1 INTRODUCTION

Elsewhere students have been exposed to the subject of "cold creation" via refrigeration cycles. This course discusses the "use of cold" to obtain and maintain low temperatures for solid products to accomplishing some process need.

The following topics are to be covered:

- Common means of supplying cold
  How do we use the cold created by a refrigeration system to cool solid food products?

- Specifics of cold facility design
  What are the features of refrigerated buildings that are different from other buildings and which are important to ensure efficiency and safety?

- Estimation of heat loads
  How big do we need to make our refrigeration system to ensure that we achieve the product temperature that we require?

- Freezing time prediction
  How long will it take to freeze a solid food product?

- Chilling time prediction
  How long will it take to chill a solid food product?

This course is designed to deliver two outcomes. At the end of the course the students are expected to have:

- Knowledge of refrigerated facilities features and the areas of importance to ensure efficient operation.
- Ability to use tools to estimate:
  - Cooling and freezing times for food products
  - Heat load

2 COMMON COOLING APPLICATIONS

Refrigeration applications in the food and bioprocess industries tend to divide into two groups

- those that seek to minimise product temperature change (cold and cool storage),
- those that seek to promote product temperature change (chilling and freezing),

Cold stores are devices for keeping product at a storage temperature below its freezing point.

Cool stores are for storage of unfrozen, but chilled product. With the exception of respiring fruit and vegetables there is no heat generation within the product itself. There may be some residual heat to be removed if product enters the store improperly frozen or chilled, but this should not occur with good operating practise.

Freezers are devices for reducing product temperature to less than its freezing point (approximately 0°C).

Chillers are devices for reducing product temperature to a temperature above the product freezing point.

Table 1 shows a summary of the above definitions. Stores are characterised by (typically) use of air an intermediary heat transfer medium, where as for temperature reduction air is one of several options.

	Temperature Reduction	**Temperature Control**
Above Freezing	Chiller	Cool Store
Below Freezing	Freezer	Cold Store

Table 1, Summary of Device Definitions

2.1 TEMPERATURE CONTROL SCENARIOS

2.1.1 Cool/Cold Stores

The aim of stores is to keep product at constant temperature. Heat can only enter the product from the air flow (if the air is above the product temperature), and the product will only cool if the air is below the product in temperature. As the product does not generate heat there is no need to provide large air flows over the product. The air flow should be directed in such a way to pick up heat before it reaches the product, and carry it to the evaporator directly. Figure 1 shows a basic store designed to maintain product temperature.

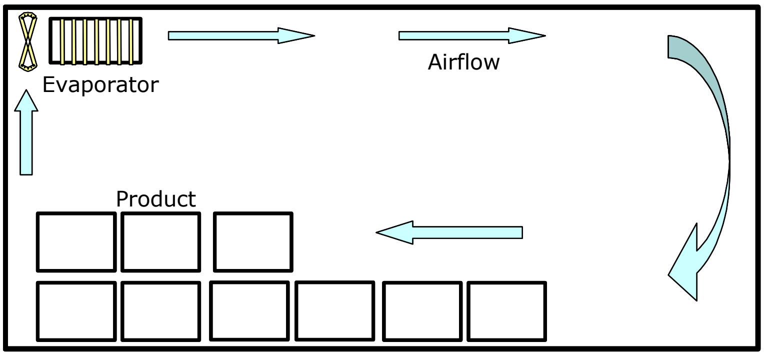


Figure 1, General store layout and airflow pattern

The prime heat flows entering a store are those through the walls, ceiling and floor, and that through doorways. Other sources are lights, people and the fans that provide the air movement.

Product should never be stacked against walls, and must be off the floor. Product should not be located close to the ceiling, and especially kept away from ceiling mounted lights as these give off considerable amounts of heat by radiation. A variety of bins and pallets that can be stacked in multiple layers are available. Figure 2 show examples of product stacked in stores.

Figure 2, Examples of product stacked in stores. (Left: Kiwifruit pallet stacking, Right: Mixed product stacked in racking)

2.1.2 Environmental Control

A number of processing industries require rooms to be air-conditioned during processing. For example:

- in meat boning and cutting rooms.
- packing lines (fruit, icecream, chilled products)
- laboratories (microbial)

The systems used are similar to those used in cool stores. Differences occur in that air flow is kept to a low flow rate and fans are exterior to the room. The evaporator is not directly

visible in the room and the cold air is supplied through ducting systems. This is to ensure the employees comfort, as these types of room are regularly occupied.

Figure 3 shows an example.

Figure 3, A fruit packing line with environmental control.

2.2 REDUCING PRODUCT TEMPERATURE

In reducing product temperature, it is usually aimed to cool the product as rapidly as possible to minimise microbial, physiological or chemical deterioration of the product.

As well as achieving rapid chilling of the product, it is also necessary to achieve even chilling. Products leaving a cooling operation should be all at the same temperature if final product consistency is to be maintained.

The design of a chiller/freezer hence focuses on:

- Increasing heat transfer coefficient to increase rate of cooling
- Lowering coolant temperature to increase rate of cooling
- Achieving cooling/freezing consistency to ensure final product quality consistency

Achieving fast, consistent and cost effective cooling rates can be achieved through the careful choice of cooling medium. A number of different chiller/freezer types exist, characterised by the cooling medium. The choice of cooling medium are:

- Air, delivered through movement by fans past the product.
- Liquid, supplied by either submerging or spraying the product.
- Cryogenic freezers in which the product is submerged in or spayed with liquid nitrogen or carbon dioxide.
- Plate freezers where the product is cooled through direct contact with the refrigeration system evaporator.

These notes comment on important engineering aspects and the advantages and disadvantages of each cooling medium.

2.2.1 Air Blast Chillers/Freezers

Air blast chillers/freezers are the most common method of cooling because air is a free and easy to use cooling medium. Products range from fruit, dairy and meat.

Important considerations with respect to the use of air are its:

- Low density and thermal conductivity, which limit the rate of heat transfer that are achievable.
- Ability to carry considerable amounts of dissolved water that tends to deposit and foul heat exchange surfaces, which then need defrosting.
- Usually drier than the product, resulting is product water/weight loss during cooling.
- Heat transfer coefficients to solids are comparatively low.
- Temperature is limited only by refrigeration system capability.
- Cheap and ubiquitous.
- Allows access to product during cooling.

To increase heat transfer rates the air velocities used are large. Air velocities of 0.5-3.0 m/s are not uncommon. Heat from the fans driving the airflow is significantly large in chiller/freezer applications.

<u>Example 1: Apple Pre-Coolers</u>

One method of cooling apples is to use air blast cooling (known as pre-cooling in the industry). Apples are packed into there final packaging and palletised directly from being picked. The apples are to be stored at 2°C (and in controlled atmospheres) to maximise their shelf life. Small time periods above this temperature can shorten the storage life of the product significantly due to product ripening. Therefore apples are cooled to 2°C before being put into storage in a period of 12 hours.

A fan is used to accelerate the cold air past the product in a cooled room. The basic design is shown in Figure 4 below.

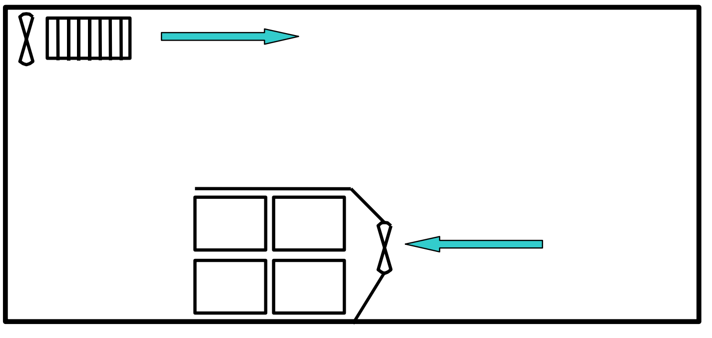


Figure 4, Basic Design of Pre-Cooling Fruit/Vegetable Products

For apples, pallets are stacked in two rows next to each other creating a central plenum. Obvious gaps, like those caused by the pallet, are sealed so that all of the air is forced to pass directly past the product through the packaging, maximising the cooling rate and ensuring good cooling consistency throughout the product. The gaps are sealed with canvas to force the air to pass through the cartons with holes that are designed for the purpose of aiding pre-cooling. Figure 5 and Figure 6 show a photo and drawing of a typical apple pre-cooler.

Figure 5, Photo of a Typical Apple Pre-cooler

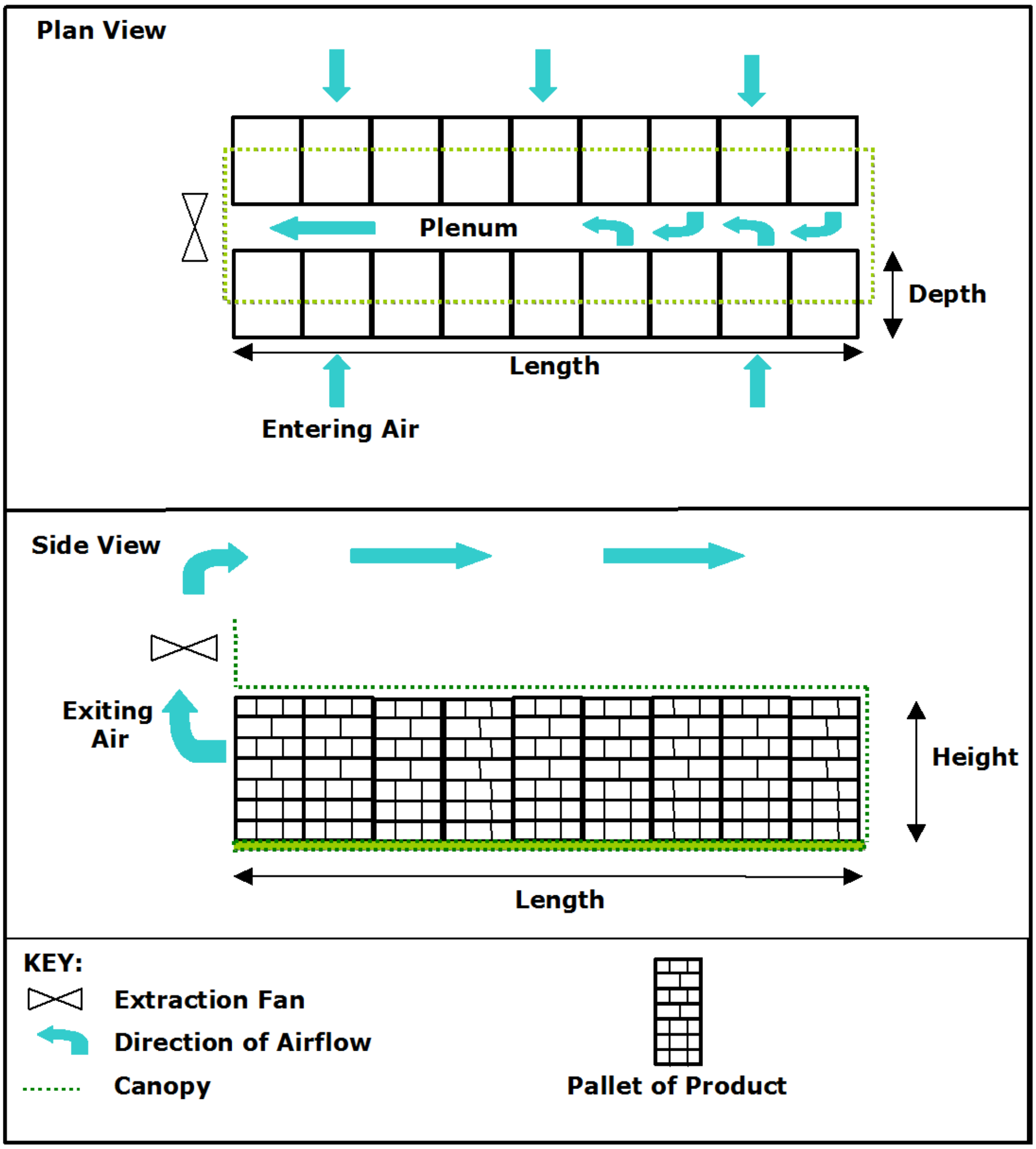


Figure 6, Drawing of Typical Apple Pre-Cooler

2.2.2 Liquid Chillers/Freezers

Liquid chilling/freezing may be accomplished by placing plastic-wrapped product items in either a chilled liquid bath or in a chilled liquid spray. Product is usually directly conveyed through such systems.

Important considerations with respect to the use of liquids are:

- Heat transfer coefficients to solids are comparatively good.
- Temperature is limited by liquid properties.
- Product is required to be packaged, increasing heat load and slowing heat transfer
- Product contamination is possible from the liquid.
- Product required to be dried after process

Water or water/ice slurry systems can be used to chill product to 2°C or greater. The temperature is limited by waters ability to freeze at 0°C. The major advantage of water is that it is cheap, however it does need to be cleaned routinely to minimise the chances of microbial contamination.

Glycol/water or salt/water mixtures could be used to chill product to lower temperatures (approx –15°C). However the product must be washed afterwards, especially in the case of glycol which is toxic. Losses of coolant with the out going product also incur an economic cost, as the glycol or salt will be required to be continually replaced. Figure 7, is a sketch of a typical immersion type liquid chiller.

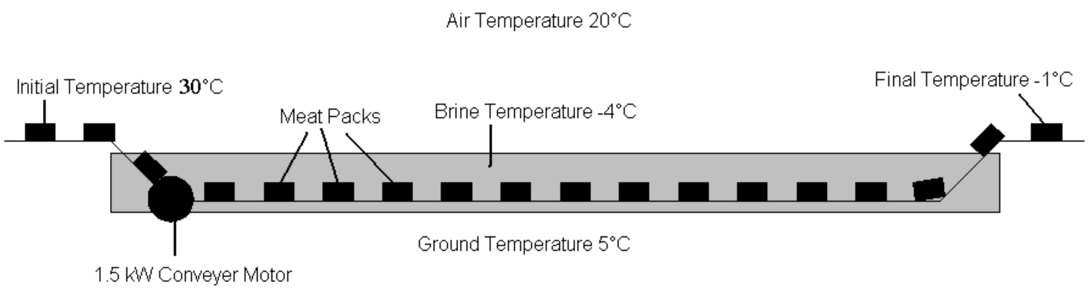


Figure 7, Schematic of Typical Immersion Chiller

2.2.3 Cryogenic Freezing

Cryogenic freezing may be accomplished by spraying or immersing product in either liquid nitrogen or liquid carbon dioxide. As the liquid evaporates, heat for vaporisation is taken from the products, resulting in very rapid freezing.

Important considerations with respect to the use of cryogenics are:

- Heat transfer coefficients to solids are extremely good.
- Temperature is limited by storage pressure, but these are extremely low.
- Minimal capital is required, the only resources required are for high pressure storage and application of cryogenic fluid.
- Cryogenic fluids are very expensive and need continual replacement.
- Ability to match variable throughput, through basic energy balances.

- Simple to operate.

Generally, the cost of the cryogenic fluids is such that cryogenic freezing is not an economical option. However, situations where cryogenics are cost effective include:

- Processing a high value product
- Processing rate is highly variable
- Cooling rates can not be achieved by other means
- Investment in traditional refrigeration cannot yet be justified, (entering a product into the marketplace).

Example 2: Whole Tuna Freezer

Whole frozen fresh Tuna receives a premium price in the Japanese marketplace. Catching rate (and hence processing rate) is variable and the speed of cooling directly relates to the quality of the product. Therefore cryogenic freezing is a viable option for freezing Tuna.

In a batch processing system, fresh Tuna are placed in bins in which liquid nitrogen is sprayed onto them. Figure 8 and Figure 9 show a drawing and photo of the basic cryogenic freezing operation used.

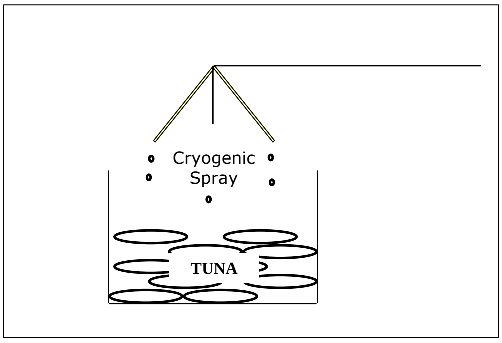


Figure 8, Drawing of Cryogenic Freezing of Tuna

Figure 9, Photo of Cryogenically Frozen Whole Tuna

2.2.4 Plate Freezers

Plate freezers accomplished freezing by directly contacting the product with the evaporator of the refrigeration system. As the refrigerant evaporates, heat for vaporisation is taken from the product via conduction through the plates, resulting in rapid freezing.

Important considerations with respect to the use of plate freezers are:

- Refrigeration distribution within the plates needs to be accurate to ensure even freezing of product.
- Plates need to move and therefore require flexible coupling to carry the refrigerants, increases chances of refrigerant leakage.
- Heat transfer coefficients to solids are very good.
- Temperature is limited by refrigeration system capability.

