacity is reduced to say 96 kW. If the relative humidity increases to 90%, the total coil performance increases to say 150 kW and 41% of the coil capacity is taken up by latent load say 61 kW and thus remaining capacity of 89 kW is available for sensible load.

In short the latent load associated with moisture can have a negative effect on the overall unit sensible performance. If the design of unit fails to take moisture into account then it would not be able to maintain proper room temperature. In case of exceptionally high latent loads selection of air cooler therefore should be based on both sensible and latent heat load requirements independently instead selecting the coil on the basis of total capacity.

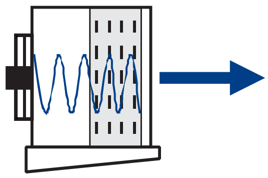
Draw through and Blow Through Execution

Draw Through: When the air first enters the coil from the room and after it is cooled the fan sucks the air from coil and delivers to room. This means the fan is after the coil



- Turbulent airstream.
- High oulet speed.
- Laminar air stream in cooler coil block (- cooler capacity).
- Evenly distributed air in the cooler coil block.

Blow Through: The air first enters the fan, and then it enters the coil after it's pressure is boosted. The fan is at the inlet of coil.



- Laminar airstream (directed through the fins!) supports the Coanda effect.
- Turbulent airstream in cooler coil block (+ cooler capacity).
- Higher delta T above the cooler coil block between the evaporating temperature and air temperature.

If we take coil inlet temperature as 7°C and supply air temperature as 5°C and evaporating temperature as 2°C and assume 0.5°C temperature rise due to fan motor heat then in case of blow through arrangement the mean temperature difference across the coil is (7.5+5)/2= 6.25°C (LMTD=4.4)
In case of draw through arrangement it would be (7+4.5)/2=5.75°C (LMTD=3.9)

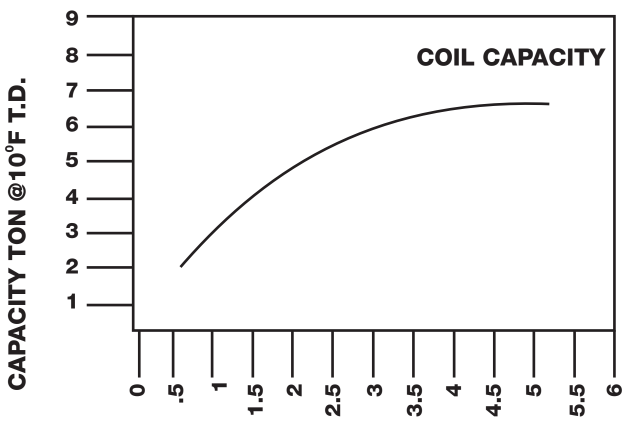
As can be seen from the example given blow through arrangement has higher mean temperature difference and hence operates more efficiently than draw through design.

The Relative humidity of Air leaving the coil is also higher in case of blow through & hence lesser product desiccation.

Parameters affecting air cooler performance

We shall now look at the effects of changing various parameters that affect the coil performance besides material of construction

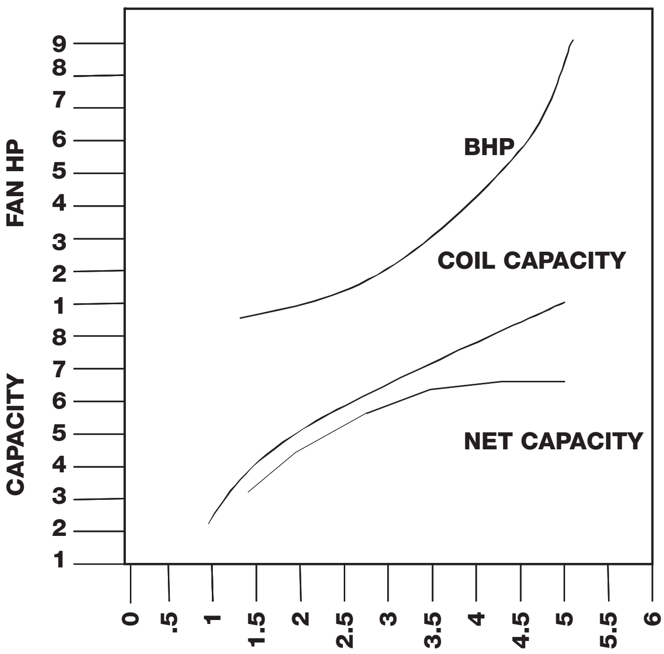
1.Velocity over the coil



AIR VELOCITY - m\s - Fig 11

The coil performance improves as the face velocity increases
The rate of increase is less at higher velocities, as can be seen from the reduced slope of the coil performance line. Typical velocities range from 2.5m\s to 5 m\s

2.Effect of fan horsepower on coil capacity



AIR VELOCITY - m/s - Fig 12

The heat transfer coefficient on the air side is, for a specific coil design, is increased by increasing the velocity of air over the coil. A point is however reached when further increase in velocity becomes unproductive because the fan horsepower increases as the cube of the velocity increases. As can be seen from the curves above that the net capacity increase is negative after certain point.

3.Tube Diameter

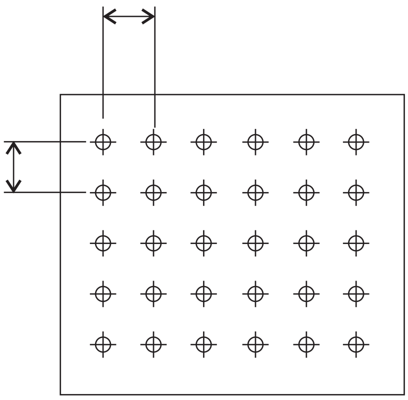


Fig 13

- More compact tube arrangement
- Large tube diameter
- Lower secondary to primary area
- Higher pressure drop
- More fan power

More compact the fin tube arrangement or larger the tube diameter for a given spacing relationship, the more efficient coil would be in terms of kW/m^2 This means lower the secondary to primary area ratio. However more compact the coil, the higher the air side pressure drop, resulting in more fan horsepower to deliver the same air quantity.

Increasing the distances between the tubes will reduce the air side pressure drop, but the surface effectiveness also will decrease. More surface then needs to be added to maintain the same performance for a given air quantity. This leads to a conclusion that for a specific tube size, there is an optimal spacing relationship for cost and horsepower per kW capacity.

