**Technological options for retail refrigeration**

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# Review of display refrigeration technologies

A large number of technologies are available to reduce the carbon footprint of retail refrigeration systems. These can either save carbon through reducing energy usage (indirect effects) and/or through reducing refrigerant leakage (direct effects). The work carried out in this publication covers available carbon saving technologies and evaluates their potential in supermarket refrigeration systems. The aim of the work was to review and assess each technology from published information and to provide the reader with an unbiased view on the potential use and benefits of each technology. In many cases, varied benefits were observed for the technologies which greatly depended on the specific method of application. Also, the evidence presented by authors of the documents reviewed varied in their depth and level of robustness. In all cases, the authors have attempted to provide a balanced overview of each technology and to quantify evidence of the level of savings that can be achieved.

All technologies are listed in alphabetical order.

## Adiabatic condensers

Adiabatic condensers operate by spraying water into the air supply of air cooled condensers. The water is often sprayed on a pad through which the air flows. This has the effect of cooling the air (due to evaporation of the water droplets) that cools the condenser, reducing condensing temperature and pressure, saving compressor energy and increasing refrigeration capacity. This has the greatest benefit in the summer months when condensing temperatures are high. During the winter there may be no benefits. The cooling effect is related to the relative humidity (RH), so the benefit is higher in dry weather (low RH) and there will be no benefit when RH is 100%. Therefore, the spray systems are designed to work for only part of the year. As soon as air with 100% RH is warmed up, its relative humidity drops below 100%, allowing for the application of the evaporative cooling. Therefore, in the instances where the condenser is divided into desuperheating, condensing and subcooling sections, the water spray can be applied between the desuperheating and condensing sections when the RH of the air is below 100%. Thus, the benefit from the evaporative cooling can be achieved even in areas where the RH of the ambient air is 100%.

With a perfectly efficient adiabatic condenser, the air will cool from the dry bulb to the wet bulb temperature. Baltimore Aircoil Company (2015) states that cooling temperature on the condenser can be reduced by approximately 1 to 2ºC above the wet bulb temperature.

Unlike evaporative condensers, all the water should be evaporated and therefore none needs to be recycled, which avoids the requirement for treatment. The need to make sure that all water is evaporated requires correct control of water temperature and air flow rate, which will be dependent on ambient conditions. The quantity of water used is much less than that used by evaporative condensers.

### Problems

Water evaporation can lead to scaling. Another problem is the dust and other contaminants in the supply air; they are more likely to pass through a dry condenser than through a condenser with a wet surface, contributing to a faster clogging of the condenser.

### Legislation

There is no direct legislation for adiabatic condensers, unlike evaporative condensers. However, the safety of each system should be considered, especially in relation to *Legionella*.

### Case studies

Hill Phoenix has installed 20 Advansor transcritical CO2 systems in Canada (Wallace, 2016). Using adiabatic gas coolers, they have been able to keep the operating pressures lower for longer in the warmer climates. However, actual savings have not been reported.

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## Air deflectors/guides

The use of guides or deflectors on open fronted cabinets has the potential to reduce air infiltration and consequently the energy consumption of the cabinet refrigeration systems. The design of the deflectors is thought to influence the levels of energy savings achieved. Work to evaluate the impact of aerofoil deflectors (Foster, McAndrew and Evans, 2014) has shown overall energy savings. Adding the aerofoil deflectors reduced maximum temperatures in a cabinet tested to the EN23953 test standard (at climate class 3) by 0.6°C as well as reducing the energy consumption by 15%. When the cabinet controller was adjusted so that the maximum pack temperatures were the same for the two tests, the air deflectors reduced energy consumption by 17%. It should be noted that open fronted cabinets do not have the same air curtain geometry and therefore different results may be seen when the technology is applied to different cabinets. It may be required to optimise the aerofoils, either by their distance from the end of the shelf and/or their orientation to the air flow. The technology for the aerofoil deflectors is being trialled in the UK by a company called Aerofoil Energy Ltd which has a patent for the air guides (The Grocer, 2014).

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## Anti-fogging glass

Fogging of glass reduces the visibility of products on display, making it hard for shoppers to find the product they are looking for and potentially preventing impulse purchases and reducing sales.

Electric heaters can be used to maintain the glass surface above the dew point or to accelerate the removal of fog once formed. Typically fog is only formed as the cold (cabinet facing) side of the door is exposed to the supermarket ambient environment upon opening the door. However, in more humid conditions, fogging or condensation can be issues on the outer side of chilled and frozen cabinets even without opening the doors.

Anti-fogging glass is typically a standard glass with a film bonded to the surface. Anti-fog coatings have hydrophilic properties such that any water on the surface does not bead up and create fog.