The plates may be either vertical or horizontal. A horizontal plate freezer is shown in Figure 10.

Figure 10, Photo of horizontal plate freezer

<u>Example 3: Ice Cream on a Stick</u>

The 'ice cream on a stick' freezer is a very specialist form of a plate freezer. Along with its freezing capability (refrigeration evaporator) it is also a conveyor and provides the mould shape for the product.

Liquid ice cream mix is poured into the product shape moulds, (Figure 11). The ice cream travels for a short time in the conveyor (also the evaporator), simultaneously cooling and subsequently hardening the product, (Figure 12). At a controlled point in the conveyor the stick is located into the middle of the ice cream, (Figure 13). The product again travels in the conveyor for a further distance, until it becomes completely solid and is removed from the mould by the stick to be externally coated, (Figure 14).

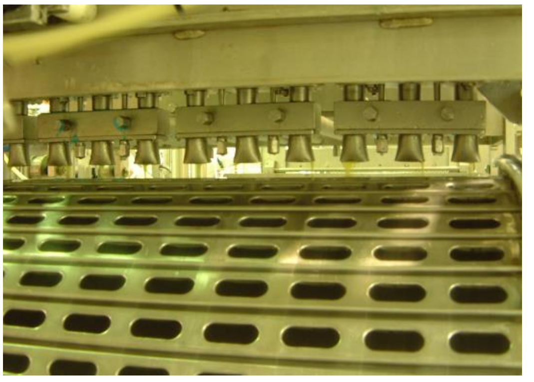


Figure 11, Ice Cream Inserted Into Moulds

Figure 12, Ice Cream is Hardened in Moulds

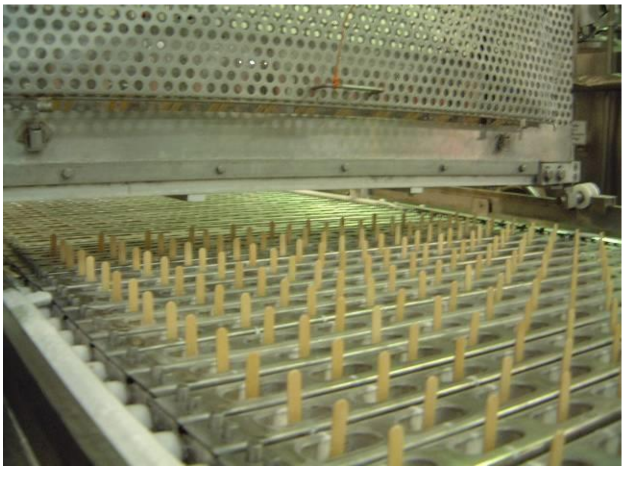


Figure 13, Sticks Inserted into Ice Cream

Figure 14, Ice Cream Removed by Stick After Fully Frozen

2.2.5 Batch vs Continuous Processing

In designing or choosing a refrigerated facility, a decision is to be made to process in a batch or continuous mode. Usually batch mode is suited for small throughput applications where as continuous is suited for large scale production.

Batch processing systems are simple in that they usually involve the filling of the facility with product, turning the cooling on and "walking away". Problems can occur with uneven cooling/freezing between products within batches as each product experiences a different experience in relation to the coolant because of its position within the cooler/freezer.

Typical designs of batch chillers/freezers for using air as the coolant (and carcasses as the product) are shown in Figure 15. The apple pre-cooler and tuna freezer (examples 1 and 2) are also batch processing systems. Batch systems usually require manual labour to load and unload product.

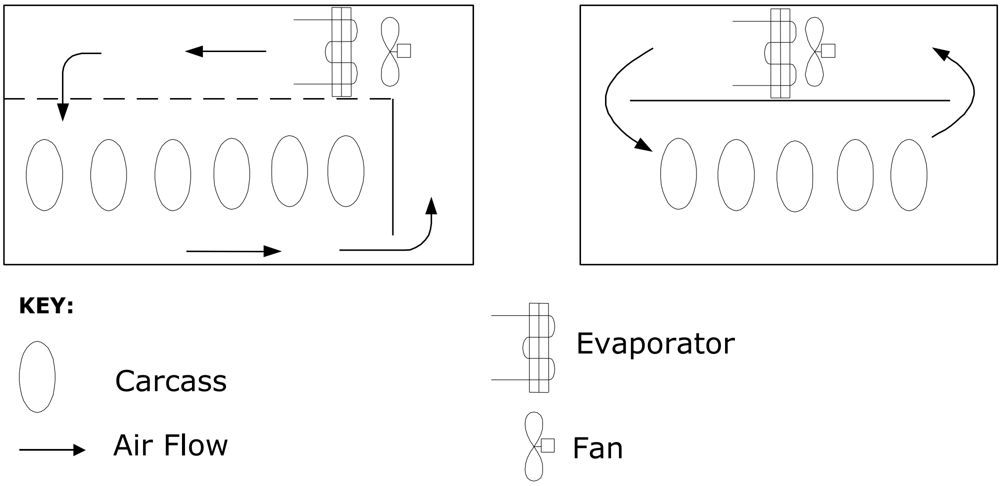


Figure 15, Carcass batch chillers/freezers

In batch processes, the air in the room, any metal fitting present, and the material of the floor (often concrete) will be at an elevated temperature at the beginning of the batch. These items will give up heat as the air temperature reduces with start-up, adding to the total heat load that needs to be removed per batch. By far the most important is the concrete floor, but the metal fittings can also be important.

Continuous processing systems involve the movement of the product through the cooling facility, usually by the means of a conveyor system. Energy requirements, (and heat load) are larger than batch systems due to the conveyor. However savings are made in that no labour is required to load and unload product. The ice-cream on a stick freezer (example 3) is an example of continuous processes. Figure 16 and Figure 17 shows two examples of continuous chillers/freezers. The spiral freezer shown in Figure 17 is used to reduce occupied floor space and evaporator size, as a long conveyor (and hence product exposure to cooling) is compressed into a small volume.

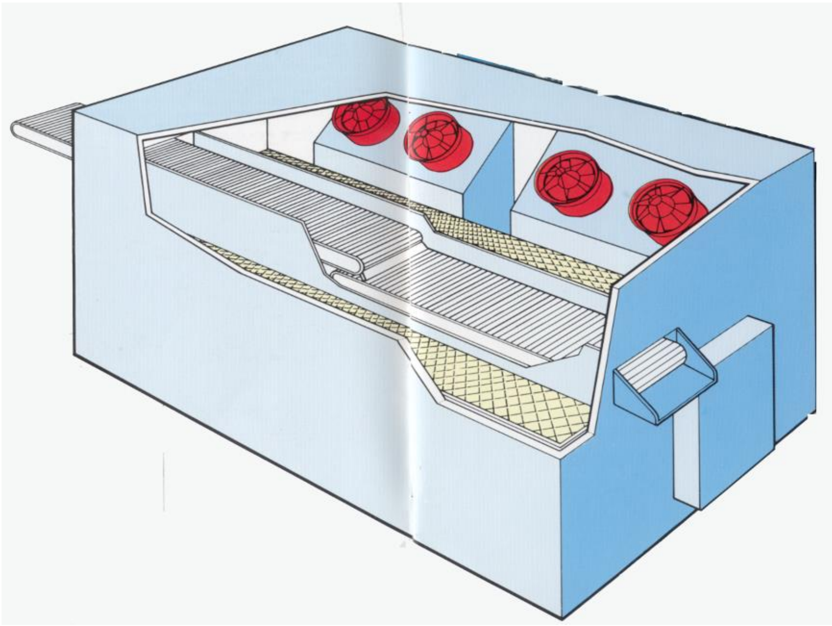


Figure 16, Continuous Freezer/Chiller Design

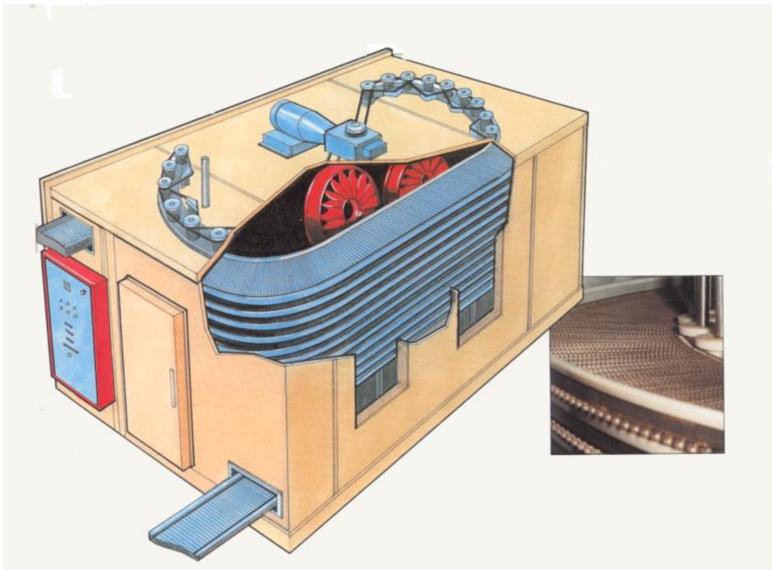


Figure 17, Spiral Freezer/Chiiler

In continuous systems the removal of heat is uneven down the length of the conveyor. The product entering the cooler is warmer than the product leaving the cooler, and hence the driving force (the temperature difference) is larger at the beginning of the cooling process. Early evaporators in the cooler are more heavily loaded, creating a design problem that can be partially solved with two passes as shown in Figure 18.

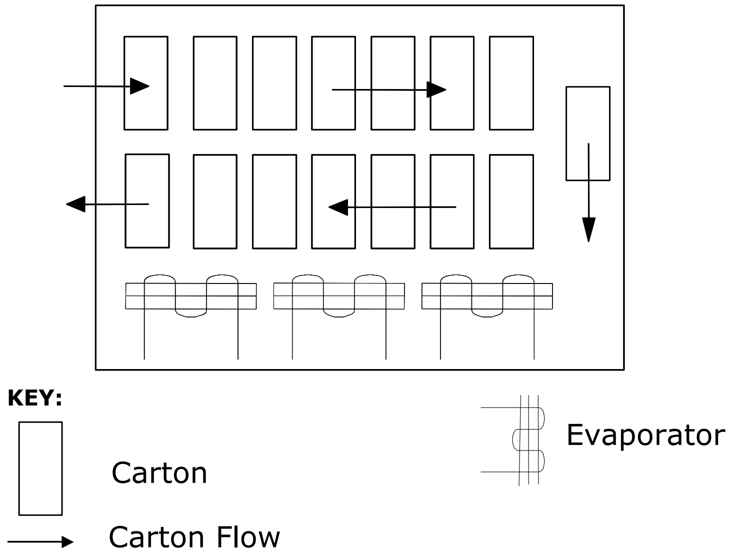


Figure 18, Design of two-pass continuous cooler

Defrosting continuous systems that use air as the coolant can be an issue. Evaporators can be turned off one at a time but shutters must be installed to completely isolate it if correct operation of the remainder of the freezer is to occur. Otherwise the entire process line needs to shut down to allow the refrigerated facility to defrost.

Another difference between batch and continuous systems is the effect on peak heat loads. In batch chillers the peak load occurs at the beginning of the process, because the majority of the cooling is being achieved at this time (because of the large temperature difference between the product and the cooling fluid). As a rule of thumb peak heat loads are 2-3 times average loads in batch chillers. In contrast, continuous systems peak loads are slightly above the average load during start up of the continuous system.

3 SPECIFICS OF FACILITY DESIGN

3.1 THE REFRIGERATION SYSTEM

Refrigeration systems have been covered in more detail in other courses. These notes are a brief introduction and revision of a basic refrigeration system. Figure 19 is a schematic of a simple refrigeration system.

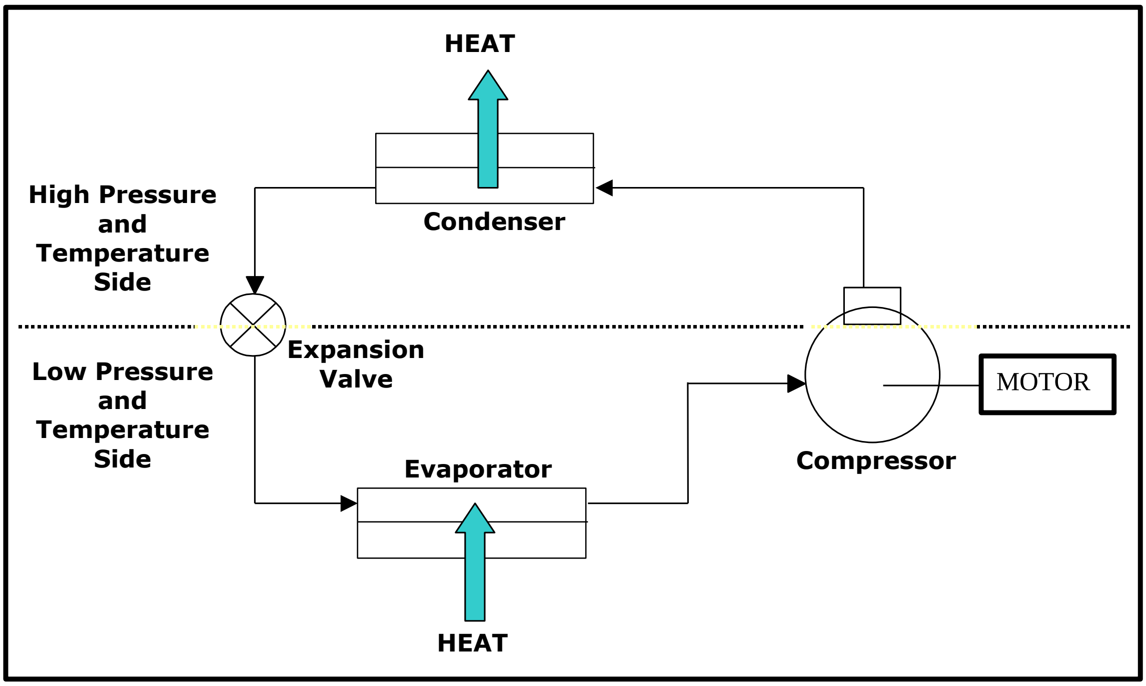


Figure 19, A Simple Refrigeration System

Cold liquid refrigerant enters the evaporator, is heated through the use of a heat exchanger, resulting in warmer refrigerant vapour leaving the evaporator. The choice of heat exchanger used for the evaporator is dependant on the choice of cooling medium being used.

The warm refrigerant vapour is compressed by the compressor to a superheated vapour at a higher pressure. Energy is supplied to the system through the motor that drives the compressor.

The condenser releases the heat gained at the evaporator and the compressor to another medium. In this process the refrigerant is cooled and condensed into liquid refrigerant. Again, heat exchanger choice is dependant on the fluid that is being warmed by the refrigeration system.

The expansion valve releases the warm high pressure liquid refrigerant to a lower pressure, creating a cold mixture of liquid and vapour refrigerant.

3.2 AIR/REFRIGERANT HEAT EXCHANGERS

As previously discussed, the most commonly used cooling medium is air, hence air/refrigerant heat exchangers are used commonly for both the condenser and evaporator in the refrigeration cycle. This course only covers air/refrigerant heat exchangers in detail as the theory for the other heat exchanger types that could be used has been covered previously in other courses.

The use of air/refrigerant heat exchangers is vital to the successful operation of refrigeration systems. This applies not just to the "cold" side, but also to the "hot" condensing side of a refrigeration plant if ambient air is used as the condenser coolant.

In the case of the condenser the metal surface in contact with the air is hot so the surface remains dry and there is only convection (+ radiation) heat transfer away from the hot metal to the air.

In evaporators both condensation (if exposed metal temperature > 0^0C) and frosting (if exposed metal temperature < 0^0C) can occur. In these cases as well as the convective heat transfer from air to metal surface there is heat transfer by condensation (and sometimes freezing) creating heat transfer performance beyond what would occur if the heat exchange surface remained dry.

Unfortunately, we do not have time to consider methods of analysis in detail. However, it is always "safe" to ignore the increase in heat transfer caused by the condensation when doing calculations (i.e. real heat transfer will be more than predicted). We will consider the heat transfer to commonly used finned heat exchangers (assuming no condensation in the case of evaporators).

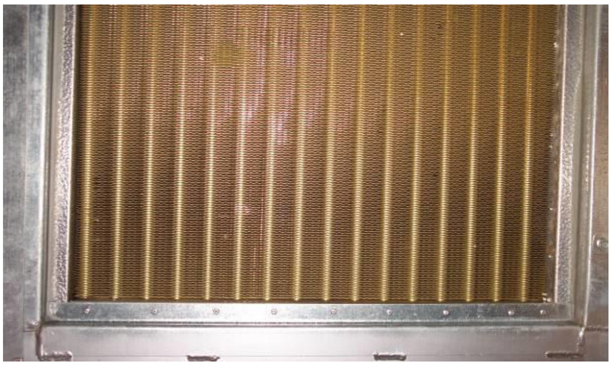


Figure 20, Typical Fin Sheet Evaporator

In the above picture (Figure 20), it is evident that the air/refrigerant heat exchanger is more than simple pipes. Refrigeration equipment typically uses the finned heat exchangers. The heat exchangers have fins on the outside for very good reasons.