4. Cold storage coil performance

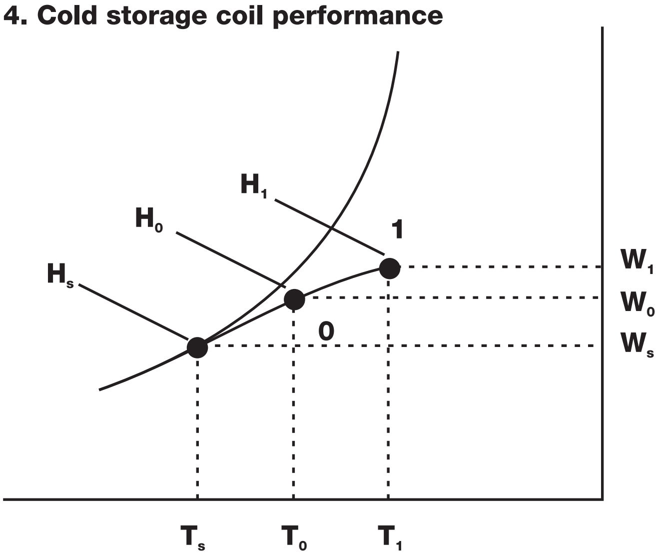


Fig 14

- Drop in temperature
- Drop in moisture
- Drop in enthalpy

This curve shows process of cooling and dehumidification. Air flowing across the coil surface changes condition from I to O and during that process, the enthalpy and moisture content both drop. A reduction in enthalpy from H_1 to H_0 indicates that the air gives up heat which is absorbed by the coil and carried away by refrigerant. The drop in moisture content from W_1 to W_o shows up as condensation over the cooling coil in the form of water or frost. The reduction in temperature is indicated by T_1 to T_o with coil wetted surface, the driving force is indicated by T_s The predominant criterion for evaluating the performance of an evaporator is the condition leaving the coil. The evaporator must hold certain temperature in the space, which it does by removing the proper amount of heat from the air passing through the coil, as indicated by the outlet temperature.

The amount of moisture being removed is also critical, In some situations, such as fruit/vegetable storage rooms, as little moisture as possible should be removed so as not to pull moisture from the product. On the other hand, the coils at the loading dock of a frozen food warehouse should remove as much moisture as possible to reduce the amount of water vapour carried into the low temperature space by infiltrating air.

5.Effect of increasing Face area

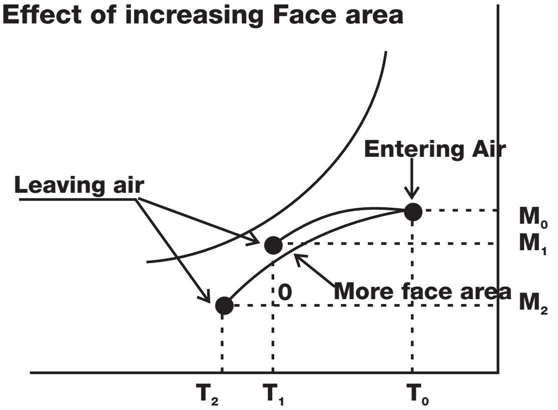


Fig 15

- Lower leaving temperature
- Lower moisture content
- Increased refrigeration capacity

As can be seen from fig 5 , increased face area provides more heat transfer surface. Looking at the coil condition curves for two similar coils, the one with more face area assumes a lower position on the chart. This indicates that the leaving air (T_2) and moisture content (M_2) drop further than T_1 and M_1 achieved by the coil with smaller face area. More face area thus increases the refrigeration capacity as well.

6.Effect of increasing Number of Rows

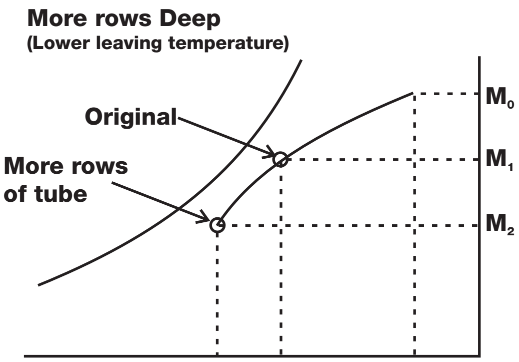


Fig 16

- Lower leaving temperature
- Lower moisture content
- Increased refrigeration capacity

More rows deep provides more heat transfer surface and lowers the leaving air temperature (T2) and moisture content (M2) by moving along the same coil condition curve. Refrigeration capacity also increases

7.Coil Condition Curve

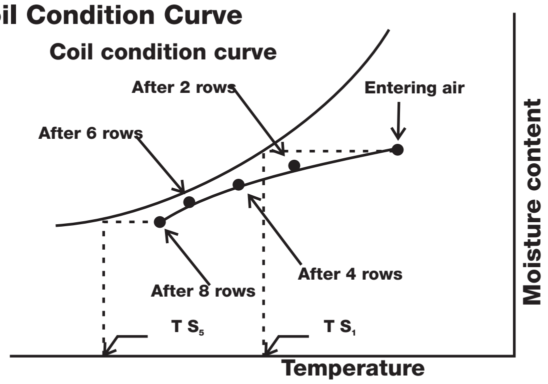


Fig 17

- Each extra row does less work
- Curve steeper as air passes
- Ratio of moisture removal to temperature drop greater at outlet

Looking at the coil condition curve and condition of air as it passes through coil that is eight rows deep, it is apparent that each succeeding row of the tubes does less work in dropping temperature and removing moisture from the air. The greatest rate of heat transfer is where air enters the coil, because it is here that the air temperature and the moisture content are highest.

Another observation is that the curve becomes steeper as the air passes through the coil, which indicates that the ratio of moisture removal to temperature drop is greatest at the outlet of coil.

8.Effect of Increasing Fin Density

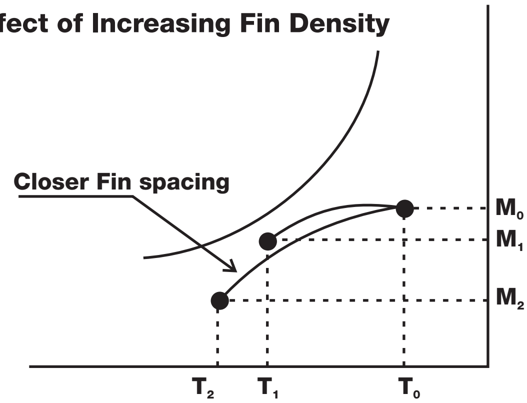


Fig 18

- More heat transfer surface
- Lower leaving temperature
- Lower moisture content
- Increased refrigeration capacity
- Higher pressure drop

Closer fin spacing lowers the coil condition curve, similar to what is observed when face area is increased. There is more heat transfer area, resulting in lower air temperature and reduced moisture content. Refrigeration capacity increases as also resistance to air flow increases increasing fan power or decreasing in air quantity. There is therefore practical limit to how much gain can be achieved by adding more fins.