Anti-fog coatings have historically been prone to degradation through cleaning or contact with skin (people holding doors open). However, Dixell (Dixell (Asia) Co., Ltd., 2010) present a case study for their anti-fogging product, claiming longer durability: “Water and cleaning products will not wear it down, although oil or grease will eventually ruin it.” Anti-fogging films can be implemented in conjunction with electrical heat but laboratory testing has shown that some anti-fog films can delaminate if the cases are allowed to return to room temperature. This is believed to be due to moisture migrating under the coating and then freezing and separating the coating from the glass surface. Delamination could obscure the view of the products and be worse than fogging. Some anti-fog coatings may lose their hydrophilic properties if washed or soaked in water (Rauss et al., 2008). Delamination needs to be solved before widespread implementation.

### Case studies

Price Chopper trialled Lexan Constant Clear film (Sabic, 2014) on a five-door freezer case. After fitting the film, the heat to the glass doors was turned off. Price Chopper then tracked and compared the energy usage over a period of two years. By turning off the anti-fogging door heater, the electrical energy usage for the five-door case was reduced by 400 W. The investment was predicted to be recouped in 10 months, depending on the store.

Dixell trialled anti-fogging film under test room conditions and concluded that, ‘…with the exception of delamination, anti-fogging films could prevent fog and sweat from accumulating on the door.’ Dixell measured energy savings of 35% in a freezer by turning off the door heater on a two door freezer cabinet operating in a 24.6°C / 48% RH environment (Dixell (Asia) Co., Ltd., 2010). It is not clear how much of this saving was directly caused by the heater and how much was due to the reduced heat load to the refrigeration system.

### Energy saving potential

Although electric resistance heaters are not commonly installed on chilled cabinets in UK supermarkets, frozen glass display cabinets may have three types of electric resistance heaters installed (case mullion, door frame and glass pane). These range from 100 to 200 W per door (Rauss, Mitchell and Faramarzi, 2008). Anti-Fog Systems LLC (2016) quote the following: “The results demonstrated that the freezer unit reduced electricity consumption by 47% when the film was applied and the anti-sweat heater was disconnected.” Dixell (2010) showed a reduction in energy use of 35% after applying anti-fogging film to a freezer cabinet. It should be noted that not all freezer cabinets have all of these heaters, so the benefits will vary depending on the cabinet.

### Cost

Payback for retrofit or new construction is expected to be 10 months (Sabic Global, 2014) to 2 years (Washington State University).

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## Anti-sweat heater control

When glass doors were first introduced, condensation and fogging were major problems, particularly on frozen cases. Although much of the published work for chilled cabinets with glass doors still refers to significant energy loads from anti-sweat heaters, field experience and anecdotal evidence from manufacturers suggest that anti-sweat heaters are not common on chilled cabinets (in the UK at least).

Anti-sweat heaters can be used to heat (and so prevent the misting of) glass fronts of refrigerated cabinets where the glass would otherwise be at a temperature significantly below the dew point of the ambient air. Similar heaters can also be used in the trim of low temperature (freezer) cabinets to prevent condensation (sweating) on the outside of the cabinet.

The amount of heat required by an anti-sweat heater to maintain a clear door will be a function of the cabinet design (which influences the rate of cooling or clearing of the mist through internal air condition and movement) and the ambient air in the supermarket.

Where the ambient air dew point is lower than the coldest surface temperature of the glass, there is no need for anti-sweat heaters.

There have been several studies investigating ‘cold aisle’ problems in supermarkets. Data recorded (in the autumn) in a range of UK supermarkets (and measured at 600 mm from the floor and 600 mm from the cold cabinet front) showed that the refrigerated aisles (incl. glass door and frozen food) typically had a dew point of 4°C while the rest of the store had a dew point of 5°C. It therefore follows that trim heaters would not normally be required for chilled cabinets in UK supermarkets. From this data it is clear that the conditions experienced in-store and in a test room are significantly different. The humidity level in-store is considerably less than in the test room; a dew point of approximately 4.5°C compared to 16°C. Significant potential, therefore, exists for energy saving where anti-sweat heaters are fitted to chilled cabinets and currently operate continuously; freezer cabinets are still likely to require trim and anti-sweat heaters.

### Outgoing technology

The most primitive anti-sweat heater controls for doors were constant power trace heating elements. Constant temperature heaters offered some temperature regulation of the glass panel but were typically over-specified in order to meet the more stringent test room (or worst-case) ambient conditions.