Consider the resistance to heat exchange through a pipe, (ignoring curvature), given by equation 1.

$$R = \frac{1}{UA} = \frac{1}{h_1 A_1} + \frac{x}{k A_{12}} + \frac{1}{h_2 A_2} \qquad (1)$$

Where h_1	=	film heat transfer coefficient on the inside of the pipe (W/m^{2}K)
h_2	=	film heat transfer coefficient on the outside of the pipe (W/m^{2}K)
x	=	pipe thickness (m)
k	=	pipe thermal conductivity (W/mK)
A_1	=	area of the inside of the pipe (m^2)
A_2	=	area of the outside of the pipe (m^2)
A_{12}	=	mean heat transfer area within the pipe (m^2)

In an air/refrigerant heat exchanger the pipe is designed for heat transfer (thin and made from highly conductive metal) and therefore can be assumed to have little effect on the overall heat transfer resistance. Therefore equation 1 can be simplified to:

$$R \approx \frac{1}{h_1 A_1} + \frac{1}{h_2 A_2} \qquad (2)$$

In the case of refrigerated facilities where air is the chosen cooling medium, the internal heat transfer coefficient (h_1) represents the heat exchange causing the refrigerant to evaporate. Heat transfer coefficients for evaporative processes are approximately 1000 W/m^{2}K. The external heat transfer coefficient represents the heat transfer from the air (being transported via a fan) to the solid surface of the pipe. In contrast to the evaporative heat transfer coefficient, heat transfer coefficients for forced convection of gases to solids are approximately 20 W/m^{2}K.

Therefore the majority of the resistance to heat transfer in an air/refrigerant heat exchanger is caused by the external or air side heat transfer coefficient. When considering equation 2, one method of counteracting the difference in heat transfer coefficients is to make the external surface area A_1 significantly larger than the internal surface area A_2, hence fins are used.

There are three types of fins that are used for air/refrigerant heat exchangers. Each type is introduced in these notes. Formula are given in order to assess heat transfer capability of any finned heat exchanger.

3.2.1 Longitudinal Fins

Fins that are parallel to the pipe carrying the fluid are called longitudinal fins. These fins are difficult and expensive to make in practice due to the long weld lengths required. The most common example of longitudinal finned heat exchangers are the “radiator” heaters used in school room heating. A drawing of a longitudinal finned heat exchanger with nomenclature definition is shown in Figure 21.

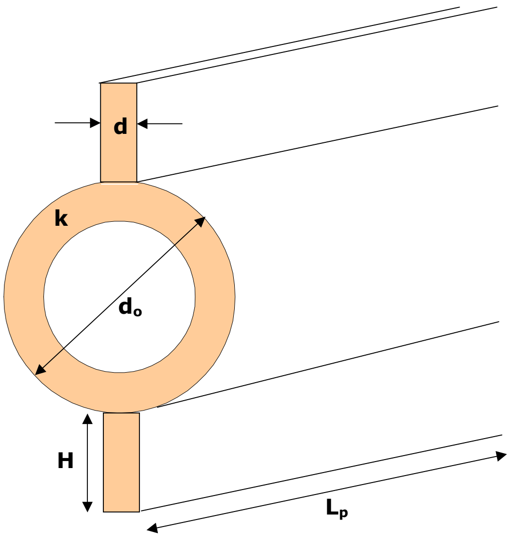


Figure 21, Longitudinal Finned Heat Exchanger with Nomenclature

3.2.2 Collar Fins

An alternative means of extending the external surface area is to attach circular "collars". Collar fins are defined as fins that are perpendicular to the pipe and are drawn in Figure 22.

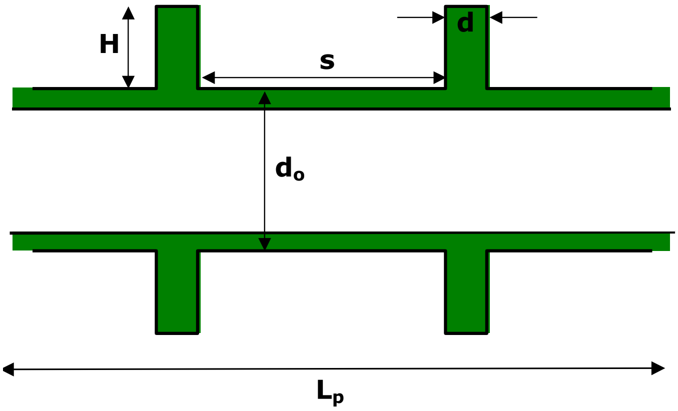


Figure 22, Collar Finned Heat Exchanger with Nomenclature

3.2.3 Fin Sheet

An extension of the collar fins and by far the most common type used is the "fin-sheet". In this fin type, perpendicular fins join multiple pipes, the tubes pass through holes cut in rectangular sheets of thin metal (usually aluminium). A car radiator is a common example of a finned sheet heat exchanger. A photo of a fin sheet heat exchanger in a refrigerated room is shown in Figure 23.

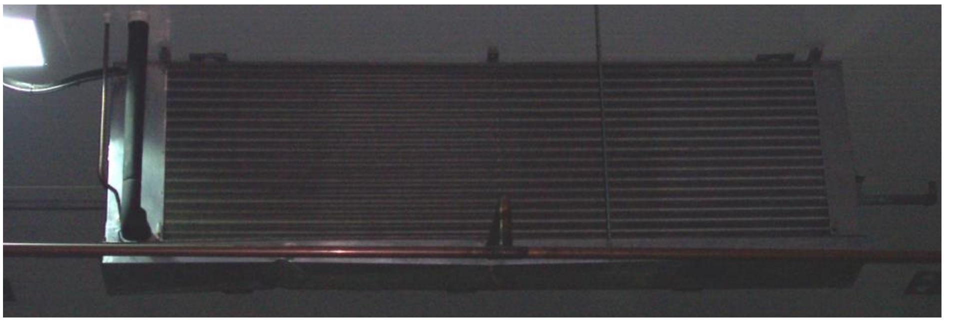


Figure 23, Fin Sheet Heat Evaporator in a Refrigerated Facility

Fin sheets, exists in two styles, triangular and square pitched, depending on the arrangement of the pipes through the sheet, as shown in Figure 24.

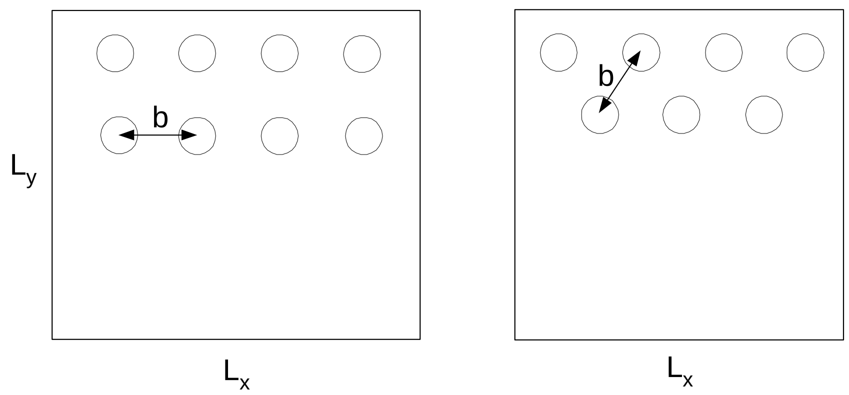


Figure 24, Fin Sheet Heat Exchanger Styles and Nomenclature

The fin sheets can be very close together (2-4 mm) or up 12 mm apart. Larger spacings (often 6-12 mm) are used in air cooling units where frost will form, and narrower spacings in condensers in which the air supply is very clean. Cleaning of finned surfaces can be a problem.

3.2.4 Calculations for Finned Sheet Heat Exchangers

The calculation of the performance of a finned heat exchanger is not straightforward. The heat transfer in a fin heat exchanger is multidimensional, causing calculation difficulties.

This is best illustrated by an example – consider air at 20^0C flowing over tubes in which refrigerant condenses at 40^0C shown in Figure 25. If virtually all the heat transfer resistance is in the air-side coefficient the tube wall will be at almost 40^0C. However along the metal of the fins the temperature will get progressively lower as heat is lost from the metal.

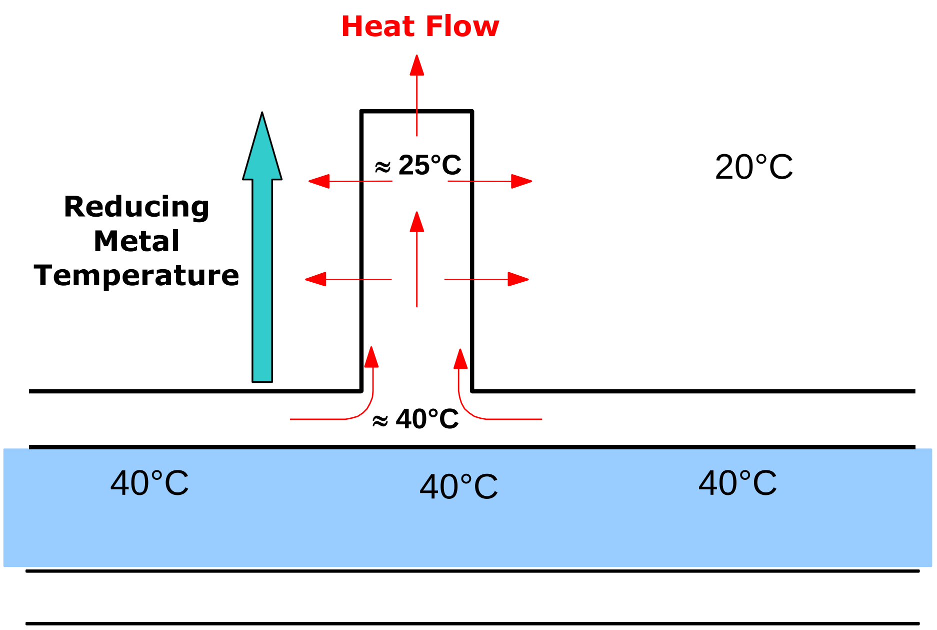


Figure 25, Heat Transfer from a Finned Condenser

The temperature driving force for heat transfer from the metal to the air gets progressively lower as the distance from the pipes increases. Hence heat transfer area on the fin is less effective in transferring heat than the pipe wall itself. To take account of this a term called the fin efficiency is introduced to equation 1. The equations for ϕ and (UA) are then written as:

$$\phi = (UA)\Delta\theta \qquad (3)$$

$$\frac{1}{UA} = \frac{1}{h_1 A_1} + \frac{x}{kA_{12}} + \frac{1}{h_2 (A_p + \eta_f A_f)} \qquad (4)$$

Where ϕ = heat flux (W)

ϕ	=	heat flux (W)
(UA)	=	overall heat transfer coefficient (W/K)
$\Delta\theta$	=	temperature difference between air and fluid in pipe (K)
h_1	=	film heat transfer coefficient on the inside of the pipe (W/m^{2}K)
h_2	=	film heat transfer coefficient on the outside of the pipe (W/m^{2}K)
x	=	pipe thickness (m)
k	=	pipe thermal conductivity (W/mK)
A_1	=	area of the inside of the pipe (m^2)
A_2	=	area of the outside of the pipe (m^2)
A_{12}	=	mean heat transfer area within the pipe (m^2)
A_p	=	primary area (area of exposed pipe) (m^2)
A_f	=	secondary area (fin surface area) (m^2)
η_f	=	fin efficiency

Although the reality is that the correct mean temperature difference for the fin is not the same mean temperature difference as for the pipe it is most convenient to use a unique temperature difference and apply the compensation to the area. The first step in the analysis is to find the temperature distribution along the fin. Once this is known the actual mean temperature difference for the fin can be found and hence:

$$\eta_f = \left(\frac{True\ \ mean\ \ T.D.\ \ of\ \ fin}{Mean\ \ T.D.\ \ pipe \rightarrow air} \right) \qquad (5)$$

The calculation of temperature difference along the fin is complicated and so will not be covered here. Instead the results are stated in Table 2 and Figure 26, and you are expected to be able to apply these.

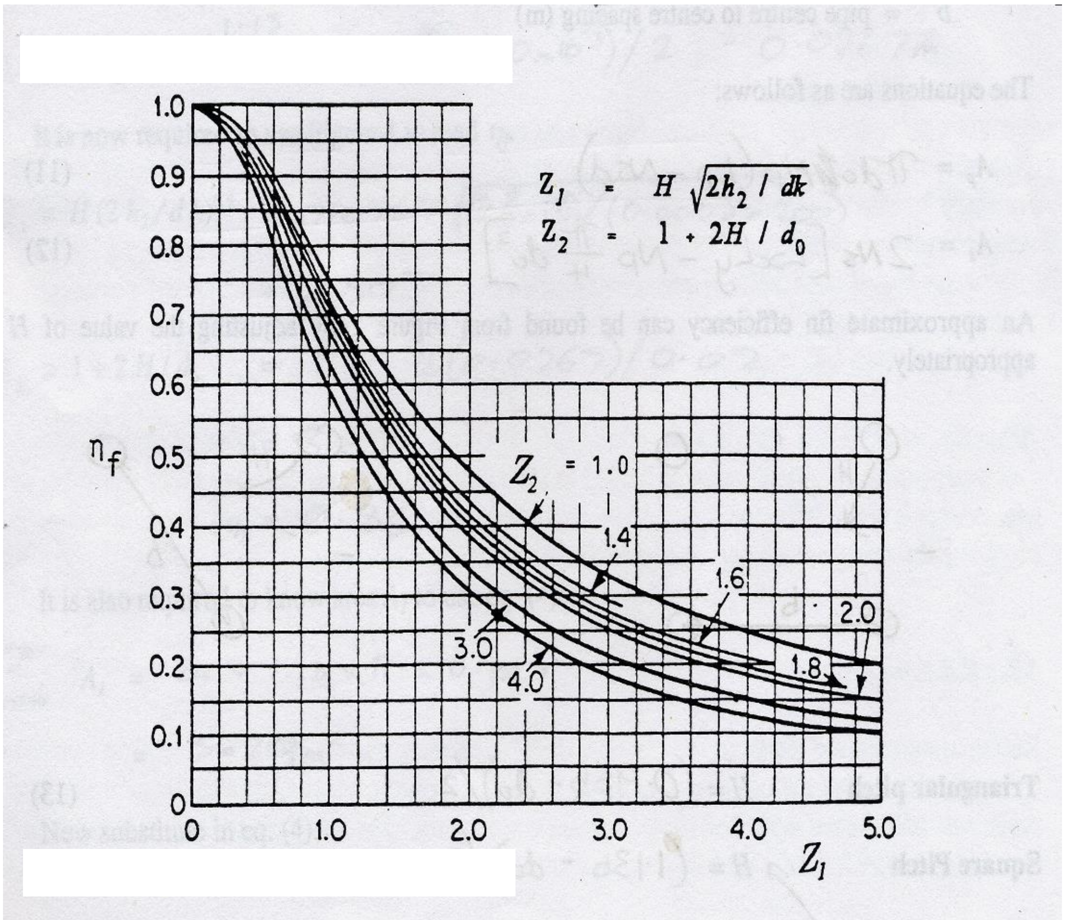


Figure 26, Fin Efficiency Chart

General Nomenclature:

d = fin thickness (m)
L_p = length of pipe (m)
d_o = pipe outer diameter (m)
N = number of fins

FIN TYPE	LONGITUDINAL	COLLAR	FIN SHEET
Specific Nomenclature			L_p= length of pipe for one pass (m) b = pipe centre to centre spacing (m) L_X = finsheet dimension in x (m) L_y = finsheet dimension in y (m) N_p = number of pipes
Specific Equations	$\tanh(x) = \frac{e^x - e^{-x}}{e^x + e^{-x}}$	$N = \frac{L_p}{S + d}$	$H = \frac{(0.95b - d_o)}{2}$ (Triangular Pitch) $H = \frac{(1.13b - d_o)}{2}$ (Square Pitch)
$\boldsymbol{A_p}$	$L_p(\pi d_o - Nd)$	$\pi d_o(L_p - Nd)$	$\pi d_o N_p (L_p - Nd)$
$\boldsymbol{A_f}$	$2NL_pH$	$2N\pi(H^2 + d_oH)$	$2N\left(L_xL_y - N_p\frac{\pi}{4}d_o^{\ 2}\right)$
$\boldsymbol{\eta_f}$	$\frac{\tanh\left[H\sqrt{\frac{2h_2}{dk}}\right]}{H\sqrt{\frac{2h_2}{dk}}}$	Figure 26	Figure 26

Table 2, Formula for Prediction of Fin Heat Exchanger Performance

3.3 WORKED EXAMPLE – AIR FIN HEAT EXCHANGERS

A coldstore is being built – the expected heat load to be removed is 18.8 kW at an air temperature onto the evaporator of –18°C. An old air cooling unit is available – it has a face area of 1.65m^2, and the fan maintains an air flow of 2.76 m/s off this face. The unit has aluminium fins (k = 200 W/mK) of thickness 0.55mm and spaced at 8.5mm centres. The unit length is 1.86m, and its depth in the direction of air flow 0.52m. the film heat transfer coefficient in the boiling refrigerant has been estimated as 265 W/m2K, and that in the air flow as 19 W/m2K. There are 112 tubes of outside diameter 21 mm and inside diameter 18.5mm, these being placed in square itch at 61mm centres. Determine the refrigerant evaporation temperature required if the unit is to accomplish the required heat transfer.