9.Effect of Increasing Air Quantity

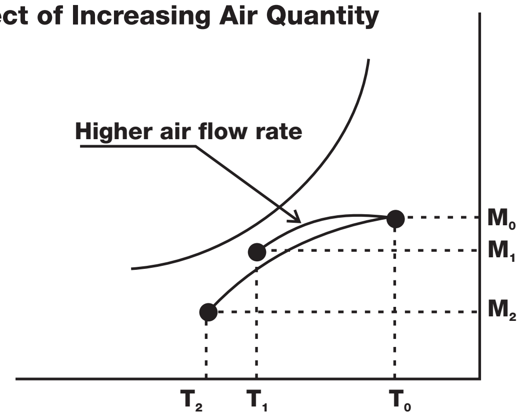


Fig 19

- Leaving air temperature high
- Moisture content high
- % Air flow increase & % decrease in enthalpy
- Higher capacity

Higher flow rate raises the coil condition curve. The leaving temperature and moisture content are not reduced as much as with lower air flow rate. But, because the air flow rate increases by greater percentage than decrease in enthalpy, the refrigeration capacity actually increases

10.Effect of increasing Evaporating Temperature

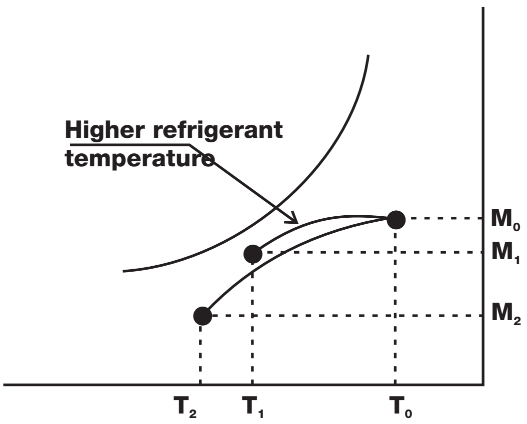


Fig 20

- Refrigeration capacity less
- Leaving air temperature high
- Moisture content high
- Ration of moisture removal to temperature drop less

Higher refrigerant temperature raises the coil condition curve by raising the temperature of coil surface. Neither the air leaving nor the moisture content is reduced as much as with a lower coil temperature. Refrigeration capacity decreases. Another important observation is the ratio of moisture removal to temperature drop is not as great with higher refrigerant temperature.

Summary: Figures 11 to 20 are based on only changing one parameter at a time. System designers may change two or more parameters to obtain desired air outlet properties.

For example available space does not permit the coil face area desired , then one may achieve the closely matching similar effect by selecting deeper coil combined with higher air flow

Air side fundamentals and guidelines

AIR SIDE

Some Important Definitions:

Moist Air:

Completely dry air never found in earth's atmosphere. Air within the troposphere always contains a variable quantity of water vapour, (the gas phase of H2O). Normally the dew point temperature at sea level altitude is around 28^{0}C which means equivalent to 0.025kg of water vapour per kg of dry air. Thus it can be seen that air always contains some water vapour. The earth's atmosphere therefore in addition to dry air contains water vapour which is usually in the form of superheated steam at a low partial pressure & temperature & called as moist air.

Super Heated vapour in Atmosphere

Atmospheric air is a mechanical mixture of dry air & water vapour. Amount of water vapour varies from zero (dry air) to maximum that air can hold i.e. up to saturation point and it depends on pressure/ temperature., The actual temperature of air/vapour mixture is higher than saturation vapour pressure, therefore the water vapour exists in the super heated condition and the air is called unsaturated air. Superheated means at a given vapour pressure the actual temperature of vapour is higher than corresponding saturation temperature.

Dry bulb temperature:

Since air is a mixture of dry air and water vapor, it follows that dry bulb temperature is the temperature of not only dry air component but also the temperature of water vapour component. Hence it is the temperature of moist air in equilibrium. It is the temperature of air as registered by the ordinary thermometer whose bulb is dry and not subjected to radiation or condensation/evaporation of moisture.

Wet bulb temperature: - It is the temperature measured with a thermometer whose bulb is covered with wetted cotton; the reading is stabilized in air stream. In order to achieve stabilized condition quickly the thermometer is moved in air at a velocity of 3 to 5 m/s. Because of evaporative cooling effect, the temperature measured with a wet bulb thermometer is lower than dry bulb except when air is saturated.

Dew point temperature- it is the temperature at which condensation of moisture begins when air is cooled. Since the temperature of saturated vapour is dependent only on the absolute pressure, the dew point temperature is simply the saturation temperature corresponding to the partial pressure of the vapour in air –vapour mixture.

Humidity Ratio – It is the ratio of mass of water vapour to the mass of dry air in grains or pounds- per pound of dry air or (gmsw/kgda.)= mw/mda

Relative Humidity(ø)- It is the ratio of actual mass of water vapour in a given volume of moist air to the mass of water vapour in the same volume of saturated air at the same temperature and pressure. It is the ratio of actual water vapour pressure of air to the saturated water vapour pressure of the air at the same dry bulb temperature.

Or simply speaking it is the ability of the air to hold additional moisture at the same dry bulb temperature. It is the dimensionless value.

The relative humidity can be misleading indicator of the mass of water vapour in a given volume or space of air. The maximum mass of water vapour approximately doubles for each 10^{0}C rise in temperature. Therefore, volume saturated with water vapour at a warmer temperature contains significantly more water vapour than an equal volume saturated with water vapour at colder temperature. This leads to confusion to a lay man because winter RH is often more than summer RH.

	90% R.H.	At 5^0C has 0.6 g/kg moisture
Whereas	90% R.H.	At 15^0C has 1.2 g/kg moisture, twice the amount.

Similarly for the same moisture content in g/kg_{da} lower the dry bulb temperature means more RH and higher the temperature means lower is the RH
e.g. At 5^0C with 0.6 g/kg_{da} moisture has 90% R.H. Whereas with same moisture content of 0.6 g/kg_{da} at 30^0C has R.H. is only 20%

Temperature ^0C	Relative Humidity-%	Absolute Humidity-gm/kg of dry air	Partial pressure of water vapour-Pa
40	25	11.59	13.86
5	85	4.6	5.56
-20	95	0.6	0.73
-40	95	0.075	0.019

In cold storages which operate at much lower temperatures of -20^0C or blast freezers operating at -40^0C, there is even a much bigger difference in vapour pressures as can be seen from the table, which becomes a driving force for water vapour to enter the cold storage.

RH therefore gives misleading information as to how much moisture is present in the air. As can be seen from above higher RH does not mean more moisture content, but only gives us indication that higher RH means air has less ability to absorb more moisture at the same dry bulb temperature.

We should therefore always need to look at absolute humidity values and not relative humidity if we want to know the actual moisture content in the air to decide on which side the vapour barrier should be installed.

Dew point temperature is therefore more intuitive description of water vapour component. The relative humidity varies throughout the day from high in the early morning to low in the afternoon while the dew point temperature tends to be nearly constant throughout the 24 hour period unless rain occurs.

Also the dew point temperature is always uniform throughout the room. On the other hand relative humidity varies throughout the room depending upon temperatures at different locations or the heat generating sources at various locations in the room. The relative humidity may be 50% in the center of room but may be even 90% at the carpet level or higher and at roof level it could be as low as 20 to 30%. Hence stating relative humidity alone without mentioning corresponding dry bulb temperature is meaningless.

Apparatus dew point. It is the average coil's surface temperature or in coil design and selection we call it as Saturated Evaporating Temperature (SET).

Sling Psychrometer
A device called the sling psychrometer is used for convenience & gives reasonably accurate results. It consists of two thermometers mounted in a frame & attached to a handle by means of swivel, one thermometer exposed to air and second wetted with cotton wick wrapped around its mercury bulb.