### New technology

It is far better to control the trim heater such that it is only used when it is required; where the store ambient dew point is lower than the inner surface temperature of the glass, there is no requirement for trim heating and the element can be turned off. Various systems are on the market to regulate the trim heater, either with a proportional or on/off control based on glass panel temperature, store RH and temperature.

Aloha Energy Group (2013) claims to achieve savings of 70-80% on anti-sweat heaters by automatically turning off the heaters on cabinets when they are not required. Supermarket Energy Technologies (Phoenix, Arizona USA) (2013) offers a product called Door Miser which claims energy savings of 80% on freezers and 90% on chilled cabinets. Emerson (2010) has developed an ACC (Anti-Condensate Controller) system that controls the heaters based on door frame temperature and the dew point temperature of the surrounding air. Dixell (2005) claims savings of 3.5% on the overall refrigeration energy consumption and 40% in savings on anti-sweat heater energy consumption by using their controls. Similar to the other controllers, the system seeks to regulate trim temperature but the information focuses on freezer cabinets (excluding chillers) which may explain the relatively modest savings claimed.

In many UK supermarkets it should be possible to remove trim heaters altogether from chilled cabinets. Heater controls will still be beneficial in stores where the HVAC system maintains a warmer dew point (to enhance shopper comfort) and on freezer cabinets.

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## Boreholes and ground sink condensers

Exchanging heat from the refrigeration condensers to the ground during the summer months can save considerable energy. Such systems can be open- or closed-loop. In open-loop systems, water within the ground (from a borehole) is used directly to provide cooling. In closed-loop systems, heat exchangers are constructed within the ground to exchange the heat.

Soil needs to be of good humidity for effective heat transfer (Kauffeld, 2016). This can be especially relevant when using CO2 as a refrigerant. Rejecting heat to ambient air from a CO2 pack will cause the pack to operate transcritically for part of the year, significantly reducing its efficiency whereas the use of a borehole or ground sink condenser enables a constant, and relatively cool, temperature sink for the “waste” heat from the condenser.

Leiper et al. (2014) further advocate the use of the ground loop as a heat source for space heating using heat pumps to extract heat from the ground loop; the ground becomes a large thermal storage medium for the system.

The use of ground source heat pump systems in mixed climate applications, where cooling requirements are dominant, is reported to save 37±18% compared to a conventional air to water heat pump system (Urchueguía et al., 2008).

Leiper et al. (2014) reported energy savings of 24.6% in a study comparing two similar stores; one running CO2 refrigeration with gas coolers in outside air and the other with ground coupling sinking the heat to boreholes. The gross internal area of the control store (rejecting heat to air) was 9,900m2, whereas that of the ground coupled store was 11,800m2. Installed refrigeration duty at the control store was 486 kW (13.7% LT) whereas 512 kW (15.5% LT) was installed at the ground coupled store. The manufacturer of the refrigeration packs was the same at both stores. The control store operated in transcritical mode for 29.5 hours during the four months monitored (September to December incl.).

According to Loose (2006) the energy saving achieved by waste heat discharged as geothermal energy in summer is dependent on the possible reduction of the condensing temperature and it is approximately 3% per Kelvin reduced condensing temperature. Rhiemeier (2009) stated that the area of soil required is not sufficient in order to discharge the entire condensing heat as geothermal energy. Instead, it is possible to subcool the refrigerant after the condenser by means of the dischargeable heat conducted into the soil. Experience suggests a 20% reduction in compressor power (Kauffeld, 2016).

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## Cabinet air flow

The air curtain is a vital feature of an open refrigerated cabinet. Apart from providing a thermal barrier and containment of the cold air within the cabinet, the air curtain also provides forced convective cooling of the front products to counter radiant heat gains. However, laboratory tests have shown that air curtain entrainment forms the greatest part of the cooling load on a vertical display cabinet (Faramarzi, 1999). Entrainment (also frequently referred to or included with infiltration in published research) of ambient air into the air curtain is estimated to account for 78 – 81 % of the total refrigeration load on open vertical display cabinets (Figure 1) (ASHRAE, 2002). In order to make any significant savings in energy consumption, the performance of the air curtain must be improved.

Entrainment is a direct result of mixing of the cold, downward flowing air curtain with the relatively still ambient air (due to mixing induced by viscous forces) at the interface between the two. In order to minimise entrainment, the air curtain velocity must be reduced to be roughly equivalent to that in the ambient region, reducing the viscous disturbances that cause the mixing. However, reducing the velocity does also reduce the integrity of the curtain and its ability to counter disturbances in the ambient air region. The consequent inability of the curtain to prohibit the exchange (due to the stack effect) of cold air between the cold volume and the ambient region, results in detrimental infiltration of ambient air (and increased heat gain).