3.3.1 Scenario diagram

3.3.2 Calculate the air off temperature

3.3.3 Calculation of the UA value

3.3.4 Calculate the required evaporation temperature

3.4 DEFROST SYSTEMS

Evaporators of refrigeration systems operate at lower temperatures than the fluid that the refrigeration system is cooling in order to cause heat transfer. In the case where air is used as the process fluid, the evaporator temperature is often less than the dew point temperature (wet bulb temperature) of the air. This results in condensation or in the case of temperatures below 0°C, freezing of water on the surface of the evaporator.

This frozen water on the evaporator can be considered as a hindrance to heat transfer in the same means as fouling of plate heat exchangers. The heat transfer surface is covered with a film that not only makes the heat transfer surface thicker but also does not have good heat transfer properties. On top of this reduction in heat transfer capability, in extreme cases, ice can bridge fins, resulting in reduced air flow through the heat exchanger and around the facility.

Subsequently evaporators in refrigeration applications that use air as the cooling fluid must be defrosted periodically. There are four common methods:

- Electric defrost
- Water defrost
- Hot Gas defrost
- Air defrost

These methods are occasionally combined to increase speed of defrosting. The frequency of defrost may be as little as once a year in some applications where water entry to the room is very slow, or every hour or two in some continuous freezers for wet product.

In choosing a defrost method, one must balance the speed of defrost against the cost of the defrost system. Storage facilities require quick defrost systems, or defrost systems that isolate the evaporator from the store in order to minimise the inevitable increase in temperature that occurs as a result of the refrigeration system being turned off and heat being added to melt the frost.

Those who observe a temperature rise when defrosting is in progress often assume that this is totally due to the defrost system heating the room. This is incorrect. Consider a coldstore at -18^{0}C with about 30 kW total heat load with 4 cooling units handling the load. When 2 are turned off the heat removal drops to 15 kW, yet the heat entry to the room is still 30 kW. Hence the heat not removed with some cooling units out of action leads to room air temperature rise.

Simultaneously, the defrost system adds some heat but this is largely used to melt ice and hence it disappears out of the room in the melted liquid. A small fraction of the defrost energy does heat the evaporator metal and air nearby. Commonly, the total heat load of a room is increased by 2-15% to account for the heat entering from the defrost system.

Another important aspect of defrosting is that it reduces the time available for room cooling. If defrosting takes 1 hour then heat entering in 24 hours must be removed in 23 hours, and the air cooling units and refrigeration system sized accordingly.

3.4.1 Electric Defrost

The evaporator fan is turned off and radiant heating elements (located between the fins) are turned on. This system has advantages in that it has a low capital cost and defrost has immediate effect. However, electric defrost may be slow, and is expensive. For this reason electric defrost is usually used in small applications only.

3.4.2 Water Defrost

The refrigeration and fans are turned off and (possibly heated) water is run over the outside of the evaporator. As with all defrost systems in negative temperatures, the water pipes and drains must be heated to stop water freezing in them. Once water drainage is complete the refrigeration can be turned back on.

Water defrost is comparatively fast and has little operational cost, (only those associated with heating pipe work and pumping the water). Capital costs for water defrost systems are middle of the range. If water is in shortage or is costly, water defrost may not be an attractive option as large amounts of water are required.

3.4.3 Hot Gas Defrost

The refrigeration system can be reversed so hot refrigerant gas from the compressors flows into the evaporator, temporarily converting the evaporator into the condenser. The ice is melted by the refrigerant condensing in the evaporator resulting in the ice losing its bond before it is fully melted.

The hot gas defrost system is fast as not all of the ice is melted and is also inexpensive as the heat used is that recycled from the refrigeration system. However the capital costs of hot gas defrost systems are large due to the sophisticated valve systems and extra piping required to be added to the refrigeration cycle. Another risk of hot gas defrost systems, is exposing the heat exchanger to swift changes in temperature (during start-up and shut-down of defrost) that can result in mechanical damage.

3.4.4 Air Defrost

In air defrost, the refrigeration system is turned off and evaporators are shut off from the refrigerated room (usually with false ceilings or walls). Air flow from outside the refrigerated room is introduced to the evaporator using the same fans that are used in the air distribution system. The exiting air flows back outside.

Air defrost is relatively slow, but offers advantages in that the operating and capital costs are very low.

3.4.5 Defrost Controls

The start-up and shut-down of defrost can be manual or automatic. Some automatic controls work purely on a timer, others monitor the ice layer build-up, or the temperatures of refrigeration or cooling fluid.

Batch freezers and chillers are usually defrosted in their “off” cycle. Continuous freezers and chillers, and cold stores are applications where more sophisticated controls are required, as defrost is required to be conducted while the facility is “working” on the product. Care must be taken to minimise the heat gain caused by the defrost system. This is achieved by installing quick defrost systems and shuttered the evaporator from the rest of the facility during defrost.

Malfunctions of defrost controls can be responsible for damage to product and the refrigerated facility. For this reason, alarm systems need to be installed to minimise the damage caused by any malfunction.

3.5 AIR DISTRIBUTION SYSTEMS

As chapter 2 introduced, the "cold" that is produced by the refrigeration systems is required to be supplied to all areas of the refrigerated facility. This is achieved with careful design of the air distribution system. There are five air distributions systems that you should be aware of:

- Natural convections systems
- Multiple unit systems
- Single unit systems
- Plenum Ceilings
- Delta Wings

Modern stores tend to use forced draught cooling (FDC) units, which consist of the refrigeration system evaporator and fans that force the air through the heat exchanger and around the facility. In very small stores it is very difficult to direct the air flow to any great extent, but for large stores it pays to carefully plan it. The product arrangement in a store should reflect the air flow pattern.

3.5.1 Fans

Fans have a single purpose in cold and cool stores; to move air to the areas where heat is entering the system, and bring this heat back to the evaporator for removal.

In freezers and chillers the air movement has two purposes, one to bring heat from the product back to the evaporator for removal and second to increase the rate at which heat is removed. The latter allows the freezer to be more compact and capital costs to be decreased, although at the expense of higher fan operating costs. There is a limiting air velocity beyond which there is no benefit in further increases. Thus, optimisation of fan inputs is important as they typically provide about 30% of the total heat load in freezers.

The energy input (as electricity) to the electric motor driving the fan is converted into mechanical energy (about 90%) and heat (10%) inside the motor because electric motors are not 100% efficient. Mechanical energy transmitted down the shaft is partly lost as friction on the fan blade, becoming heat immediately, and partly converted to kinetic energy as air movement. The friction among the molecules of air causes the air movement to slow down and finally to grind to a halt if the fan was turned off. In the process the mechanical energy is converted into heat. Thus all the energy input to a fan motor ends up as heat load to be removed by the refrigeration system.

3.5.2 Natural Convection Systems

Older stores had grid pipe around the walls and on the ceiling as shown in Figure 27. Liquid previously cooled by the refrigeration system is pumped through the pipes.

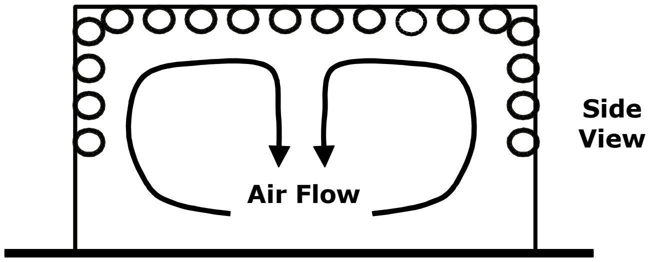


Figure 27, Grid Piped Refrigerated Facility Air Flow

Such an arrangement picks up the thermal heat load through the walls well, but does not cope well with the sudden large heat flows when doors are opened. Hence product located close to the doors may fluctuate markedly in temperature. A good feature of this type of system is that the evaporation that can occur from unwrapped product (known as weight loss) is low due to the minimal movement of air over the product.

3.5.3 Multiple Unit Systems

In multiple unit system, FDC units are staggered down the length of the refrigerated facility, effectively creating zones of "responsibility" for each FDC unit. An example of a multiple unit system configuration is shown in Figure 28. An example store is shown in Figure 29.

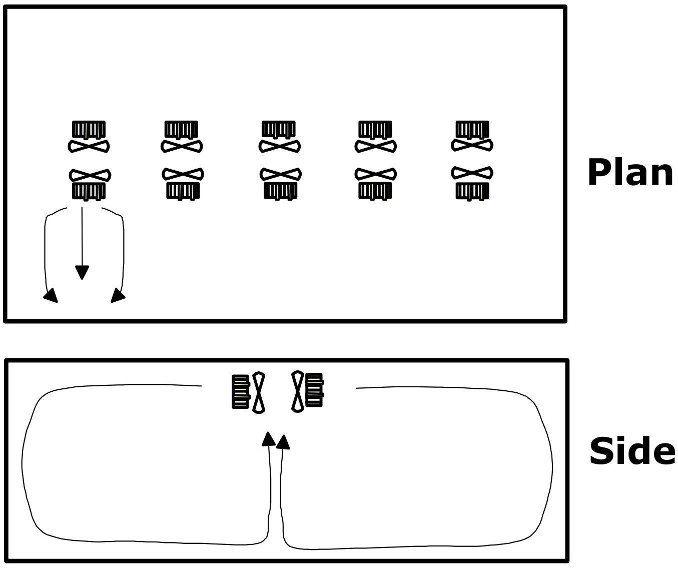


Figure 28, Typical FDC Configuration for a Multiple Unit Store

Multiple unit system provide advantages in that defrost in the facility can be staggered amongst the FDC units, reducing the increases in product temperature caused by defrost. Similarly, breakdown of a FDC unit does not have catastrophic effect on temperature, as the other FDC units are able to "take up the slack".

Multiple units systems do have comparatively large operating and capital costs, due to the multiplication of valves, pipe work, and fans required.

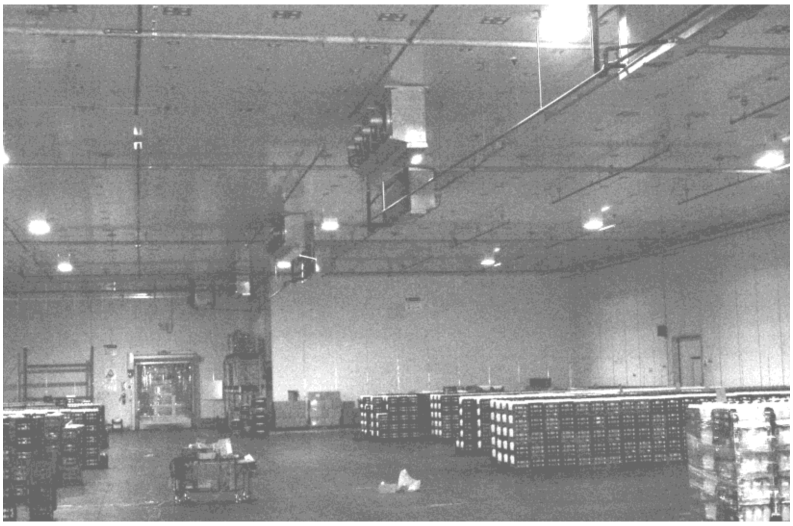


Figure 29, Multiple Unit Store used for Milk Distribution

3.5.4 Single Unit Systems

Single unit systems use one large FDC unit and carefully designed ducting systems to ensure that the "cold" is delivered to the entirety of the room. A typical arrangement for single unit systems is shown in Figure 30 and an example shown in Figure 31.

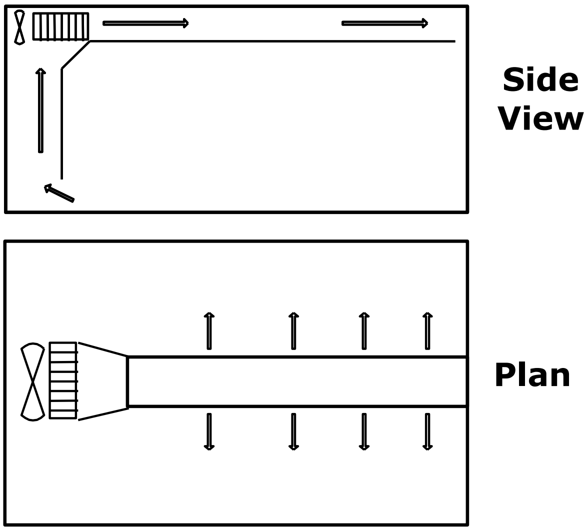


Figure 30, Typical Design for a Single Unit Distribution System

The air intake to such units is usually located on the wall opposite to the doors. The air leaving the evaporator is passed across the ceiling through the ducting and directed perpendicular to the ducting so that it will then fall down the walls that are parallel with the ventilation system.

The use of single unit systems results in a reliance on the FDC not breaking down. If in any situation that it does, there are no mechanisms to stop temperature increase. Similarly, in storage situations, defrosting single unit systems will result in a peak temperature occurring. However single unit systems offer benefits in terms of capital and operational cost savings.

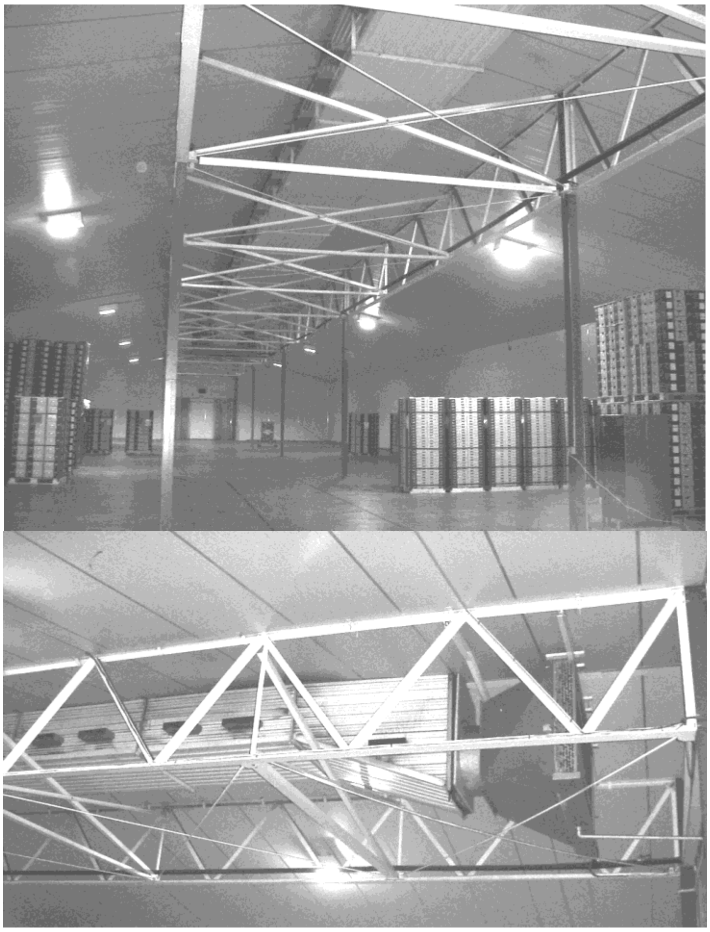


Figure 31, Typical Single Unit Ducting System

3.5.5 Plenum Ceilings

Plenum ceilings are used in freezer/chiller scenarios where vertical airflow is required to be produced. These were first developed for and are most commonly used in carcass chiller/freezers. Figure 32 shows a carcass freezer with a plenum ceiling.

Plenum ceilings feature sealed (and pressurised) roof spaces with small gaps that force the air to pass vertically downward past the product. Unless the gaps in the plenum ceiling are individually adjusted uneven air flow will occur. Provision for adequate chilling when the chiller is not full should also be made. The chiller shown in Figure 32 is run in batch mode, however it is possible to have continuous chillers with plenum ceilings.