Air Changes per hour: Air flow in volume units per hour divided by the space volume in identical units-Normally expressed as ACPH or ACH. It indicates how many times the cold storage air passes through the cooling coil in one hour.
Having covered the coil selection details we shall now look at how to determine quantity of air required to be circulated to meet the required room conditions.

AIR SIDE CONSIDERATIONS:
It is experienced by most of the practicing engineers that a well designed refrigeration system, with poor air distribution cannot satisfy design conditions whereas an excellent air distribution can compensate to a very large extent, minor deficiencies in plant capacities.

The primary objective in designing the air distribution system can therefore be stated as-Proper combination of air temperature/humidity and motion to assure designed conditions are maintained.

In order to maintain designed conditions in the cold room it is the air circulating through the room which removes heat and moisture. It is therefore important to consider how much air should be circulated and at what temperature and humidity conditions so that the designed conditions are achieved.

The total heat removal capacity of the cooler depends on adequate air flow to ensure rapid pull down temperature and uniform air distribution to achieve uniform temperatures in all the areas of the cold room.

Too little air would lead to wide temperature differences between one area to another, heavier frosting, and poor air movement in store leading to hot spots and spoilage of product, and too long a cooling period.

Too high air flow is also objectionable as it would lead to excessive noise and high power consumption, imposing additional load on refrigeration system.

Many times air coolers are selected on the basis of calculated refrigeration capacity and then from the manufacturers catalogue the quantity of air is automatically provide. In low temperature applications where the product is loaded in pre cooled condition there is hardly any product load and the refrigeration unit capacity is based only on infiltration, transmission and equipment load. The cooler capacity is therefore very small and accordingly the cfm of the fan is also less. If the room volume is sufficiently large then the air quantity provided becomes in adequate and the uniform room temperature maintenance becomes a problem. Such complaints are very common in practice.

It is therefore important to provide adequate air quantity and circulation to maintain uniform room temperatures throughout the space.

ASHRAE Refrigeration volume indicates following guidelines for velocities over the coil and air changes desired.

Low velocity coolers, low fin density	Velocity over the coil face 0.5 - 1.0 m/s	High humidity application, meat, floral walk in coolers
Medium velocity coolers	1.0 - 2.0 m/s	Vegetable preparation rooms, Wrapped fresh meat cooling, Dairy coolers
Standard air velocity	2.8 - 3.3 m/s over the coil face	Potato coolers, chili stores
High velocity coolers	3.0 - 4.5 m/s over coil but 10.0 m/s at air outlet	Blast and tunnel freezers, where products are not likely to be adversely affected by moderate dehydration during rapid cooling

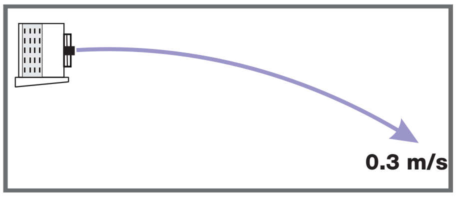
In order to facilitate selection of air quantities following are the general guide lines for air changes.

Type Of Application	Recommended number of air changes MINIMUM	MAXIMUM
Holding Freezer	40	80
Packaged holding cooler	40	80
Cutting rooms	20	30
Meat chill room	80	120
Boxed banana ripening	120	200
Vegetable and fruit storage	30	60
Blast freezer	150	300
Work areas	20	30
Unpacked meat storage	30	60

Since this air coming out of duct outlet enters the space or free zone, it forms an expanding cone.

Air Throw: As the air coming out of air cooler enters the space it forms the expanding cone. As the cone spreads, the air velocity drops down. Air Throw is a distance to which air travels till its velocity drops to about 0.3m/s

How is Air Throw defined



Most sources recommend a theoretical air speed of 0.3 m/s at the end of the room. This is pure theory, however, because any airflow of less than 0.5 m/s is dispersed and becomes impractical to measure.

Fig 21

SPREAD- Spread is the angle of divergence of an air stream after it leaves the outlet. The spread is both in horizontal and vertical direction.

The throw is directly proportional to air outlet velocity. The amount of secondary air getting inducted also depends upon air velocity. Higher the velocity, longer is the throw/blow higher is the induction.

The temperature of air is lowest at the outlet and as the air travels, while the velocity is reducing; its temperature is gradually increasing since the secondary room air is mixing with the primary stream. During this process the air is absorbing the heat from the space to be cooled.

At the extreme end which means at the end of the throw, the temperature of air is more or less same as room temperature.

In the in-between region, where the velocity of air is 0.8 m/s or higher, the density difference between primary cold air and relatively warm secondary air does not have much impact on the shape and direction of the envelop and the flow of air in the room is normally along the axis.

Beyond this point, the temperature difference, which means density difference, affects the direction and the shape of the envelope as well as throw/blow.

If supply air temperature and room temperature difference is high, the air envelope has drooping tendency, since cold air would tend to rush to floor level due to it's higher density

Until now we discussed air stream behaviour in free blow applications. It is necessary to remember that air is a fluid and therefore invariably follows properties of fluids. It means the air has a tendency to cling to the surfaces and travel along. The jet of air directed towards the ceiling will hug the surface and travel along the ceiling.

Contrary to normal belief this results in longer throw compared to free delivery projection, it is therefore always a good practice to direct the air stream by installing the coolers as close to ceiling as recommended by cooler manufacturer to get longer air travel.

This also helps in pushing the air envelop near the surface & improving overall air distribution.

The psychrometric properties and fan laws play important role and we need to understand the basics of them

External pressure and air volume

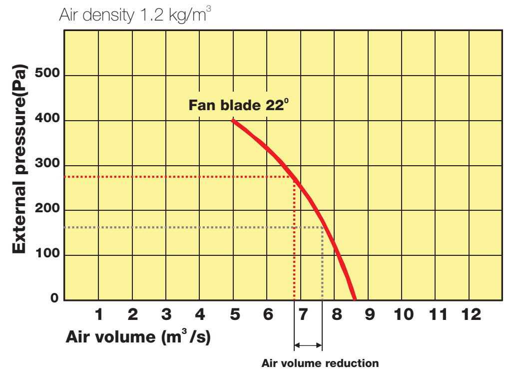


Fig 22

Useful facts: fan laws / air density

■ = constant
■ = change
$\propto$ = changes

Speed Dimension Air density	• volumetric flow rate	$\propto$	Speed
	• Pressure (static, dynamic and total)	$\propto$	Speed ^{2}
	• Required capacity	$\propto$	Speed ^{3}

* Only for geometrically similar ventilators

Speed Dimension* Air density	• volumetric flow rate	$\propto$	Fan blade^3
	• Pressure (static, dynam. and total)	$\propto$	Fan blade^2
	• Required capacity	$\propto$	Fan blade^5

Speed Dimension Air density	• volumetric flow rate	=	No change
	• Pressure (static, dynam. and total)	$\propto$	Air density
	• Required capacity	$\propto$	Air density