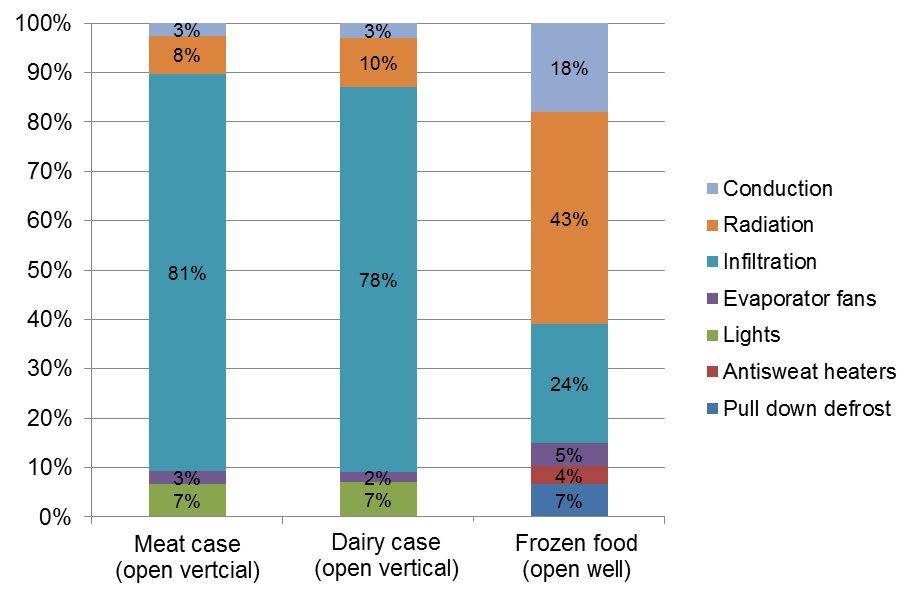


Figure . Components of refrigeration load for several display case designs at 24 °C dry bulb and 55 % RH (ASHRAE, 2002).

### Air curtain performance

The performance of a vertical air curtain depends on a large number of parameters. Table

1 compares the parameters considered to be most dominant by a number of authors.