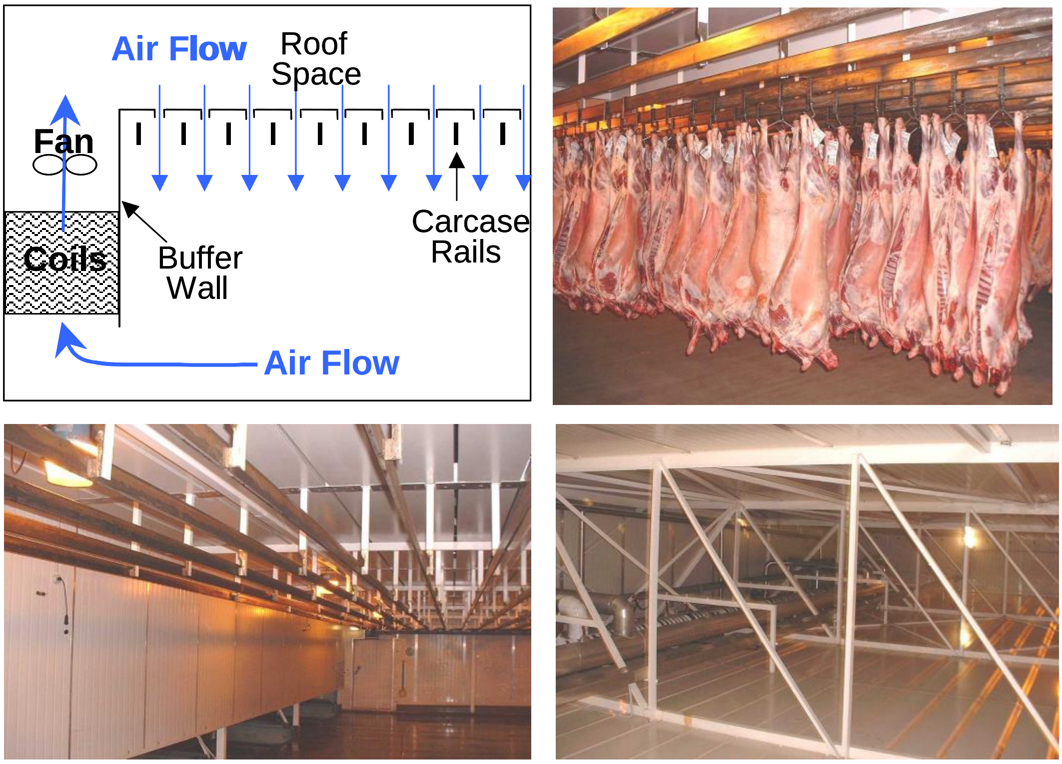


Figure 32, Carcass Freezer with Plenum Ceiling. (From left to right, top to bottom: drawing; full freezer; empty freezer; roof space).

3.5.6 Delta Wings

Delta Wings are shown in Figure 33. These devices are a relatively new invention in refrigerated facility air distribution and are used in chiller/freezer applications. As shown in Figure 34, the principle of Delta Wings is to create vortices.

The vortices created have two advantages, the first being that they enhance air mixing and hence create a more consistent air temperature across a facility. The second benefit is that they increase localised air velocities (and hence heat transfer coefficient) without the addition of energy, as a linear airflow pattern is converted into a turbulent flow.

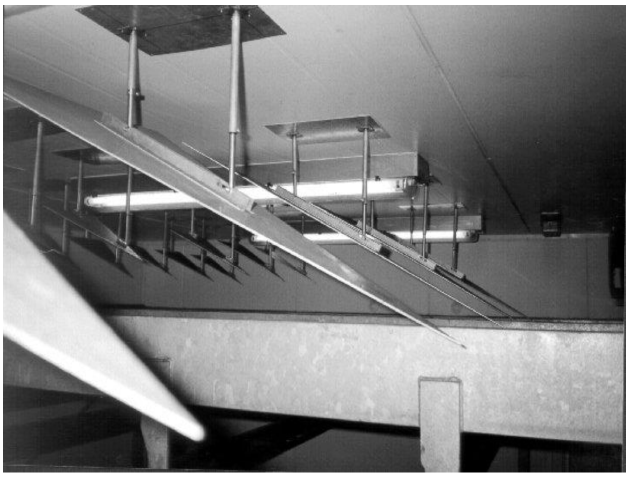


Figure 33, Installed Delta Wings on a Ceiling of a Chiller

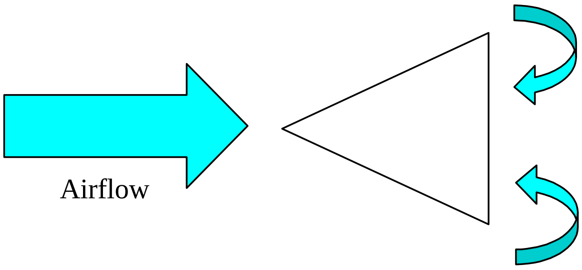


Figure 34, Basic 2-Dimensional Drawing of Delta Wing Vortex Formation

3.6 INSULATION PANELS

All refrigerated facilities use a layer of thermal insulation around them to reduce the heat load on the refrigeration system. By far the most common insulation used in New Zealand in expanded polystyrene (EPS) panels, as shown in Figure 35.

Figure 35, An EPS Coolstore

Other options for insulation include fibreglass, corkboard and polyurethane foam. Approximate thermal conductivity values for these materials are as below.

Fibreglass	0.040 – 0.050 W/mK
Corkboard	0.035 – 0.040 W/mK
Polyurethane foam	0.022 – 0.030 W/mK
Polystyrene foam	0.028 – 0.035 W/mK

3.6.1 Wall Insulation Performance

Such a layer of insulation is only effective if the insulation remains in good condition. Reduction in insulative performance can be caused by:

- Thermal bridges being formed by necessary pipework or construction materials (i.e. bolts) that pierce the insulative material.
- Air gaps between panels that allow heat to be transferred by natural convection.
- The insulation becoming wet and hence more thermally conductive.

Ensuring that good construction techniques are used can solve the first two of these issues. However , particular importance in refrigerated facilities is the dryness of the insulation. Ice has a thermal conductivity 80 times that of most insulants. Hence if insulation becomes wet it loses most of its insulating value. Further, buildup of ice within the wall leads to expansion of the wall and an increase in panel weight with mechanical damage as a result.

A vapour barrier is required to keep the insulation dry. Common materials are plastic films and aluminium sheets. The structural integrity of the vapour barrier must be maintained for the barrier to work. The joins between aluminium sheets must be sealed.

In calculations, the insulation effectiveness factor, E_i, takes into account shortcomings in real insulation performance compared to theoretical. As an estimate:

- E_i is always greater than 1.0,
- For small rooms (less than 100 m^3) it can be as high as 2 to 3,
- For large rooms (more than 500 m^3) it is more likely to be 1.1 to 1.5.

3.6.2 Wall Surface Temperatures

In most cases the inner wall temperature (θ_i) is not very different from the bulk air temperature in the room (θ_a). For the walls and roof of a cold or cool store the outside wall temperature (θ_o) can be assumed to be approximately equal to the air temperature in shaded areas (15-25°C average in New Zealand), but if direct sunlight falls on the roof or wall panels the temperature of the surface ca be in excess of 50°C.

The temperature increase caused by the sun, can be avoided by having a weather shield (a roof that shields the refrigerated facility from the weather) that not only reflects sunlight but also deflects the rain. It can be assumed for calculations in this case that the outer surface temperature of a cold or cool store will probably lie in the range 0-20°C greater than the ambient air temperature.

3.7 FLOORS

All store floors are usually concrete in order to handle the large weight of stock and also provide a washable surface. However, refrigerated facility floors require other features:

- insulation to reduce heat load
- vapour barriers to protect the insulation
- heating to avoid structural damage.

Subsequently the floors of refrigerated facilities are often constructed in the manner shown in Figure 36. Polystyrene is usually used as the insulation in floors.

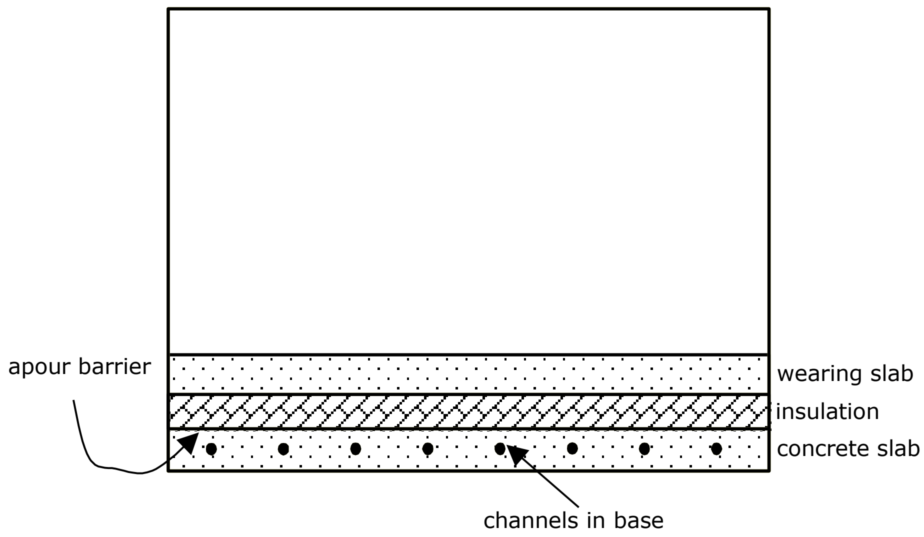


Figure 36, Typical Refrigerated Facility Floor

It is important to ensure that these features of refrigerated facilities are constructed correctly, as once the concrete is poured, any mistakes that may have been made are irreversible.

3.7.1 Floor Vapour Barriers

As with insulation panels, it is important to ensure that moisture is not allowed to infiltrate the protective insulation. A number of moisture barriers that can handle the weight stresses could be used:

- polythene
- rubberised bitumen
- aluminium foil
- polyurethane

The use of aluminium foil in series with one of the other vapour barriers is the recommended method to ensure good protection of the floor insulation.

3.7.2 Floor Heating

If the soil freezes under a store, the ice formed will physically lift the floor, causing frost heave mechanical damage.

In order to prevent the ground freezing under a refrigerated facility heating under the floor is provided. This can take a number of forms:

- electrical heating tapes in the concrete floor under the floor insulation
- air channels through the concrete relying on wind movement.
- Heated water of oil through pipes (usually heated with waste heat from the refrigeration system)

Ground temperature control can be automatic or manual, and aims to keep the soil temperature at about 5°C. A number of measurement probes are installed when the floor is built as they cannot be repaired once the floor is finished.

3.8 DOORS

Doors (or gaps in the walls) are obviously a necessity to allow product to flow into and out of a refrigerated facility. Unfortunately, these gaps in the facility also facilitate air interchange. Air interchange is the term used to describe the replacement of cold air in refrigerated air space by warm air from the outside. Whilst there is leakage through door seals and some interchange due to pressure equalisation systems the bulk of interchange occurs during door openings.

Air interchange has significant effects on the refrigerated facility including increases in:

- Heat Load – Warm moist air entering the building must be cooled and dehumidified to the conditions in the building. This heat load is one of the major causes of temperature variation and consequential product quality deterioration.
- Refrigeration System Costs – The capital and operating cost of the refrigeration system is proportional to the heat load. Use of door protection systems can reduce heat load but increase the capital cost of the building structure.
- Defrosting – The moisture in the entering air is deposited as frost on the cooling coils resulting in a reduction of refrigeration efficiency and the need to defrost.
- Frosting – The moisture in the entering air can also be deposited as ice on floors, walls, ceilings and product, often creating a hazard and impairing productivity.

When a door is opened, half of the doors cross-sectional area is used for airflow into the store, the other half used for airflow out of the store, as shown in Figure 37.

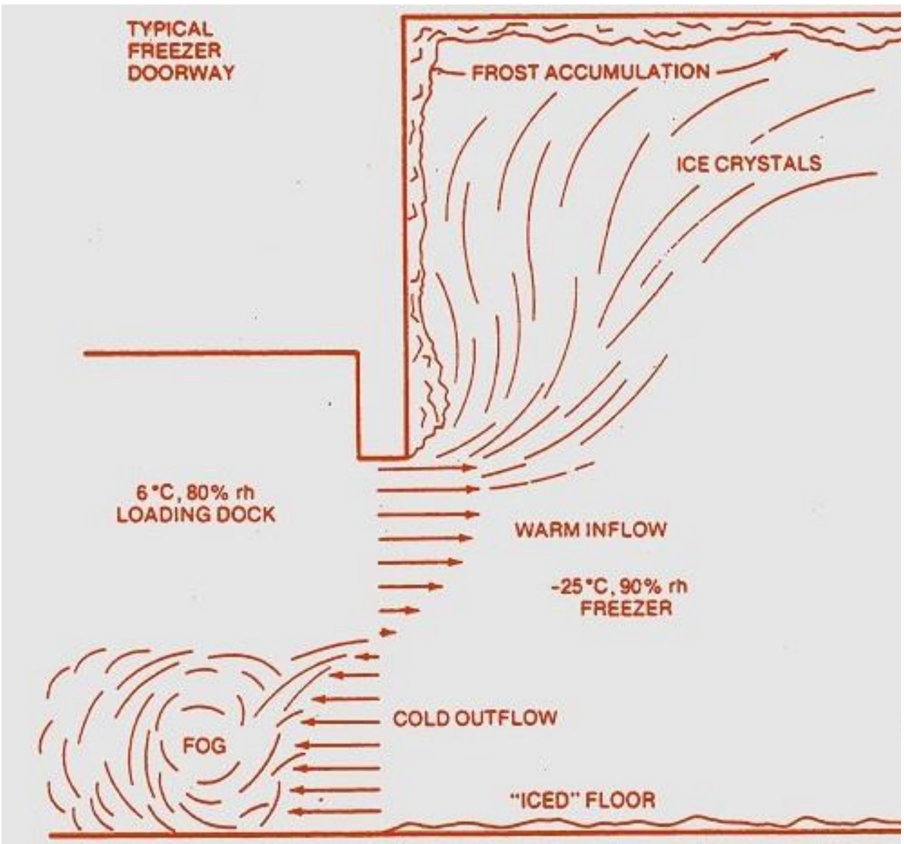


Figure 37, Airflow Pattern and Resulting Effects During Door Opening

The level of air infiltration affects the operating costs and efficiency of the store dramatically. Bad case scenarios like that shown in Figure 38, not only indicate a large unnecessary heat load, but cause large amounts of product (and packaging) damage and present a safety issue (due to the build up of ice).

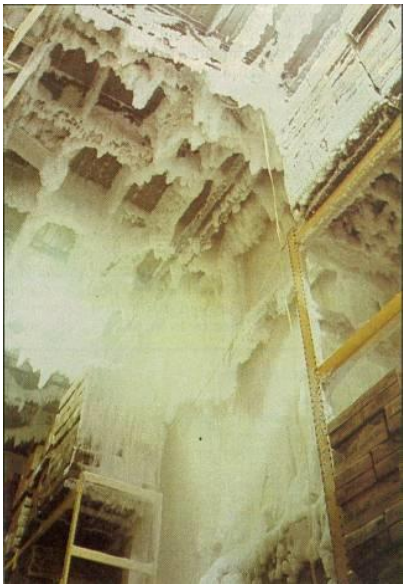


Figure 38, A Bad Case of Air Interchange in a Freezer

Reduction of air interchange can be significantly reduced by effective operation of doorways (i.e. not leaving doorways open), and the use of doorway protective devices. A number of options exists, some of which can be used in combination including:

- rapid roll doors , (Figure 39)
- strip curtains (Figure 40) and folding doors,
- air curtains (Figure 41)

If forced draught fans are in operation during door openings, and these pressurize the area around the door interchange can be increased. Simultaneous door openings can lead to a total interchange rate less than the sum expected by treating them independently. However, in some cases, e.g., multi-level stores opening of doors simultaneously at both high and low levels can lead to a “funnel” effect with high velocities.

3.8.1 Rapid Roll Doors

Rapid roll doors provide advantages in that the time the door is opened is minimised. Typically doors open (and close) in the order of 2 seconds and can be set to stay open for a time in which it takes a forklift (or product) to pass through the gap. Activation of the opening door can be conducted in a number of ways, including with magnetic presence detectors, onboard forklift controls or with simple pull cords.

Figure 39, Rapid Roll Door with Open EPS Sliding Door

As shown in Figure 39, rapid roll doors should be installed with sliding EPS panel doors. The rapid roll door does not provide insulative protection for the store. The EPS sliding door should be used in times when the facility is not being operated (i.e when no stock is being moved) to reduce heat load by heat infiltration.

3.8.2 Strip Curtains and Folding Doors

Strip curtains and folding doors provide physical barriers to air interchange. Hanging in the door way, traffic pushes through the curtains to enter the room as shown in Figure 40.

Figure 40, Strip Curtain

Strip curtain have been shown to reduce air interchange by 95%, although this is reduced with increase traffic. Disadvantages of strip curtains include reduced visibility through the doorway creating a safety hazard, and a subsequent reduction in traffic speeds through the doorway reducing facility productivity.

3.8.3 Air Curtains

Air curtains blow large velocity airflow across the door space. As a result air interchange is unable to occur and insects can be prevented from entering the facility. Air curtains have been shown to reduce air interchange by 85%.

Figure 41, An Air Curtain Installed on a Coldstore

Air curtains have advantages over strip curtains in that there is no physical barrier and hence provide visibility and reduce traffic safety concerns. However this advantage must be balance against the reduced air interchange protection and the fact that air curtains contain an additional operation cost (for the fan to provide the airflow).