Fan Types

Propeller fans

- Air is dispersed
- Good value
- High air volume
- Low additional pressure (30 - 120 Pa)

Axial fans

- Air is focused
- More expensive
- High air volume
- High additional pressure (40 - 400 Pa)

Air density

Standard air density = 1.2 kg/m^3
(at 16 °C, 100 kPa barometric pressure, and 65% relative humidity)

Temperature-related changes

$$\text{Current density} = \text{previous density} \times \frac{273 + \text{previous temp. (°C)}}{273 + \text{current temp. (°C)}} \ [\text{kg/m}^3]$$

Changes related to the installation height

$$\text{Current density} = \text{previous density} \times \frac{288 - 0.00649\ H\ (\text{m})}{288} \times 4.256 \ [\text{kg/m}^3]$$

(H = Height in m above sea level)

We shall now look at some important psychrometrics equations which help us to determine air quantity required to be circulated

Following equations are useful to determine air flow

1. **Sensible heat capacity in W =** J/s = m x cp x ΔT
   W = 1.204 kg/m3x l/s x 1.0216 kJ/kg.k x ΔT k
   W = 1.23 x L/s x ΔT

2. **Latent heat capacity in W=** J/s = 1.204 kg/m^3 x L/s x 2500 kJ/kg x ΔW gm/kg_{da} = 3.010 x L/s x ΔW
   W = 3 x L/s x ΔW

3. **Total cooling capacity of coil in W =** J/s =1.204 kg/m^3 x L/s x ΔH kJ/kg
   W = 1.2 x L/s x ΔH

(Air density =1.204Kg/m3)
(Specific heat =1.0216 KJ/kg.K)
Latent heat =2500 kJ/kg
q_s= Quantity of air in L/s
Qs= Sensible cooling capacity of coil

Considering equation 1 the quantity of air 'q' in L/s = Qs/1.23 x ΔT
Example =
Let us assume sensible load as 6800W and latent load as 1500W

The SHR would be = 6800/(6800 +1500) = 0.82

If we consider ΔT as 10K for 0.82 SHR then quantity of air required to be circulated would be

q_s = 6800/1.23 x 10 = 552.84 L/s
The air quantity can be calculated using any one of the above equations

The fan laws allow us to decide whether air quantity provided is able to maintain uniform temperature throughout the room without causing undue surface drying or unattended hot spots in the room

Defrosting Methods

The evaporator being the coldest surface in the cold room attracts moisture from the air. This moisture condenses on the evaporator surface, and when the surface temperature is below 00C frost is formed. Although light frost accumulation slightly improves heat transfer of the coil (ASHRAE Refrigeration handbook 2010 page 14.4) Continuous accumulation of this frost is not removed, the performance of the evaporator deteriorates since the frost acts as resistance to heat flow as also increases the air side resistance reducing the air flow. If the frost is not removed in time and plant is allowed to operate, the evaporator may become totally ineffective, as there will be no air flow or heat transfer and un-evaporated liquid ammonia coming back is likely to damage compressor. It is therefore essential to defrost the coolers in time to maintain efficiency levels and avoid damage to components. Improper and incomplete defrosting can damage compressor and evaporator coil to the extent that irreparable refrigerant leaks develop when ice is allowed to build up and crush one or more coil tubes. The fan blades are also likely to be damaged if ice builds up on the fan ring. The drain pan gets totally blocked by ice slab and water spills over to floor and ice is formed on the floor as well.

Defrosting is therefore necessary but not in excess also. Defrosting is doubly expensive procedure because energy is used to pump heat into cooler and its surroundings, after which further energy is used to extract the heat from the cooler and its surroundings before the system gets back to its operating temperature. The energy is thus consumed twice, once for forming ice and second time for melting ice.

Defrosting as the name suggests should be activated when frost is formed and not wait till ice is formed on the coil surface. The total energy required to form the ice and defrost it again is estimated to be nearly 1.5 kW/kg of ice (IIAR condenser magazine May 2010 issue). One can thus estimate, based on condensate water amount collected as to how much extra energy and money one is spending. There are various methods of defrosting the coolers and these are described below with their advantages and disadvantages as well as which method is more suitable for the application under consideration.

Air Defrost:

1. Off cycle defrost: The cold stores operating above $2^{0}C$, the evaporator coils can be defrosted by simply turning off the refrigerant flow to the evaporator while maintaining fans running and allowing room air to pass over the evaporator, thus melting of frost. The disadvantage of this process is it is very slow; however it is of lowest cost and requires no additional controls or energy. This method also does not help in removing accumulated oil in the cooler.

2. Warm outside air: Outside warm air can be ducted inside to defrost the coil. This method can be adapted to any temperature in the cold room. It requires ducting and personnel to carry out this defrosting. In colder climates this method is either ineffective or less efficient. The outside air brings moisture and additional heat load on the system.

3. Electric Defrost: This is one of the popular methods for small size air coolers particularly for HFC/HCFC refrigerants. The method can be applied to any cold room application operating at any temperature. While manufacturing and assembling of coils; the dummy tubes are inserted in the coil blocks in a particular pattern and these tubes contain electrical heating elements.

In some designs the heating elements are strapped to the outside of the fin/tube assembly. The advantage of this method is the manufacturer does not have to provide extra dummy tubes.

The advantage of electric defrost is it does not interfere with the refrigerant circuit, and chances of liquid coming to compressor or hydraulic hammer are eliminated.

The system is low in initial cost but high in running cost since it consumes lot of electrical energy, also it does not help in oil removal from evaporator. In general for a cooler of 30 to 40 kW capacity one requires heaters of nearly 18 kW including drain pan heating and if defrost is done 4 times a day for 30 minutes each cycle then the cost of defrosting alone is 36kWhxRs.5.0/kWh = Rs.180 per day or Rs. 5400 per cooler per month. High maintenance due to frequent failure of resistance heating elements & replacement of burnt heaters is also a tedious job.

4. Water Defrost: The second most popular method of defrosting air coolers is spraying water on the coil. The mixture of water and melted frost collects in the drain pan and taken outside the refrigerated space. The advantages of water defrost over other methods are

1. Inexpensive source of defrost medium
2. Short defrost time say 30 to 45 minutes
3. Provides automatic cleaning action of coil
4. Water defrost is most advantageous when there is only one or two coolers and out of which one needs to be defrosted. In such cases enough hot gas is not available to defrost and the hot gas defrost becomes ineffective
5. Normally spiral freezers or blast freezers prefer this method since these are many times single compressor and single cooler units.
6. Water defrosting provides rapid defrosting of coils for virtually all room temperatures. Water is sprayed over the coil and the mixture of water and melted frost flows in the drain pan. The normal water temperature should be around 16 to 18^{o} C or more depending upon wet bulb temperature in the area and flow to be 1 to 3 litres per second per square meter of coil face.
7. This method is less desirable when temperatures decrease below freezing; however it can be successfully used in many applications for temperatures as low as -40^{o} C.
8. The water used for defrosting needs to be with neutral PH value so that it does not damage fins and filtered so as to prevent choking of spray nozzles.
9. The quantity of water sprayed and the velocity needs to be controlled to ensure that water droplets are not carried in the air stream and into cold room.
10. A warm water from heat reclaim unit can also be used for defrost purpose.