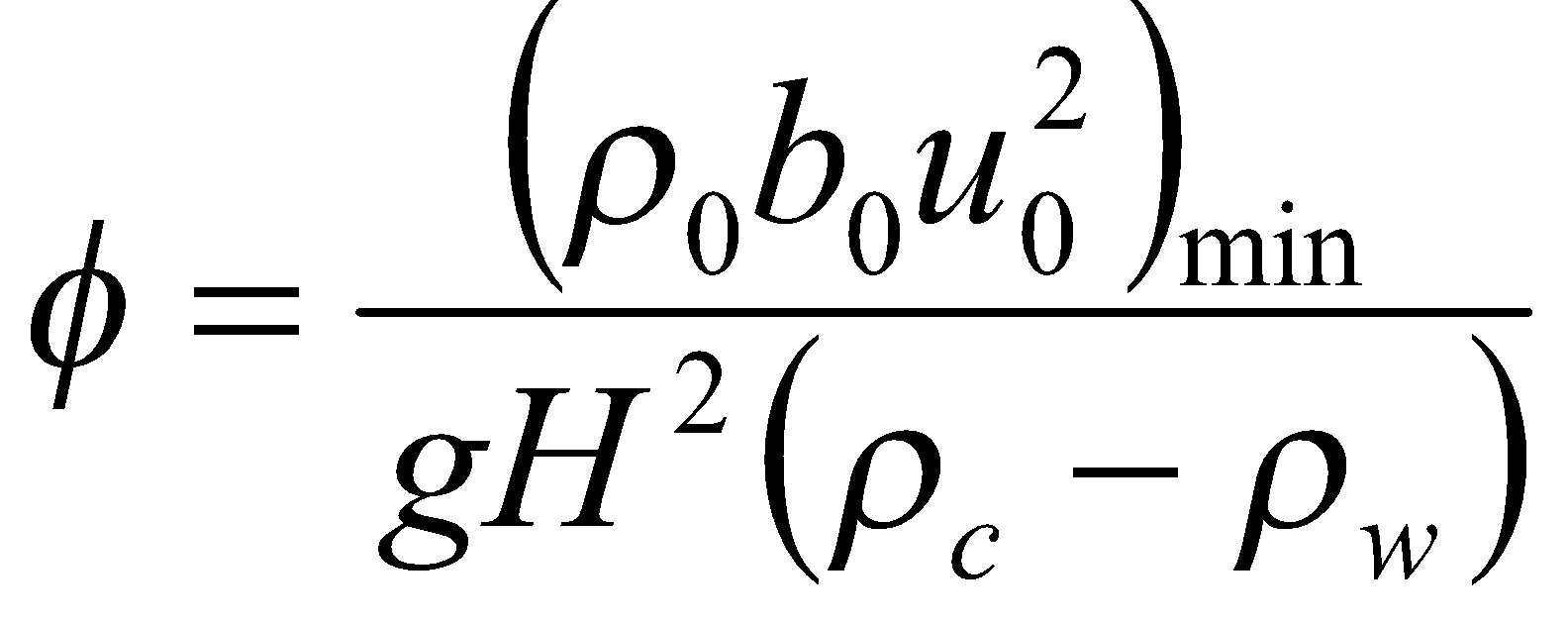
Table

. Parameters affecting air curtain performance, order as in original publications.

|  |  |
| --- | --- |
| Alamdari, 1997 | The width and length of the air jet  The initial jet velocity  The jet initial turbulence  The position and dimensions of the return air grille  Conditions of air on either side of the air curtain |
| Ge and Tassou (2001) | The width and length of the air jet  The initial jet velocity  The jet initial turbulence  The position and dimensions of the air return grille  Conditions of air on either side of the air curtain |
| ASHRAE, 2002 | Air curtain velocity and temperature  Number of curtains  Air jet width and thickness  Dimensional characteristics of the discharge air honeycomb  Store and display case humidity  Rate of air curtain agitation due to shoppers walking by  Boundary conditions in the initial region of the jet  Initial turbulence of jet |
| Navaz, et al. 2005c | Reynolds number based on the discharge air grille (DAG) width and with being the average velocity at the DAG  Normal vertical distance from the DAG to the return air grille (RAG),  DAG and RAG widths  DAG and RAG lengths  Absolute average temperature at the DAG  Absolute room temperature  Turbulence intensity at the DAG  Angle between the line that connects the centres of the DAG and RAG and the vertical direction, offset angle when the RAG is shifted laterally  Throw angle, the angle between the surface normal to the DAG and RAG. |

DAG is the discharge air grille and RAG is the return air grille.

Hayes and Stoecker (1969b) define the deflection modulus (Equation 1) as a dimensionless means of comparing air curtains and determining whether a curtain is likely to have adequate momentum to seal an opening; the numerator of the equation being the minimum outlet momentum and the denominator being a function of the density change and height.

**** Equation 1

*Φ = deflection modulus*

*ρ0 = density of air at nozzle outlet (kg.m-3)*

*b0 = slot width or nozzle diameter (m)*

*u0 = mean velocity at nozzle outlet (m.s-1)*

*g = acceleration due to gravity (m.s-2)*

*H = height of curtain (m)*

*ρc = density of air on cold side of the curtain (kg.m-3)*

*ρw = density of air on warm side of the curtain (kg.m-3)*

Hayes and Stoecker (1969b) produced charts to determine (non-recirculated) air curtain performance using such a deflection modulus. This enabled the prediction of the lowest velocity that ensured a continuous, unbroken curtain and also the heat transfer across the curtain. By assuming the minimum recommended velocity for a range of curtain heights, the model revealed a clear correlation between the curtain length and the heat transfer per unit area across the curtain; shorter curtains with lower velocities being most efficient. Furthermore, the data showed that the thickness of the curtain was less critical than curtain velocity in minimising heat transfer across the curtain; thick ones result in slightly lower heat transfer coefficients than thin ones. Foster et al. (2006) compared the model by Hayes and Stoecker to both experimental data and 2-dimensional CFD. Their investigation concluded that although the model was unable to give any guide on air curtain effectiveness in practice, it did provide a guide to selecting an optimal jet velocity.

### Turbulence

Turbulence is an important process in most fluid flow processes and contributes significantly to the transport of momentum, heat and mass (Jorgensen, 2002). Initial turbulence intensity is the turbulence intensity as measured immediately after the air grille discharge, the starting point of the air curtain. With proper conditioning (adequate duct dimensions, flow straightening and honeycomb) turbulence intensity can be reduced to less than 1% of mean velocity, but space and cost restrictions in industrial applications often make this impossible. Typical values for initial turbulence intensity in air curtains are in the range 2 – 20 % (Van, 1975).

Van and Howell (1976) noted that the initial turbulence intensity was significant in short curtains (length / thickness < 10) but had little effect on long curtains (length / thickness > 20). Van (1975) found that decreasing the initial turbulence intensity from 14 to 1 % could reduce energy consumption for operating the air curtain by approximately 40 %, mainly through the extension in length down the curtain of the potential core of the air jet.