3.9 LIGHTING

Lights should be located in alleyways between product stacks to maximise effectiveness, hence product stacking arrangements must be decided before the store is built.

The energy output of a light source is in two forms – light and heat. The total energy output of a light source can be directly read from the wattage of a bulb. The energy used by lights must be paid for twice – both to put it in, and to take it out via refrigeration. To decrease the energy used two factors must be looked at:

1. Use of more energy-efficient lighting, e.g., fluorescent, which gives more light per unit of energy input than filament type lights. Cold colour can be a psychological problem to workers so that the yellow glow of sodium vapour lighting has become increasingly common.

2. Provision for more independent controls of lighting in low temperature areas so that only the necessary working area is illuminated, and necessary lights can be turned off with a minimum of operator effort.

3.10 STORING RESPIRING PRODUCTS

Many fruit and vegetable products in particular, but also some fermented products (i.e. salamis) generate heat during cool storage as a result of continuing biochemical activity (especially respiration). Typical rates of heat generation are 5-40 W/tonne for long-storing produce to 200-400 W/tonne for produce with short life. The rate of respiration drops rapidly as storage temperature is decreased. Values of respiration rate (R_r) can be found in many published text books or reference books.

As the heat of respiration must be removed from the product to keep it at constant temperature the planning of air movement in a store for respiring produce is larger and is quite different to conventional stores. Some of the air flow must still be around the walls and doors, but a considerable amount of it must pass through the product. As a result produce is often packaged in the store in packaging with slotted slides or in ventilated cartons. However as air will normally follow the path of least resistance, these will have only a small effect.

Other issues with storing respiring products, include controlling other factors (on top of temperature) that effect metabolic rates and hence product quality and storage life. Controlled atmosphere storage is often used in the fruit industry. In these stores, the levels of carbon dioxide, oxygen and ethylene are controlled at optimum levels to slow down the metabolic rates of the produce to lengthen shelf life in similar means to temperature control. A generation and decontamination system creates the appropriate atmosphere in combination with the refrigeration system and then this is distributed through the same air distribution system. Controlled atmosphere stores are unable to be operated in the same fashion as regular cool stores because of the safety issues associated with modified gas environment.

4 ESTIMATION OF HEAT LOADS

Heat load is defined as the amount of heat that needs to be removed from a facility either to maintain product temperatures (in storage situations) or reduce product temperatures (in chillers or freezers). The amount of heat load determines the size of refrigeration system required and largely defines the capital and operation cost of the facility. There are a variety of factors that contribute to the heat load:

- Product (in freezing and chilling)
- Fans, defrost and lights
- Other mechanical devices (e.g. forklifts, conveyors) and people
- Heat infiltration (through walls, ceiling, floor)
- Air interchange (through doors)
- Cooling of room structures, particularly concrete floors

A typical heat lead breakdown is as follows

	Coldstore/Coolstore	Chiller/Freezer
Product	0-20%	50-75%
Fans	10-25%	10-40%
Heat Infiltration	20-50%	0-15%
Air Interchange	20-50%	0-15%

Some heat loads do not occur at a constant rate, e.g. in freezers when a new batch is introduced the rate at which heat is released to the air from new units will be much larger than the heat release rate from products nearing complete freezing. Hence the average heat load is important, but the peak heat load can also matter if it adds to the overall peak load on the plant, and hence to the plant peak electricity demand.

The general methodology for calculating heat load is as follows:

a) Individual load components are found. Normally average loads are used, but peaks must be also considered.

b) The total loads (average and peak) are found by summation:

Total load = $$\phi_T = \phi_p + \phi_f + \phi_l + \phi_{md} + \phi_{pe} + \phi_i + \phi_a + \phi_s + \phi_d \qquad (15)$$

where
ϕ_p = product load (W)
ϕ_f = fan load (W)
ϕ_l = lighting load (W)
ϕ_{md} = mechanical devices load (W)
ϕ_{pe} = people load (W)
ϕ_i = heat infiltration load (W)
ϕ_a = air infiltration load (W)
ϕ_s = room structure load (W)
ϕ_d = defrost load

c) A load analysis is carried out by plotting the predicted loads versus time on a 24 hour scale. Each load component is plotted separately versus time, and then the summation conducted to get the load versus time profile. The refrigeration plant size must be such that **peak** heat loads can be adequately handled.

This load analysis will illustrate total peak, average and minimum loads of a facility. This load analysis can be used to change operating procedures to ensure that peak loads are minimised and occur during times for cheap electricity.

4.1 THE EQUATIONS

HEAT LOAD	SCENARIO	EQUATION	NOMENCLATURE
Product Load (ϕ_p)	Chilling Average	$\dfrac{m_p c_L(\theta_{in} - \theta_{out})}{t}$	θ_{out} = product out temperature (°C) θ_{in} = product in temperature (°C) m_p = mass of product processed (kg) t = time of the process (s) c_L = specific heat of unfrozen product (J/kg°C)
	Freezing Average	$\dfrac{m_p\lfloor c_L(\theta_{in} - \theta_f) + \Delta h_f + c_S(\theta_f - \theta_{out})\rfloor}{t}$	θ_f = product freezing temperature (°C) c_S = specific heat of frozen product (J/kg°C) Δh_f = enthalpy change in freezing (kJ/kg)
	Respiring Product Storage Average	$m_p R_R$	R_R = respiration rate (W/kg)
Fan Load (ϕ_f)	Average	$N_f W_f \eta_m$ **or** $\dfrac{\Delta PQ}{\eta_m \eta_{fan}}$	N_f = number of fans W_f = nominal fan power (W) ΔP = pressure drop fan is working against (Pa) Q = volumetric flow rate of air (m^3/s) η_m = fan motor efficiency η_{fan} = fan efficiency
Light Load (ϕ_l)	Peak	$N_l W_l$	N_l = number of lights W_l = light wattage (W)
Mechanical Devices (ϕ_{md})	Peak	$N_m W_m$	N_m = number of mechanical devices W_m = mechanical device nominal power (W)
Room Structural Load (ϕ_s)	Average	$\dfrac{m_s c_s(\theta_{st} - \theta_{fal})}{t}$	θ_{st} = structure start temperature (°C) θ_{fal} = structure final temperature (°C) m_s = mass of structure(kg) t = time of the process (s) c_s = specific heat of structure(J/kg°C)

HEAT LOAD	**SCENARIO**	**EQUATION**	**NOMENCLATURE**
People (ϕ_{pe})	Peak	$500N_{pe}$	N_{pe} = number of people
Infiltration Through Building (ϕ_i)	Average	$\frac{kA(\theta_o-\theta_i)}{x}E_i$	x = insulation thickness (m) A = surface area of the wall, roof etc. (m^2) E_i = insulation effectiveness factor θ_o = temperature of the outer surface of the wall, roof, floor or ceiling(°C) θ_i = temperature of the inner surface of the wall, roof, floor or ceiling (°C) k = thermal conductivity of the insulation (W/mK)
Air Infiltration (ϕ_a)	Peak	$\frac{1}{2}A_D v p_{ai}(h_o-h_i)F_D$ **where** $v=5.91\sqrt{\frac{H(1-S)}{\left(1+\sqrt[3]{S}\right)^3}}(1-PF_D)$ **and** $S=\frac{\rho_{ao}}{\rho_{ai}}$	A_D = door cross-sectional area (m^2) v = mean air velocity through the open door (m/s) ρ_{ai} = density of inside air kg/m^3) ρ_{ao} = density of outside air (kg/m^3) h_o = enthalpy of outside air (kJ/kg) h_i = enthalpy of inside air (kJ/kg) F_D = Fraction of time door used during operation time PF_D = Protective factor of protective device H = height of the door (m) S = ratio of warm air to cold air density
Any Peak Load	Daily Average Load	$Peak\times\frac{x_h}{24}$	x_h = time operated per day (hrs)
Defrost Load (ϕ_d)	Average	$0.15\times\sum(all\quad other\quad average\quad loads)$	

4.2 USEFUL DATA

4.2.1 Product Thermal Properties

To perform calculations involving chilling and/or freezing and to calculate product heat loads it is necessary to know the various thermal properties for the product under consideration. This subject was covered in the second year thermal properties of biological materials courses.

4.2.2 Fan Efficiency Values

η_m = fan motor efficiency (typically 0.85 – 0.95)

η_{fan} = fan efficiency (typically 0.55 – 0.70)

4.2.3 Wall/Ceiling/Floor Temperature Assumptions

Internal surface temperatures are approximately the internal operating temperature.
Increments to add to ambient temperatures due to radiation effects to estimate external walls and ceiling temperature are:

	Protected (Weather Sheild)	Not Protected
Walls facing the sun	5	10
Walls not facing the sun	0	0
Roof	10	25

These apply for light coloured surfaces. Double for dark coloured surfaces and half for 24-hour averages.

4.2.4 Air Interchange Protective Devices

- Well installed air curtains reduce interchange by 85 %
- Plastic strips in good condition reduce interchange by 95%
- Forklift traffic increases the calculated interchange on doors with protection by 30% for plastic strips, and 100% for vertical air curtains, but reduces it by 20% for unprotected doors, and by 20% for horizontal air curtains.

4.2.5 Structural Thermal Properties

Thermal Conductivities

Fibreglass	0.040 – 0.050 W/mK
Corkboard	0.035 – 0.040 W/mK
Polyurethane foam	0.022 – 0.030 W/mK
Polystyrene foam	0.028 – 0.035 W/mK

Specific Heat and Density

Concrete	c =	600 - 900 J/kgK;	ρ =	2300 kg/m^3
Steel		500 – 900 J/kgK;		7800 kg/m^3
Aluminium		880 – 900 J/kgK;		2700 kg/m^3

4.2.6 Air Density

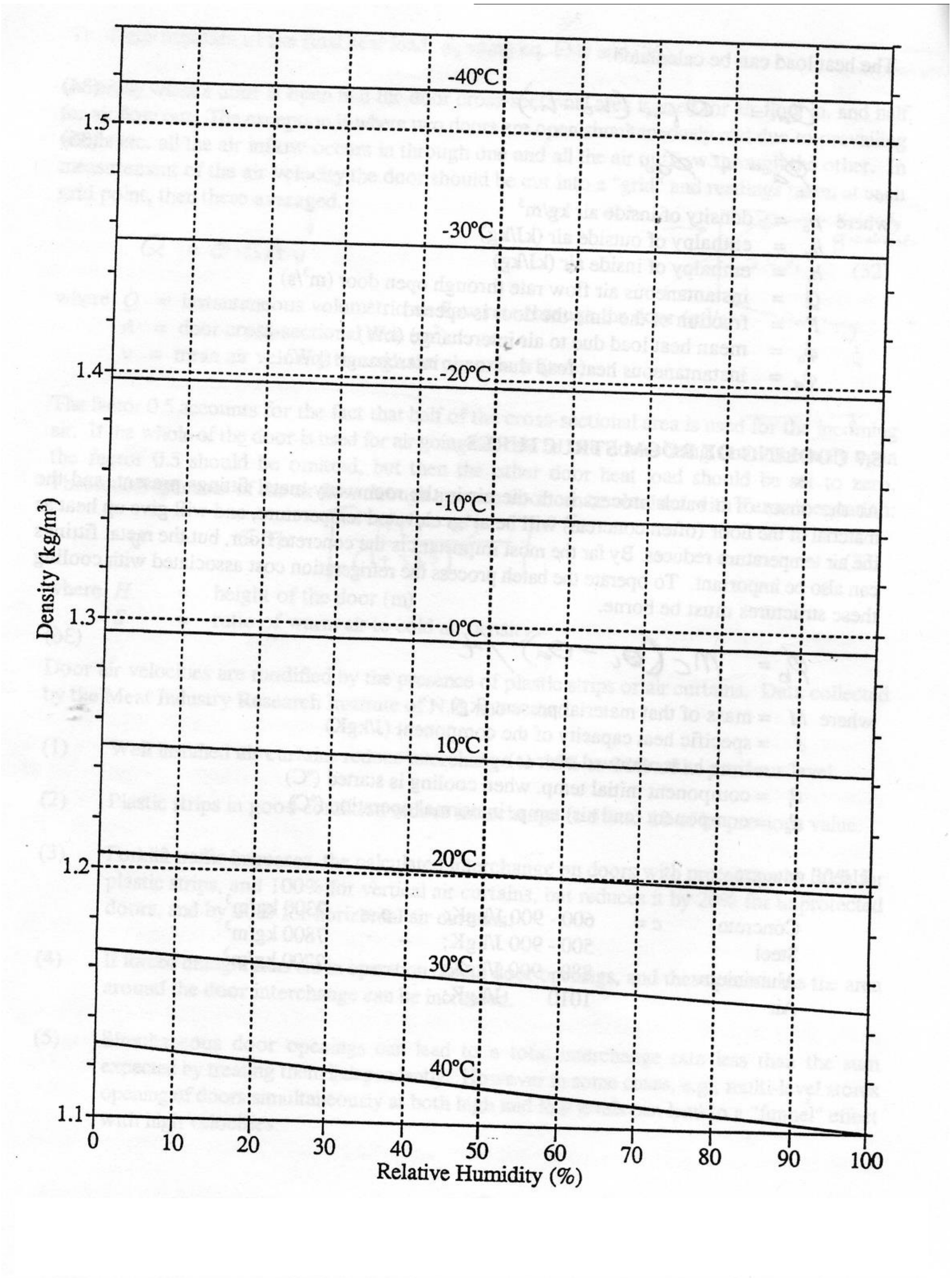


Figure 42, Plot of air density (kg/m^3) as a function of air relative humidity (%) and dry bulb temperature (^{0}C).

4.2.7 Air Enthalpy

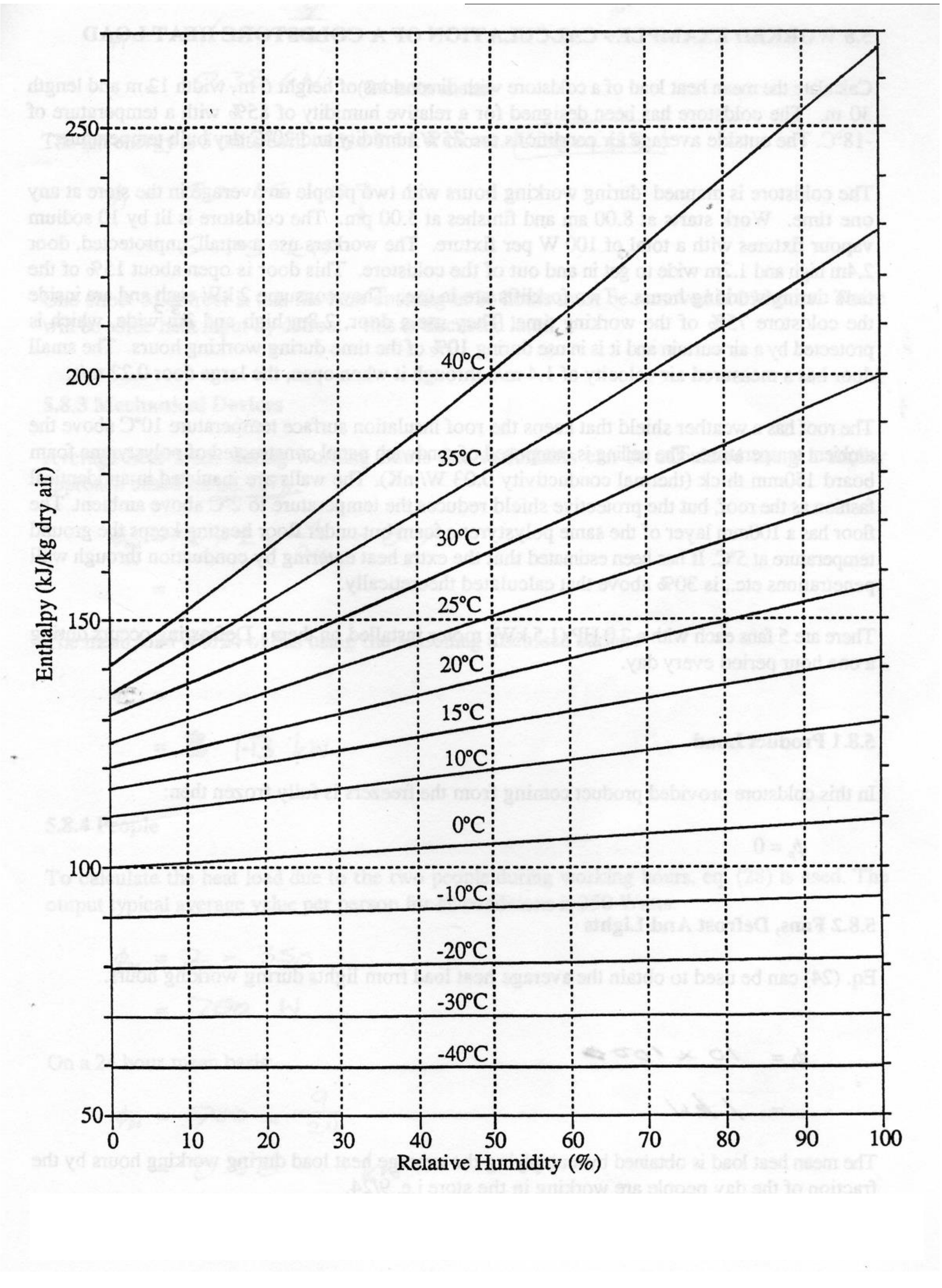


Figure 43, Enthalpy of air (J/kg) as a function of air relative humidity (%) of dry bulb temperature (^{0}C).