5. Brine Defrost: In case of coils using brines instead refrigerant, the coils can defrosted by remotely heating brine for the defrost cycle. This system is effective since it provides heat from inside and is therefore as rapid as hot gas-defrost. The heat source for brine could be steam, electricity or condenser water.

6. Reverse Cycle Defrost: This defrosting method is used in air cooled applications where condenser and evaporator both work on air as cooling medium. The ideal defrosting should terminate the defrost cycle when the whole cooler is sufficiently warmed above the melting point of ice to ensure the cooler is dry and frost free. This is done most easily in reversed cycle defrosting system where the pressure within cooler gradually rises till the frost disappears and then the defrost cycle may be terminated. The reverse cycle defrost is very efficient but seldom used since the very reliable four way reversing valve is required in the refrigeration circuit. Also this system is used where single cooler and single compressor are working in a system. Rotation of the four-way valve through 900routes the hot gas to cooler instead to the condenser. When multiple coolers working on single compressor in the system this system cannot be used for the obvious reason that all coolers cannot be defrosted at a time by reversing the refrigeration cycle. This system is popular in truck/container refrigeration units

7. Hot Gas Defrost: It is necessary to thoroughly understand details of working of this system before using the same

1. Hot gas defrost is the best and most efficient alternative as heat source acts from within whereas water/electrical defrost heat source is from outside.
2. During hot gas defrost cycle, evaporator acts as condenser giving up the heat and converting gas to liquid.
3. Although hot gas defrost is the most effective way of defrosting, it is equally complicated, troublesome & may be inefficient if not properly designed.
4. The basic procedure in hot gas defrost method is to interrupt the supply of liquid refrigerant to evaporator, pump out the liquid to empty the evaporator, restrict the liquid outlet by closing the valve, supply hot gas at high pressure either from compressor discharge or from high pressure receiver to warm the evaporator coil &melt surface frost/ice formation.
5. During the operation the heat from hot gas is absorbed by the metal in the coil/plate and its temperature rises. Once the temperature is high enough, ice/frost on the surface melts and is drained off.
6. Out of the total heat supplied by the hot gas, nearly 50% is used for heating the metal and balance 50% or even more is lost to space surrounding tubes/plates since the temperature of surrounding air is much lower than temperature of the unit.
7. Typical freezer coils have internal volume between 4-6 litres/kW. A coil of 35 KW will have approximately 27 to 50 kg of ammonia liquid and with the initial boil off rate of approximately 1.2kg/min it will take about 20 to 40 minutes to boil out all the liquid from the freezer.
8. Lower the temperature/pressure of hot gas supply,

lower would be the loss to space.

9. If the temperature of hot gas is too high, the tendency of coil is to steam. Also as the air temperature goes up, its relative humidity drops. This leads to increased evaporation of surface water. It also adds to refrigeration load if it is a cold storage or if the freezers are in the open area then it leads to fog/mist formation.
10. Warmer temperatures will not necessarily improve defrost efficiency. This is because most of the defrost heat comes from latent heat of hot gas, rather than sensible heat. Following table for ammonia refrigerant will make the matter more clearer:

Temperature (Deg C)	Pressure (Bar)	Latent heat (KJ/Kg)
4^0C	4	1240
10^0C	5	1220
16^0C	6	1200
21^0C	8	1180

From the above it can be seen that 21^0C defrost temperature would actually require 5% (1240-1180)/1180, more hot gas than 4^0C to provide the same latent heat content.

11. At lower defrost pressures the defrosting takes slightly longer time say around 20 to 30 minutes. However with slightly extended defrost times at lower temperature, the overall defrost efficiency is much better than at higher temperature/pressures due to reduction of refrigeration requirements.
12. A pressure regulator in the plant room is therefore required to be installed on the hot gas defrost pipe, set at 7.0 barg max outlet pressure. Another advantage of this lower pressure is less liquid would condense in hot gas line as the condensing temperature is reduced between 11 to 16^0 C. It is also recommended to have this valve with electric shutoff feature. When no coils are calling for hot gas flow, this regulator will be closed, minimizing the ammonia condensate formed in hot gas supply header.
13. Also having higher pressure in the evaporator if warmer water is used means slowing down the flow of hot gas as the pressure difference between hot gas supply pressure and evaporator pressure reduces, since pressure difference is the driving force which allows the hot gas to flow
14. It is also necessary to Keep the defrost gas mains free of liquid-A condensate drainer needs to be installed to drain trapped condensed liquid in the hot gas defrost line. The hot gas tends to continuously condense during cold climate conditions if the pipe is running outside the building or in the cold space in processing areas. The liquid formed must be drained to low pressure liquid line or vessel. The defrost relief regulator setting or OFV setting should be around 5.0barg.
15. It is most important to remember that at the most only 1/3 of all evaporators/freezers can be defrosted at a time to ensure availability of adequate hot gas for defrost generated due to load on other 2/3 working coolers/freezers. If only one or two coolers are operating and if one of it needs defrost then hot gas defrost system will not work as not enough hot gas would be available for defrost.
16. This means if the system has 6 freezers each of 70 kW capacity then total load, when all freezers are operating is 420 kW. In such condition only maximum two coolers can be defrosted at a time.
17. Hot gas pipe line sizing should be done to 3 times the working capacity. It means for installation having 3 nos. 70 kW coolers each, the hot gas defrost line should be sized for 70x3=210 kW (50mm) and the main header from machine room to the production area should be sized for 2x70x3= 420 kW (80mm)
18. Defrost condensate return line from freezer should be sized one size higher than liquid supply line as this condensate line may contain return hot gas in addition to condensed liquid.
19. The critical periods during defrost is at its initiation and at its termination. In both the situations high pressure vapours moving at considerable speed come in contact with cold liquid causing pressure shock waves. One stream is nearly at 7.0 barg, whereas other side is at nearly at atmospheric pressure or below it.
20. To prevent this soft hot gas system is adopted for coolers/freezers of larger than 50 kW capacity, which has two solenoid valves, the smaller one opens first reducing pressure gradually in the coil before returning to refrigeration operation. This is pressure sensing operation either through microprocessor or with set pressure device or electronic adjustable timer. Similarly at the initiation of defrost cycle, two solenoid valves are used, the smaller one opens first thus gradually increasing pressure in the coil before the second bigger valve is opened. Refer ASHRAE Refrigeration volume 2010 page 2.26
21. The soft hot gas defrost system is designed to gradually increase the coil pressure as the defrost cycle is initiated. Sometimes this is done by using small hot gas feed with 25 to 30% of the duty with solenoid valve and hand expansion valve adjusted to bring pressure up to 2 to 2.5barg within 3 to 5 minutes, before the main defrost valve opens. Now a day this is done with help of two step valves which perform the above function. The use of these valves avoids extra solenoid valves, expansion valves and extra piping and timers.
22. Similarly once the defrost period is completed a small

suction line solenoid valve is opened so that coil/plates can be gradually brought down to operating pressures before full liquid is admitted.