### Discharge air grille design

In 2002, Navaz determined through CFD simulations that the infiltration (plus entrainment) rate increased approximately linearly with discharge air grille velocity (2002). Navaz (2005a) later expanded on this, adding that creating a laminar flow at the discharge and also having a velocity profile that resembled a laminar flow situation would be helpful in reducing the infiltration rate.

Since entrainment is a result of viscous mixing between the air curtain and the ambient air, it is logical that minimal entrainment will occur when the velocity and temperature difference between the (relatively still, warm) ambient air and the (relatively fast moving, cold) air curtain are minimal. This could be achieved by reducing the velocity of the air curtain and using a warmer discharge temperature. Conversely, as the cold side of the curtain performs useful cooling of the stored product, a faster velocity and cooler discharge temperature would be advantageous there.

Similarly, Stribling et al. (1995) claimed that the ideal air curtain for a vertical display cabinet should have a low velocity on the outside and a high velocity on the inside (case side). This would minimise entrainment on the outside whilst still ensuring effective mixing on the case side. Navaz et al. (2005b) tested this using CFD, DPIV (Digital Particle Image Velocimetry) and LDV (Laser Doppler Velocimetry) to study optimal velocity profiles at the DAG, concluding that the optimal profile was with a higher velocity at the inner surface / side of the case. Navaz used simple changes to the geometry of the discharge to experimentally reproduce (as close as possible) the idealised velocity profile; significant reductions in entrainment rate were measured.

The angle of the discharge slot or grille will also influence the stability of the air curtain. Hayes and Stoecker (1969b) determined that an air curtain would be most stable when the discharge was directed towards the warm side. Chen also noted this and found the optimum to lie between 15 and 20° for a curtain sealing a refrigerated cavity (Chen, 2009).

### Back panel flow

A common way to distribute cool air in a multi deck display cabinet is through perforated plates in the rear of the cabinet. Air is blown out of the back panel between the shelves to cool the load and stabilise the air curtain, also allowing a lower discharge velocity to be used for the air curtain. Typically the air velocity between the shelves will be 0.05 to 0.4 m.s-1, in the air curtain it is 3 to 5 times higher (Axell, Fahlen, and Tuovinen, 1999). However, a well-stacked shelf may block the rear panel holes and hinder the stabilisation of the air curtain.

Back panel flow effectively creates a low velocity curtain over the stored product on each shelf, and the low velocity enables minimal entrainment and minimal cooling duty. However, a weak spot will always exist at the underside of each shelf, affecting the shelf below; this is a particular problem for the top shelf. Negatively buoyant spilled air from the tiers above each shelf (or for the top shelf, the air curtain) prevents the air from infiltrating the cabinet to enable the cabinet to maintain a constant cold storage temperature. Where a cabinet has only back panel flow, the top shelf can never maintain the desired product storage temperature. Significant research has been carried out to determine the ratio of back panel flow to air curtain flow. Gray et al. (2008) claimed that 70 % of the air circulated should be used to supply the curtain and 30 % of the flow should be discharged through the back panel for the best results.

Gray (2008) also noted that a large amount of air supplied directly onto the shelves increased air spillage, and that the resulting "weak curtain" could not prevent infiltration at the top tiers. Work by Stribling et al. (1995 and 1995b) also investigated the effect on cabinet performance of reducing the air curtain flow rate by 50 % and 100 %, forcing the remaining cooling air flow through the back panel. It was concluded that the low velocity air curtain provided inadequate resistance to infiltration of room air due to a lack of momentum. Utilising high volume flows through the back panel of the cabinet yielded a 3.5% decrease in heat transfer across the curtain compared with the original design, suggesting that there would be potential to develop a vertical cabinet without a single continuous vertical air curtain over its full height. Assuming infiltration was 80% of total power, this would equate to a 2.8% reduction in total power.

Whilst an air curtain has been found to be essential, back panel flow is only one means to enable slower and therefore more efficient air curtains to be used. Back panel flow is strongly affected by shelf loading and cannot maintain product temperature within a cabinet on its own. Furthermore, it supplies the coldest air directly onto the stored product where the heat gain is least, which is not conducive to providing tight storage temperature control.

### Multi-layer air curtain

In an attempt to reduce heat and moisture exchange between the cabinet and the ambient air, and better stabilise the product temperatures, multiple layer air curtains have been used, particularly where frozen product is being stored. With a multiple air curtain setup, the innermost (cabinet side) curtain will be the coldest, being supplied with air from the evaporator coil. The second curtain will be slightly warmer, taking its supply from the return air grille and recirculating without any further cooling. Where a third curtain is employed, it is normally at ambient temperature, taking its supply from ambient air above the cabinet, and is used to reinforce the jet inertia and reduce the cold feet effect (Ge and Tassou, 2001; D'Agaro et al., 2006).