4.3 WORKED EXAMPLE – CALCULATION OF A COLDSTORE HEAT LOAD

Calculate the mean heat load of a coldstore with dimensions of height 6m, width 12m length 30m. The coldstore operates at -18°C and is manned during working hours with two people on average in the store at any one time. Work starts at 8.00 am and finishes at 5.00 pm. The coldstore is lit by 10 sodium vapour fixtures with a total of 199 W per fixture. The workers use a small, unprotected, door 2.4m high and 1.2m wide to get in and out of the coldstore. This door is open about 15% of the time during working hours. Two forklifts are in use. They consume 2 kW each and are inside the coldstore 75% of the working item. They use a door, 2.8m high and 3m wide, which is protected by an air curtain and it is in use during 10% of the time during working hours. The small door has a measured air velocity of 1.4 m/s through it when open, the large door 0.22 m/s. The roof has a weather shield that keeps the roof insulation surface temperature 10^0C above the ambient (20°C and 75%RH). The ceiling is composed of sandwich panel constructed of polystyrene foam board 150mm thick (thermal conductivity 0.03 W/mK). The walls are insulated in an identical fashion as the roof, but he protective shield reduces the temperature to 2^0C above ambient. The floor has a 100mm layer of the same polystyrene foam but under floor heating keeps the ground temperature at 5^0C. It has been estimated that the extra heat entering by conduction through wall penetrations etc., is 30% above that calculated theoretically.

There are 5 fans each with a 2.0 HP (1.5 kW) motor installed on them. Defrosting occurs during a one hour period every day.

4.3.1 Scenario Diagram

4.3.2 Product

4.3.3 Fans

4.3.4 Lights

- Peak

- Average

4.3.5 Mechanical Devices

- Peak

- Average

4.3.6 People

- Peak

- Average

4.3.7 Room Structures

4.3.8 Heat Infiltration

- Ceiling

- Floor

- Walls

- Total Heat Infiltration

4.3.9 Air Interchange

- Small Door Peak

- Large Door Peak

- All Doors Peak

- All Doors Average

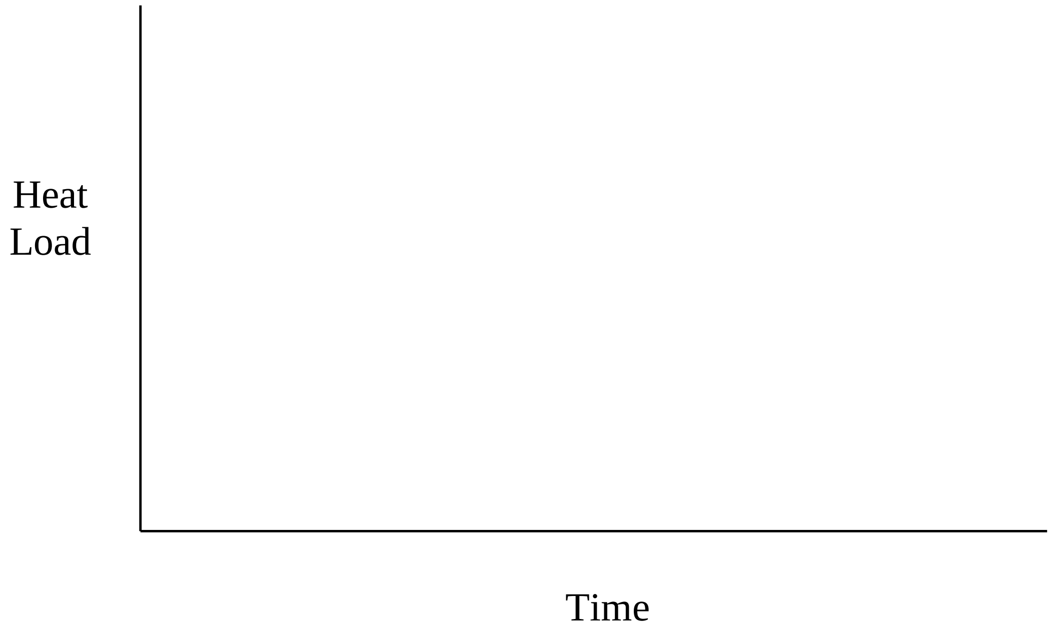
4.3.10 Defrost

4.3.11 Summary

Heat Sources	Operational Load (kW)	Night Time Load (kW)	24 hour Mean Load (kW)
Product			
Lights			
Fans			
Forklifts			
People			
Structure cooling			
Heat infiltration			
Air interchange			
Defrost			
TOTAL			

Note that the cooling system will only operate 23 hours per day, but would be sized to cope with the maximum demand, and so in 23 hours at full capacity it has plenty of spare capacity to remove the total heat load coming in over 24 hours.

4.3.12 Load Analysis



5 FREEZING TIME PREDICTION

The factor most limiting the throughput of a freezer is the product residence time needed for freezing. Reducing this time allows better use of the invested capital. Heat load is also consequential on freezing time, (consider the equation used to estimate product heat load).

Freezing times depend on the following factors:

- product size, measured as the half-thickness of product (use "radius" from "thermal centre" to surface on shortest route), R(m)
- shape of the product, indexed by the equivalent heat transfer dimensionality, E
- thermal properties of the product,
    - Thermal conductivity of unfrozen material k_l (W/mK)
    - Thermal conductivity of frozen material k_s (W/mK)
    - Specific heat capacity of unfrozen material c_l (J/kgK)
    - Specific heat capacity of frozen material c_s (J/kgK)
    - Density ρ (kg/m^3)
    - Apparent latent heat of freezing Δh_f (J/kg)
- cooling medium temperature, θ_a (°C)
- product initial temperature, θ_{in} (°C)
- required final product temperature, θ_{out} (°C)
- surface heat transfer coefficient, h_e (W/m2K)

Of these various factors, only the cooling medium temperature and the surface heat transfer coefficient are normally "available" to the engineer to try to change the freezing time.

Lower θ_a implies:

- shorter freezing time
- higher refrigeration costs (due to lower evaporation temperature)

Higher h_e implies:

- higher air velocity
- shorter freezing time (depending on the extent to which heat transfer within the product is limiting)
- possibly higher costs (e.g. increased fan power and hence refrigeration load leading to higher energy costs)

It would thus be useful to be able to predict freezing times accurately, and the methodology should allow the effect of changing h_e or θ_a to be evaluated.

In situations where the conditions (θ_a or h_e) change during freezing the calculation of freezing time is less straightforward. An example would be where the object is crust frozen in one medium, and then freezing completed in another. There are techniques to cover these situations, but they are too detailed to be covered here. Similarly, uneven heat transfer on different faces of a container can be handled, but the detail is beyond this course. For small changes it is quite reasonable to use average values of θ_a and h_e.

5.1 PLANKS EQUATION

In second year you were introduced to Plank's equation, which was an early attempt to derive a method for freezing time prediction. It makes several assumptions that are not true in practice:

- the product is initially at its freezing temperature,
- the specific heat capacity of the frozen material is 0,
- all latent heat is released at a unique temperature,
- all thermal properties are constant,

As a result it tends to underestimate freezing times by (typically) 30%.

The physical mode (Figure 44) is of a moving phase change front at which heat is released, and conduction of this heat through a frozen material with thermal resistance but no thermal capacity.

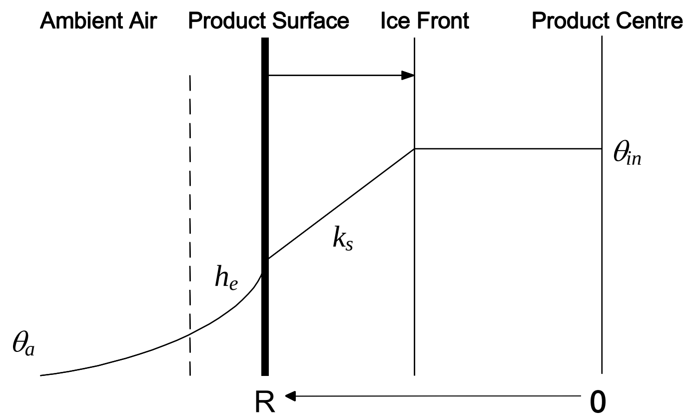


Figure 44, A Physical Model of Planks Equation for a Slab

5.2 PHAMS MODIFICATION TO PLANK'S EQUATION

A variety of modifications to Plank's equation have been suggested that overcome the limitations – there are at least 3 or 4 that perform quite well. We will cover just one of these. The modification to Plank's equation that we will use was derived by:

Pham, T.T. (1986). Simplified equation for predicting the freezing time of foodstuffs. *J. Food Technol.,* **21**, 209-219.

This method is generally accurate to within ± 10%. The main differences are addition of a shape factor and reforming of the temperature difference and heat to be removed terms.

Phams equation also provides advantages in that the numerical values in the right hand bracket of the equation can indicate reasons for slow freezing times. The value of R/h_e is the so-called <u>external</u> resistance to heat transfer whereas $R^2/2k_s$ is the so-called <u>internal</u> resistance (heat transfer resistance within the product itself). In the case that there is more external heat transfer, changing h_e will change the freezing time significantly. Looking at the contributing heat transfer resistances (in the equation to determine h_e) can be used to decide where effort is best applied when trying to change freezing time.

5.3 FINITE DIFFERNCES OR FINITE ELEMENTS

In practice when a biological material is frozen the latent heat of freezing is released over a range of temperature. It is therefore mathematically possible to treat the process as solely heat conduction. The process of freezing is taken into account by using temperature variable thermal conductivity and specific heat capacity. Typical shapes for the shape of the relationships of these thermal properties to temperature are shown in Figure 45.

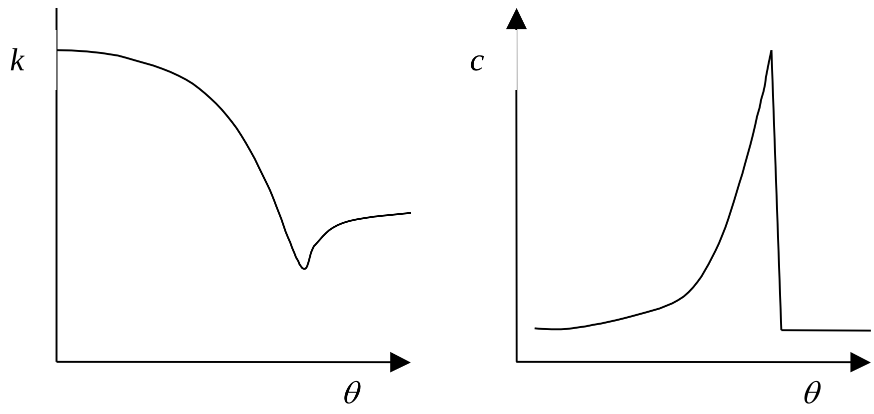


Figure 45, Typical Shapes for Food Thermal Properties

The area under the peak in the specific heat versus temperature curve is the enthalpy change due to freezing.

Because the thermal properties are so temperature dependent the techniques for calculation of the heat transfer processes that have been covered in previous lecture courses are no longer appropriate. Hence alternative mathematical methods are used. Those most valuable are finite differences and finite elements. These are numerical methods that solve the basic equations of heat conduction, but with variable thermal properties.

As an example consider a slab. For finite differences to be applied the slab is divided into elements of thickness Δx. Normally about 10 Δx's across a slab from the surface to the centre would be used. Time is divided into elements of Δt seconds. At time zero the temperature at each of the "nodes" between elements is set to the initial temperature. The finite difference method then uses heat conduction calculations to calculate the temperatures at each of the nodes Δt seconds into the process.

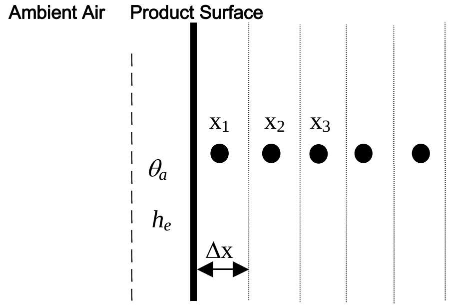


Figure 46, Finite Difference Model for a Slab

Hence, over a number of steps the temperature/time history at each of the nodes in the slab is predicted. Once the temperature at the centre reaches the temperature at which freezing is deemed complete the time that the integration is up to is taken as the freezing time. Typically about 1000 steps are used, so a computer is used to carry out the calculations.

You will have the opportunity to run a finite difference program in your freezing experiment. Finite difference programs exist for all the regular shapes (slabs, cylinders, spheres, rods, bricks). Engineering options will cover the methodology of creating finite difference models in fourth year.

5.4 THE EQUATIONS

<table>
<tr><th>PREDICTION METHOD</th><th>TERM</th><th>EQUATION</th><th>NOMENCLATURE</th></tr>
<tr><td>Planks</td><td>Freezing Time (t_f)</td><td>$\frac{\rho \Delta h_f}{(\theta_f - \theta_a)}\left(\frac{R}{h_e} + \frac{R^2}{2k_s}\right)$</td><td>$h_e$ = surface heat transfer coefficient (W/m^{2}K)
k_s = thermal conductivity of frozen material (W/mK)
R = product size, measured as the “shortest” half-thickness of product (m)
θ_a = cooling medium temperature (°C)
θ_f = product freezing temperature (°C)
Δh_f = apparent latent heat of freezing (J/kg)
ρ = density of product (kg/m^3)
t_f = freezing time(s)</td></tr>
<tr><td rowspan="6">Phams Modification to Planks</td><td>Freezing Time (t_f)</td><td>$\frac{1}{E}\left(\frac{\Delta H_1}{\Delta \theta_1} + \frac{\Delta H_2}{\Delta \theta_2}\right)\left(\frac{R}{h_e} + \frac{R^2}{2k_s}\right)$</td><td>$E$ = shape factor
ΔH_1 = heat released in precooling (J/m^3)
ΔH_2 = heat released in freezing (J/m^3)
$\Delta \theta_1$ = temperature driving force for precooling (K)
$\Delta \theta_2$ = temperature driving force for freezing (K)</td></tr>
<tr><td>ΔH_1</td><td>$\rho c_l(\theta_{in} - \theta_{fm})$</td><td>$\theta_{fm}$ = mean freezing temperature (°C)
θ_{in} = product initial temperature (°C)
c_l = specific heat capacity of unfrozen material (J/kgK)</td></tr>
<tr><td>ΔH_2</td><td>$\rho c_s(\theta_{fm} - \theta_{out}) + \rho \Delta h_f$</td><td>$c_s$ = specific heat capacity of frozen material (J/kgK)
θ_{out} = target final temperature (°C)</td></tr>
<tr><td>$\Delta \theta_1$</td><td>$\frac{(\theta_{in} + \theta_{fm})}{2} - \theta_a$</td><td></td></tr>
<tr><td>$\Delta \theta_2$</td><td>$\theta_{fm} - \theta_a$</td><td></td></tr>
<tr><td>θ_{fm}</td><td>$1.8 + 0.263\theta_{out} + 0.105\theta_a$</td><td></td></tr>
</table>

PREDICTION METHOD	TERM	EQUATION	NOMENCLATURE
Phams Modification to Planks	Shape Factor (E)	$1+\dfrac{1+\dfrac{2}{Bi}}{\beta_1^2+\dfrac{2\beta_1}{Bi}}+\dfrac{1+\dfrac{2}{Bi}}{\beta_2^2+\dfrac{2\beta_2}{Bi}}$	Bi = Biot number
	Biot number (Bi)	$\dfrac{h_e R}{k_s}$	
	β_1	$\dfrac{b}{a}$ **OR** $\dfrac{A}{\pi R^2}$	a = shortest half thickness of product through the thermal centre (m) b = second shortest half thickness of product perpendicular to a. and through thermal centre (m) A = smallest cross-sectional area of product through thermal centre (m^2)
	β_2	$\dfrac{c}{a}$ **OR** $\dfrac{3V}{4\pi R^3 \beta_1}$	c = longest half thickness of product perpendicular to both a and b. and through thermal centre (m) V = volume of product (m^3)
Both Planks and Phams Modification	Surface Heat Transfer Co-efficient (h_e)	$\dfrac{1}{h_e}=\dfrac{1}{h_a}+\dfrac{x_p}{k_p}+\dfrac{x_a}{k_a}$	h_a = heat transfer coefficient form cooling fluid (W/m^2K) x_P = packaging thickness (m) k_p = packaging thermal conductivity (W/mK) x_a = thickness of air film trapped by packaging (m) k_a = thermal conductivity of air (W/mK)
	Air Heat Transfer Co-efficient (h_a)	For large oval objects (R > 0.05m): $12.5v^{0.6}$ For large planar objects (R > 0.05m): $7.3v^{0.8}$	v = air velocity (>0.25m/s) (m/s)

5.5 USEFUL DATA

5.5.1 Product Thermal Properties

If thermal properties of the material undergoing freezing cannot be found in data books you should use the methods covered in the second year biological properties course to estimate appropriate values. Particular care must be taken in finding values for k_s, and c_s and Δh_f. You will remember from your second year course that the ice fraction changes with temperature below the initial freezing temperature – this means that k_s and c_s change with temperature as well. A convention that is normally used for finding single values to represent the complex real situation is to assume that values of k and c evaluated at about $–20^0$C represent the frozen phase values k_s and c_s.