23. A manual initiation of defrost for larger coils/freezers is recommended based on physical condition of freezer with respect to amount of ice/frost formed on the surface.
24. It is recommended that evaporator defrost liquid should be returned to intermediate pressure vessel and not to low pressure vessel in case of two stage system. This has two advantages. Firstly it does not disturb the LP vessel pressure /temperature conditions during defrost operation and thus other operating coolers work without any disturbance. Secondly defrost liquid pressure and intermediate vessel pressure difference is much lower than defrost pressure and L.P. vessel pressure and thus saves considerable energy.
25. the non-return valve in main supply line after solenoid valve is essential to ensure that high pressure developed during defrost in the coil does not exert back pressure at the outlet of solenoid valve , since the inlet pressure is normally either equal to evaporator pressure or pump discharge pressure.
26. Advantage of regulating pressure to 7.0 barg in the equipment room is, there is less chance of coils getting damaged or bent or ruptured as low side of the system is normally designed for 10.0 barg and hot gas at 8.0 barg or more pressure is dangerous to the low side parts of the system. To be safer it is recommended that low side should be also designed for 20 barg same as high side design pressure for condensers/receivers and pressure tested to 1.25 time's pneumatic pressure.
27. Example- a 70kW coil defrosting for 12 minutes will condense up to 11kg/min of ammonia of totalling to132 kg. The enthalpy difference between returning low stage -40^{0} C and intermediate vessel at-7^{0} C is (172.34-19.17)=153.17 kJ/kg i.e. 153.17x132=20218.44 kJ=5.61 kW or in 12 minutes 5.61x60/12=28 kW removed from the -40^{0}C booster compressor for 12 minutes during each defrost.
28. An excessive noise and shock or vibrations observed during defrost is not normal and if observed cause must be corrected.
29. Many times hot gas is taken from top of receiver instead from compressor discharge, to ensure adequate amount of hot gas availability.

Defrost Control Initiation Alternatives:

Defrost initiation can be done once the frequency and duration of defrost cycle is established. Control schemes are generally implemented by means of an electric or electronic timer, or a computer based control logic.

1. **Mechanical Timer clock:** This allows coolers to be defrosted at fixed but adjustable set intervals. The advantage is the operator does not have to remember when to defrost. The disadvantage is even when coil does not need defrosting the timer would automatically activate defrost cycle. Many times when the products are freshly loaded the required defrosting frequency is more and once the products are kept at desired temperatures, the defrosting frequency is less. This requires re adjusting timer settings.
2. **Microprocessor controllers:** They have mostly replaced mechanical timer clocks. The use of microprocessors reduces energy consumption and helps in maintaining product quality.
3. **Ice thickness sensor:** An ice thickness sensor is attached to the coil and when thickness builds to a particular size it touches the sensor and activates defrost cycle. This method is used many times in ice bank/ice reserve units for dairies.
4. **Ait Temperature difference:** Sensors are provided at coil inlet air path and outlet path and defrost controller is set at particular T. If the coil gets frosted this temperature difference reduces and the defrost sequence is activated.
5. **Air Pressure differential controls:** Instead of temperature difference, one can use a differential pressure monitor/controller. As the frost accumulates the ΔP across the coil increases, activating defrost cycle.
6. **Reverse Cycle defrost;** this system uses four way valve and ant preset intervals the refrigerant flow is reversed so that condenser acts as evaporator and evaporator acts as condenser. These systems are popular in truck refrigeration units using HFC refrigerants.

Many times combinations of hot gas defrost for coil and electric defrost for drain pan is used. It is also essential to provide ring heaters for fans to avoid fan blades getting damaged.

To terminate hot gas if the room temperature tends to rise is also provided which overrides the pressure/temperature differential sensors and activates cooling cycle even when coils are not defrosted fully so that product and room temperatures are maintained within the allowable limits.

A standard soft hot gas defrost cycle diagram is shown below:

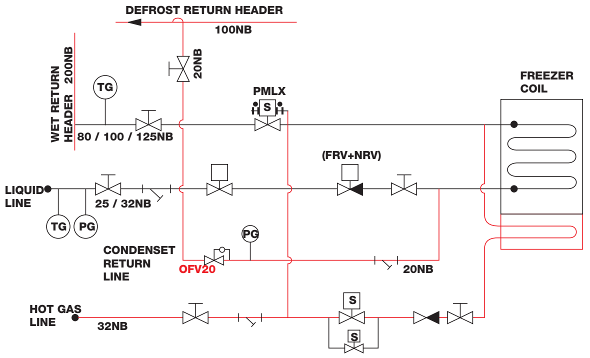


Fig 23

Sequence of Operation of Hot gas Defrost Cycle: Initiation of Defrost

1. Based on the condition of ice formation on the freezer defrost toggle switch provided on the control panel is activated manually. In certain cases the defrost sensors which either sense ice buildup thickness, or preset pressure drop across the coil or fixed timer setting frequency is used to defrost based on demand.

   For Batch load applications like Blast freezer/Plate freezer or Individual Quick Freezer (IQF), the demand defrost method should not be used. The defrost sequence should be initiated manually. A separate Electric switch to manually activate defrost cycle shall be provided. Once the cooling cycle is over and the doors are opened the control can be put on defrost mode. Before the product is reloaded again and doors are closed then the control can be put on cooling mode. The cooling cycle time is variable based on the product to be frozen and hence a manual operation of initiation and termination through control switch has to be carried out.

2. **Closure of Liquid Line Solenoid valve: Liquid Pump down**

   On defrost cycle activation; first the liquid line solenoid valve shall be closed which starts evaporator liquid pump out cycle.

3. **Fan time delay Phase to switch off**

   All the fans at this stage must be running to provide high liquid refrigerant boiling off rate. Fan motor heat also additionally provides quicker boiling off liquid. If the fans are on VFD, then during starting of defrost cycle the fans must be run at full speed. This is to ensure the coil gets empty as quickly as possible. After a time delay of 3-5 minutes the fans are stopped through a preset timer thereby stopping the air circulation. The period required is around 3 to 5 minutes depending on size of evaporator and the internal volume to ensure that entire liquid has been pumped out. During this period the suction line or wet return line valves remain open and pumps out the liquid from evaporator.

4. **Closure of Wet Suction valve**

   After the time delay of 3 to 5 minutes based on adjustable set point timer, the wet Suction line solenoid valve is closed and the fans were switched off thus isolating the cooler from the system.

5. **Supply of Hot gas**

   A. Soft Gas Phase: (For coolers having capacity higher than 50 kW) on low temperature pump recirculation systems, a small solenoid valve should be installed in parallel with the larger hot gas solenoid valve. This smaller valve opens & gradually introduces hot gas in the coil. Opening of this valve first further reduces the likelihood pressure shocks. At the conclusion as per electronic adjustable timer settings this solenoid closes, simultaneously opening main Hot gas solenoid valve, admitting hot gas in the evaporator and warming up the Evaporator surface.