### Overall energy savings

There is limited information on the overall savings that can be achieved by optimising the air flow in conventional cabinets. In work by Foster, Madge and Evans (2005), the energy consumed by a multideck cabinet was reduced by 5.8% and at the same time the average test pack temperature in the cabinet was reduced by 1.7°C. Based on a simplistic heat transfer calculation (with 25°C external ambient temperature, as the test was at EN23953 climate class 3), an increase in set-points temperature of 1.7°C (possibly due to the reduction in test pack temperature) would reduce energy consumption by a further 7.5% (i.e. total energy savings of 13.3%).

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## Cabinet lighting controls – dimming/switching using occupancy sensors

Cabinet lighting can be a considerable part of the energy consumption of a refrigerated display cabinet. Fricke and Becker (2010) showed 10% of the energy consumption of an open display case was for the lights. This increased to 27% for a doored display case. This was for fluorescent lamps; these values would be lower for LED lights (approximately half). According to 4E (2012), in the US in 2011, well over half of vertical glass door chilled cabinets used fluorescent lighting.

Lighting is only needed when a customer is looking at the product, which is likely to be a small proportion of the time. When nobody is looking in the cabinet, lighting energy is wasted and it is also a heat load that the refrigeration system needs to remove.

Turning the lights off or dimming the lights during non-busy periods is simple, as most cabinet lighting systems in stores can be switched on and off at different times of the day, with obvious savings on energy, refrigeration load and potentially the life of the lamps. This is becoming more common practice as lighting is often centrally controlled in large stores.

For closed cabinets, door openings can be detected. For open cabinets, movement detection (occupancy) sensors/detectors can be used. Intelligent controllers can learn the busy periods and predict when to switch/dim lights.

There is considerable potential in stores with a variable trading profile to implement this technology. It is readily available and could be transferred from other areas of the cold chain (e.g. cold storage) where it is currently used. Lighting occupancy sensors are particularly compatible with LED lighting as this can be rapidly switched on and off, unlike more conventional fluorescent lighting. Fluorescent lamp life can be seriously reduced by constant turning on and off and therefore may not be compatible with this technology. Fluorescent lighting in glass-door freezers will take a long time to “strike” when down to temperature.

Fluorescent lamps can be dimmed once the discharge arc within the light has struck. Modern electronic ballasts can allow the light to run down to 1% of full power. Whenever new lamps are installed in dimming luminaires, they must be conditioned at full output for 100 hours before dimming (Apollo Lighting Ltd). However, if saving lighting energy is the primary concern, it would be more prudent to change to LED lights rather than dim fluorescent lights.

LEDs can be dimmed in one of two ways. Analogue dimming means that the drive current to the LED is reduced, reducing the LED power in proportion. This is a simple solution, but the colour temperature may vary as the drive current varies, reducing the quality of the light. The quality of the light is an important factor for the retailer and therefore this is not acceptable. A better solution, which is currently available at low cost, is pulse-width modulation (PWM). This drives the light with full current pulses. The width of the pulses varies the brightness of the lamp. Provided the pulse rate is high enough (approximately 200 Hz), the eye does not perceive the pulsing at all but only the overall average. PWM dimming may be implemented in the power supply or in the driver. 25% brightness is achieved by having the pulses on for 25% of the total time.

### Case studies

A demonstration (Richman and Tuenge, 2009) conducted at an Albertson’s grocery store in Eugene, Oregon, involved the retrofit of freezer case fluorescent lighting with LED strip lighting. This lighting was combined with a WattStopper FS-705 occupancy sensor. Prior to the demonstration, the case lighting remained on continuously with no provision for shutting off after business hours. The project revealed substantial energy savings as a result of the LED lighting retrofit, as well as additional savings attributable to the use of occupancy sensors. The occupancy sensors installed as part of the system were designed to drop the power draw of the system down to 20% of full power after 30 seconds of no activity. The system operated such that the presence of occupants would only increase the illuminance of nearby cases to full output and not the entire aisle. When customers passed near an end of the aisle, only the lighting in the case closest to the end of the aisle increased. Customers had to walk down the aisle to illuminate adjacent cases in that aisle. Adding occupancy-based control to turn the LED lights off in individual cases generated a 30.7% reduction in the relative full power hours compared to a constantly-on LED system. Further savings were realised from associated compressor operation reduction.