Note that Δh_f is the equivalent of a "latent heat". It is obtained by subtracting the sensible heat fraction from an enthalpy change between the initial freezing temperature and a lower temperature such as $–18^0$C. e.g. if the specific heat capacity of the fully frozen material (c_s) and the enthalpy change from θ_f to $–18^0$C (Δh_{18}) are known then:

$$\Delta h_f = \Delta h_{18} - c_s\left(\theta_f - {}^{-}18\right)$$

5.5.2 Packaging Thermal Conductivity Values

Typical values of k_p are:

Plastics	0.08 to 0.15 W/mK
Solid cardboard	0.06 to 0.10 W/mK
Corrugated cardboard	0.04 to 0.06 W/mK
Stagnant air	0.025 W/mK

5.5.3 Heat Transfer Co-efficient Values

Typical values of h_a are:

Plate freezers (good contact)	500 W/m^{2}K
Plate freezers (poor contact	100 W/m^{2}K
Moving liquid	500 W/m^{2}K
Air natural convection	4-7 W/m^{2}K
Fluidised bed freezing	150 W/m^{2}K
Air flow through packed beds	15-60 W/m^{2}K

5.5.4 Shape Factors

The "equivalent heat transfer dimensionality", *E* is a measure of how much each of the three space dimensions, x, y and z contribute to the heat transfer that brings about product temperature change. *E* values are always between 1 and 3.

Sphere	$E = 3$
Infinitely long cylinders	$E = 2$
Infinite slabs	$E = 1$
Lamb leg	$E = 2.1$
Lamb shoulder	$E = 1.4$
Ewe leg	$E = 2.0$
Beef leg	$E = 1.3$
Tuna	$E = 1.4$

For 2-dimensional shapes β_2 is assumed to be infinitely large.

5.6 WORKED EXAMPLE

Cartons of boneless beef containing 27 kg have dimensions of about 165mm x 360 mm x 530 mm. The product starts at 10^0C and is frozen to -18^0C. A 48 hours cycle carton freezer using air at -22^0C and an average air velocity of 2 m/s is used. The cartons presently used are 2.5 mm thick solid walls. As the meat will not contact the sides, top and bottom of the carton perfectly, assume that there is a 1 mm air gap as well as the packaging that is a plastic liner as well as the carton. Will the carton freezer work?

In the calculation first we need thermal properties of the boneless beef. These were looked up in data tables:

c_l	=	3600 J/kgK
c_s	=	1900 J/kgK
k_s	=	1.50 W/mK
ρ	=	1060 kg/m^3
Δh_f	=	215,000 J/kg

5.6.1 Scenario Picture

5.6.2 Planks Equation

- cooling fluid heat transfer co-efficient

- heat transfer co-efficient

- **Planks Equation**

5.6.3 Phams Modification to Planks Equation

- Shape factor E

- Smallest cross-sectional area of product through the thermal centre

- β_1

- Volume

- β_2

- Bi

- **Shape Factor**

- θ_{fm}

- ΔH_1

- ΔH_2

- $\Delta \theta_1$

- $\Delta \theta_2$

- **Phams Modification to Planks Equation**

6 CHILLNG TIME PREDICTION

As was the case with freezers, the factor most limiting the throughput of chillers is the chilling time. In second year you covered methods to predict chilling and heating times of regular shapes (slabs, cylinders, spheres, rectangular rods, finite cylinders, bricks). In this section an approximate method for predicting the chilling times of irregular shapes is presented so that you have methodology that can be applied to all shapes. Many of the data required are the same as used for freezing:

- product size, measured as the half-thickness of product (use "radius" from "thermal centre" to surface on shortest route), R (m)
- thermal properties of the unfrozen product:
  - Thermal conductivity k_l (W/mK)
  - Specific heat capacity c_l (J/kgK)
  - Density ρ (kg/m3)
- cooling medium temperature, θ_a (°C)
- product initial temperature, θ_{in} (°C)
- required final product centre temperature, θ_c (°C)
- surface heat transfer coefficient, h_e (W/m2K)
- equivalent heat transfer dimensionality E. This has different numerical values to freezing.

6.1 MODEL PHYSICAL INTERPRETATION

The model can be conceptualised as shown in Figure 47.

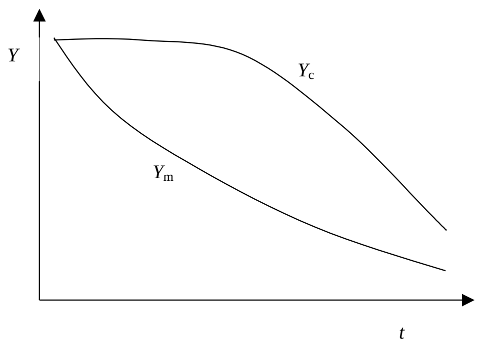


Figure 47, Conceptual Model of Chilling Time Prediction Technique

After an initial period there is a linear relationship between *ln Y* and time as shown in Figure 48.

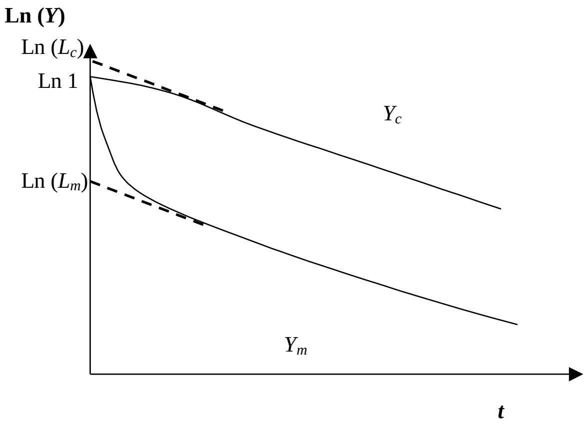


Figure 48, Conceptual Model of Chilling Time Prediction Technique

6.2 THE METHOD

The various calculation steps are given below. As expressed, the method applies to all shapes. The source is:

Lin, Z., Cleland, A.C., Cleland, D. J. and Serrallach, G. F. (1996). A simple method for prediction of chilling times: extension to three-dimensional shapes. *International Journal Refrigeration*: **19**; 107-114.

1. Measurement of the object dimensions D_1, D_2 and D_3:
   - D_1 = shortest dimension through the geometric centre of the object (first dimension):
   - D_2 = shortest dimension through geometric centre of the object taken at right angles to the first dimension (second dimension);
   - D_3 = longest dimension through geometric centre of the object taken as close as possible to right angles to both the first and second dimensions (third dimension).

   For irregular shapes this implies approximation of the real shape to an equivalent infinite ellipse or ellipsoid. In using this measurement technique any obvious protrusions are ignored.

2. Calculation of characteristic dimension R and two dimension R and two dimensional ratios β_1 and β_2:

$$R = \frac{D_1}{2} \qquad \beta_1 = \frac{D_2}{D_1} \qquad \beta_2 = \frac{D_3}{D_1}$$

3. Calculation of the Biot number (Bi).

$$Bi = \frac{h_e R}{k_L}$$

4. Calculation of the equivalent heat transfer dimensionality at $Bi = 0$ (E_o) using:
   (i) Ellipsoid or three-dimensional irregular shapes:

$$E_o = \frac{3\left[\beta_1 + \beta_2 + \beta_1^2(1+\beta_2) + \beta_2^2(1+\beta_1)\right]}{2\beta_1\beta_2(1+\beta_1+\beta_2)} - \frac{\left[(\beta_1-\beta_2)^2\right]^{0.4}}{15}$$

(ii) Ellipse or two-dimensional irregular shapes:

$$E_o = \left(1+\frac{1}{\beta_1}\right)\left(1+\left(\frac{\beta_1-1}{2\beta_1+1}\right)^2\right)$$

(iii) Finite cylinders, bricks, infinite rectangular rods:

$$E_o = 1+\frac{1}{\beta_1}+\frac{1}{\beta_2}$$

(iv) sphere (E_0 = 3), infinite cylinder (E_0 = 2), infinite slab (E_0 = 1);

5. Calculation of the equivalent heat transfer dimensionality at $Bi = \infty$ (E_∞) using

$$E_\infty = 0.75 + P_1 f(\beta_1) + P_2 f(\beta_2)$$

where

$$f(\beta) = \frac{1}{\beta^2} + 0.01P_3\exp\left(\beta - \frac{\beta^2}{6}\right)$$

values of P_1 and P_2 are found in Table 3.

6. Calculation of E:

$$E = \frac{Bi^{\frac{4}{3}} + 1.85}{\left(\frac{Bi^{\frac{4}{3}}}{E_\infty} + \frac{1.85}{E_o}\right)}$$

7. Calculation of a so-called lag factor L at $Bi = \infty$ (designated L_∞):

$$L_\infty = 1.271 + 0.305\exp\left(0.172\gamma_1 - 0.115\gamma_1^2\right) + 0.425\exp\left(0.09\gamma_2 - 0.128\gamma_2^2\right)$$

where values of λ are given in Table 3.

8. Calculation of the lag factor for the object thermal centre position (L_c):

$$L_c = \frac{Bi^{1.35} + \frac{1}{\lambda}}{\left(\frac{Bi^{1.35}}{L_\infty} + \frac{1}{\lambda}\right)}$$

where values of λ are given in Table 3.

Shape	N	P_1	P_2	P_3	γ_1	γ_2	λ
Infinite slab ($\beta_1 = \beta_2 = \infty$)	1	0	0	0	∞	∞	1
Inf, rectangular rod ($\beta_1 \geq 1$, $\beta_2 = \infty$	2	0.75	0	-1	$4\beta_1/\pi$	∞	γ_1
Brick ($\beta_1 \geq 1$, $\beta_2 \geq \beta_1$)	3	0.75	0.75	-1	$4\,\beta_1/\pi$	$1.5\beta_2$	γ_1
Infinite cylinder ($\beta_1 = \beta_2 = \infty$)	2	1.01	0	0	1	∞	1
Infinite ellipse ($\beta_1 > 1$, $\beta_2 = \infty$)	2	1.01	0	1	β_1	∞	γ_1
Squat cylinder ($\beta_1 = \beta_2$, $\beta_1 \geq 1$)	3	1.01	0.75	-1	$1.225\beta_1$	$1.225\beta_1$	γ_1
Short cylinder ($\beta_1 = 1$, $\beta_2 \geq 1$)	3	1.01	0.75	-1	β_1	$1.5\beta_2$	γ_1
Sphere ($\beta_1 = \beta_2 = 1$	3	1.01	1.24	0	1	1	1
Ellipsoid ($\beta_1 \geq 1$, $\beta_2 \geq \beta_1$)	3	1.01	1.24	1	β_1	β_2	γ_1

Table 3, Values of geometric parameters N, P_1, P_2, P_3, γ_1, γ_2 and λ for a variety of shapes

9. Calculation of the lag factor for the mass-average temperature of the object (L_m)

$$L_m = \mu L_c$$

where

$$\mu = \left(\frac{1.5 + 0.69Bi}{1.5 + Bi}\right)^N$$

N is the number of dimensions of an object in which heat is significant; values are stated in Table 3.

10. Calculation of the first root of the transcendental equation for a sphere (α)

$$\alpha \cot \alpha + Bi - 1 = 0$$

Note that $0 \leq \alpha \geq 3.14159$. Table 4 gives some numerical values.

Bi	α	*Bi*	α	*Bi*	α	*Bi*	α
0.01	0.173	0.60	1.264	1.80	1.959	8.50	2.786
0.02	0.244	0.65	1.310	2.00	2.029	9.00	2.804
0.03	0.299	0.70	1.353	2.20	2.092	9.50	2.821
0.04	0.345	0.75	1.393	2.40	2.148	10.00	2.836
0.05	0.385	0.80	1.432	2.60	2.200	11.00	2.863
0.06	0.422	0.85	1.469	2.80	2.246	12.00	2.885
0.08	0.486	0.90	1.504	3.00	2.289	13.00	2.904
0.10	0.542	0.95	1.538	3.20	2.328	14.00	2.921
0.12	0.593	1.00	1.571	3.40	2.364	15.00	2.935
0.14	0.639	1.05	1.602	3.60	2.397	16.00	2.948
0.16	0.682	1.10	1.632	3.80	2.427	18.00	2.969
0.18	0.722	1.15	1.661	4.00	2.456	20.00	2.986
0.20	0.759	1.20	1.689	4.50	2.518	25.00	3.017
0.25	0.845	1.25	1.716	5.00	2.570	30.00	3.037
0.30	0.921	1.30	1.741	5.50	2.615	35.00	3.052
0.35	0.990	1.35	1.766	6.00	2.654	40.00	3.063
0.40	1.053	1.40	1.791	6.50	2.687	50.00	3.079
0.45	1.111	1.45	1.814	7.00	2.717	60.00	3.089
0.50	1.16	1.50	1.837	7.50	2.742	80.00	3.102
0.55	1.27	1.60	1.880	8.00	2.765	100.00	3.110

Table 4, Values of the first root of the transcendental equation for a sphere (α)

11. Calculation of the chilling time to reach achieve a desired temperature (t_c or t_m) or temperature reached after a defined chilling time (θ_c or θ_m):

For thermal centre:

$$t_c = \frac{3\rho c_L R^2}{\alpha^2 k_L E} \ln\left(\frac{\theta_{in} - \theta_a}{\theta_c - \theta_a} L_c \right)$$

or

$$\theta_c = L_c \exp\left(\frac{-k_L t_c E \alpha^2}{3\rho c_L R^2} \right)(\theta_{in} - \theta_a) + \theta_a$$

For mass-average:

$$t_m = \frac{3\rho c_L R^2}{\alpha^2 k_L E} \ln\left(\frac{\theta_{in} - \theta_a}{\theta_m - \theta_a} L_m \right)$$

or

$$\theta_m = L_m \exp\left(\frac{-k_L t_m E \alpha^2}{3\rho c_L R^2} \right)(\theta_{in} - \theta_a) + \theta_a$$

where θ_m = mass-average temperature (^{0}C)

12. Checking of the range of fractional unaccomplished temperature change (Y). If $Y_c > 0.7$ or $Y_m > 0.55$, results calculated using the prediction method may be unreliable.

$$Y_c = \frac{\theta_c - \theta_a}{\theta_{in} - \theta_a}$$

$$Y_m = \frac{\theta_m - \theta_a}{\theta_{in} - \theta_a}$$

This method is approximate only but generally gives chilling time estimates within ±10% provided accurate data are used – any data error, particularly the need to average time-variable conditions, or uncertainties in heat transfer coefficients increases the error significantly.

As a general principle for regular shapes (slabs, cylinders, spheres, finite cylinders, rectangular rods, bricks) you should use the Schack and Gurney – Luries charts covered in second year for greatest accuracy and restrict the above approximate method to irregular shapes.

6.3 WORKED EXAMPLE

A 125 kg side of lean beef is being chilled to a mass-average temperature of 8°C from an initial temperature of 40°C using air at 4°C and a velocity of 2 m/s. The critical area is the deep leg position (hip bone). After excursion of the leg and the body through the rib cage the measured dimensions are 194 mm x 380 mm x 610 mm.

Thermal properties of beef are:

c_L = 3400J/kgK
ρ = 1030 kg/m^3
k_L = 0.46 W/mK

We can follow the calculations routinely:
Picture:

(1) Evaluate D_1, D_2 and D_3:

= = =

(2) Evaluate R, β_1 and β_2

= = =

(3) Evaluate $Bi = h_e R/k_L$. First find h_e for large oval shapes.

Bi =

(4) Calculation of E_0.

E_0 ≈

≈

≈

(5) Calculation of E_∞

P_1 =

P_2 =

P_3 =

$f($ $)$ =

=

=

$f($ $)$ =

=

=

E_∞ =

=

(6) Calculation of E

E =

=

(7) Calculation of L_∞

γ_1 =

γ_2 =

L_∞ =

=

=

=

(8) Calculation of lag factor for thermal centre position

λ =

L_c =

=

(9) Calculation of L_m

N =

μ =

and L_m =

=

(10) Determination of α.

α =

You might check this by substitution into the equation.

(11) Calculation of the chilling time to be found:

Noting temperatures: θ =

θ_{in} =

θ_m =

t_m =

=

=

(12) Check Y_m

Y_m =

Lastly, we might want to know the value of θ_c at the completion of chilling.

θ_c =

=

=