   B. Main solenoid valve in hot gas line then opens by using two solenoid valves thus achieving soft gas defrost for coils above (0.14m3) of internal volume.

   C. During this period the condensate liquid line valve also remains closed so that evaporator has no outlets open and thus allows the coil pressure to build up around 5 Bar as the OFV valve is preset for this pressure.

6. **End of Hot gas Defrost Cycle:**

   A. Once hot gas defrost cycle is completed (normally 5 to 15 minutes) based on the size of the coil), the suction line opens gradually by using a two step Solenoid valve and pressure from freezer is released to wet return line.

   B. The condensate accumulated due to condensation of hot gas is also drained to wet return line as the OFV valve opens at this time. Some systems use condensate float trap also.

   C. There is also an overriding thermostat which terminates the defrost cycle if the room temperature tends to increase beyond acceptable limits.

   D. Liquid line solenoid valve and suction stop valves will now open and would allow liquid refrigerant to evaporator. The amount of liquid admitted is controlled by pre adjusted flow regulating valve cum non return valve or hand expansion valve or motorized valve as the case may be. This initiates cooling operation.

   E. If the duel opening valve has not been installed in wet return line and normal solenoid valve has been provided in line, then similar to liquid line and additional solenoid valve in parallel is required to be installed. This valve opens first and allows the pressure in the coil to reduce slowly. This eliminates system disruptions, which would occur if warm refrigerant were released quickly in to suction piping. This also reduces vapour propelled liquid, and prevents sudden loading of compressor if suction pressure rises quickly.

7. **Fan Delay time:** The fan is not yet energized. Instead, the coil temperature is allowed to drop, freezing any water droplets that might remain on the coil surface after the hot gas defrost phase, thereby preventing the possibility of blowing water droplets off the coil in to refrigerated space.

8. **Start of cooling cycle:** After the fan delay has elapsed, the fan gets energized automatically based on time setting. The refrigeration phase continues until the next defrost cycle is initiated.

The entire process can take maximum 15 to 30 minutes depending on size of evaporator and available quantity of hot gas.

The steps 1 to 8 are all built into the control circuit of the controller. The timings can be adjusted to suit particular evaporator model and size since adjustable electronic timers are provided in the controller.

Trouble-shooting guidelines

Following are the general guidelines to identify problems if air cooler is not giving satisfactory performance While inspecting the air cooler it is necessary that the coil and fins and all other sheet metal components are thoroughly cleaned. If the cooler is for negative temperature application then it must be cleaned of all the ice accumulated.

Ice has much higher density than frost and will require more time to melt than normal frost formation on a coil. If not defrosted in time the ice accumulation on coil can lead to many problems like damage to coil and fins, fans breaking, drain pan chocked with ice etc.

It must be therefore ensured that correct defrost cycle setting and procedure is followed.

Possible Cause	**Corrective Action**
Blown Fuses	Replace fuses, check for short circuit as also overload conditions
Faulty Motors	Replace Motors
Unit is in defrost mode	Wait till completion of defrost cycle
Low Refrigerant Charge	Add refrigerant in system Check piping
Coil iced up	Manually defrost coil and then adjust defrost cycle
High infiltration load	Ensure all openings are properly sealed
Low refrigerant flow through evaporator	Check fan direction of rotation, Check and clean strainers Adjust hand expansion valve/flow regulating valve setting
Defrost taking too long	Adjust defrost settings
Defective defrost timer, thermostat	Replace defective component
Fan delay not set correctly	Reset fan delay duration
Insufficient defrost cycles in 24 hours	Increase defrost frequency
Defrost cycle too short	Adjust setting for increased duration
Fans continue to run in defrost mode	Adjust setting to prevent fans running
Uneven coil frosting	Defective heater elements, Unit located too close to door or opening Refrigerant feed insufficient to properly feed evaporator Defrost duration too short Fans not functioning correctly
Ice accumulation in drain pan	Defective heater elements, Unit incorrectly pitched Condensate drain line plugged, Defective drain line heater Insufficient hot gas, Defective defrost timers, thermostats, regulating valves
Low Air Flow	Coil iced up, Unit mounted too close to wall, VFD setting defective if used Fans not functioning correctly
Insufficient Air Throw	Air discharge area obstructed by products Fit streamers or duct sox on air unit, Keep air return area passage free
Air cooler performance checking	Take air inlet and out let temperatures and evaporator saturation temperature derived from pressure readings, Compare coil TD and TD across the coil Ensure air is not getting short circuited back instead going over the product

Sample project specification sheet

The purchaser/consultant should ask following information if he is obtaining offers from more than one supplier so that correct technical comparison & evaluation is possible

Sr. No.	Specifications	Manufacturer-1	Manufacturer-2
1	Make		
2	Model		
3	Manufacturing Location		
4	Capacity-kW		
5	Type-Blow Through/Draw through		
6	Type-Gravity Feed/Pump Feed		
7	Type-Floor mount/Ceiling or wall mount		
8	Over feed rate		
9	Circulation rate(1+8)		
10	Ammonia evaporating temp. At coil outlet-SET-^{0}C		
11	Air Temperature-Coil in-^{0}C		
12	Air Temperature Coil out-^{0}C		
13	SHR		
14	Effective heat transfer area-m^2		
15	Coil Face area-L x H- mm		
16	Total internal volume-dm^3		
17	Refrigerant side Pr. drop-Pa		
18	Fin spacing-mm		
19	Fin Thickness-mm		
20	Fin Type-plate/corrugated/enhanced surface		
21	Fin material		
22	Tube diameter OD -mm		
23	Tube Thickness- mm		
24	Tube- no of rows deep		
25	Tube -no of rows on face		
26	Tube Type-plain/internally grooved		
27	Tube material		
28	Tube arrangement-Pitch-Triangular/Square		
29	Total Air Volume at cooler outlet-m^3/hr		
30	No of fans		
31	Fan diameter-mm		
32	Fan external static available-Pa		
33	Air Throw-m		
34	Air Side pressure drop-Pa		
35	Fan Speed -RPM		
36	Fan noise level combined		
37	Fan power consumption-W at operating condition		
38	Installed Motor power per fan -kW		
39	Casing Material		
40	Defrost Tray material		
41	End plate material		
42	Defrost arrangement - for coil/for drain pan		
43	Over all Dimensions-L x W x H mm		
44	Liquid inlet -Nos. x diameter OD mm		
45	Out let -Nos. X diameter OD mm		
46	Dry -weight-Kg		

Alfa Laval in brief

Alfa Laval is a leading global provider of specialized products and engineering solutions.

Our equipment, systems and services are dedicated to helping customers to optimize the performance of their processes. Time and time again.

We help our customers to heat, cool, separate and transport products such as oil, water, chemicals, beverages, foodstuffs, starch and pharmaceuticals.

Our worldwide organization works closely with customers in almost 100 countries to help them stay ahead.

How to contact Alfa Laval

Contact details for all countries are continually updated on our web site. Please visit **www.alfalaval.com** to access the information.