Diebel et al. (2013) replaced the fluorescent lighting (T8 with electronic ballasts) for a five-door case in a middle aisle of a store with LED lighting. The connected lighting load for each five-door case was 352 W. Operating hours for the case lighting were from 5 a.m. to 11 p.m. on weekdays and from 6 a.m. to 11 p.m. on weekends (for a total of 6,205 hours of operation per year). The case lights were turned off between 11 p.m. and 5 a.m. through the use of an energy management system. The connected lighting load for the case with the LED lighting was reduced to 189 W. The LED system also included dimming power supplies that were controlled by motion sensors. When no customers were near the case, the LEDs were dimmed down from maximum output. When a shopper approached the case, the LEDs were smoothly ramped up to full output. The motion sensors reduced the lighting power of the LED lamps by 43%. This was on top of the considerable saving by changing from fluorescent to LED lights. 80% of surveyed customers did not notice the dimming system. Of the 20% who did notice, most said it would not affect their shopping experience. This finding suggests that ramping is superior to low-high switching, in that switching between high and low output may be distracting to shoppers.

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## Cabinet selection

Replacement of refrigerated cabinets with the most energy efficient cabinets on the market is an effective way to reduce energy consumption. This could involve replacing cabinets with cabinets which appear identical but have more efficient fan motors, evaporators etc., or changing cabinets to ones which will operate differently, for example changing from open to closed and vertical to horizontal cabinets.

Evans, Scarcelli, and Swain (2007) tested a number of refrigerated cabinets under the EN441 test standard (this has been superseded by EN ISO 23953 that has broadly similar tests but slightly more stringent test room conditions) and found that energy savings of 10–20% could easily be achieved by selecting the best model within each cabinet type. They found that the most efficient cabinets (i.e. those using the least energy per unit of total display area) were those with the least exposure to the ambient conditions, i.e. those with doors or lids.

In the UK, a list of the most energy efficient refrigerated cabinets is available. This is called the Energy Technology List (ETL) and is operated by the Carbon Trust. By selecting cabinets from this list the buyer is able to claim Enhanced Capital Allowances (ECAs). According to data from Tait (2014) which includes data from Eurovent, ETL UK, CEC and Australia, there is a wide difference (more than 50%) in energy consumption for vertical open cabinets with the same display area.

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## Cabinet set-points

Cabinet temperatures are either controlled by individual controllers on each cabinet, or for larger retailers via a central controller. The primary control parameter is the temperature set-point, and this is set with the aim of keeping all the food within a safe temperature range.

As product within the cabinet will be at a range of temperatures, the set-point temperature will be only an indication of the product temperature. This indication is often based on a weighted average of temperatures measured in air at the exit and entry to the evaporator.

For frozen cabinets the set-point is low enough to maintain quality of the frozen product but not so low as to use excessive energy. For chilled meat and dairy cabinets, the set-point is low enough to maintain safety and quality for the shelf life of the product, but high enough to avoid freezing. For produce cabinets the set-point is somewhat of a compromise as different fruits and vegetables require different temperatures.

A tool is available (Geeraerd, 2014) which allows the user to look at the impact of temperature on the shelf life of a product. According to the tool, increasing the display temperature of cooked ham from 4 to 5ºC would reduce the shelf life by 20 hours, based on predicted growth of *Listeria* monocytogenes. The consequence on energy consumption is to reduce the heat load on the cabinet by approximately 5% (based on a store temperature of 25ºC).

According to Lindberg and Jensen (2014), green salad (ready to eat in a bag) has a shelf life of 12 to 13 days at 4ºC; this reduces to 8 to 9 days at a storage temperature of 6ºC. The subsequent reduction in heat load would be 10% (based on a store temperature of 25ºC).

Other food quality tools and experimental data are available to allow a more considered assessment of ideal display temperatures of each of the products displayed in supermarkets. The benefits of increased set-point should be weighed against the effect of reduced shelf life.

A survey carried out by Cemagref and ANIA (2004) which measured temperatures of 4 chilled products showed that 30% of products were above the 2°C recommended storage temperature. Willocx et al. (1994) measured the temperature of processed vegetables in Belgian retail display cabinets. This showed that differences of more than 5°C were measured on the shelves. Temperature also increased towards the end of the day in certain locations by 4°C and towards the end of the week by almost 7°C.

In Ireland, the Food Safety Authority found that the temperatures of pre-packed sandwiches were above 5°C in 57% of cases (FSAI, 2002). In a similar study in 2003, the FSAI found that 12% of pre-cooked sliced ham was stored above 5°C.

Studies in the USA have shown the temperature of foods in chilled food distribution channels are frequently in the range of 45-55°F (7.2°-12.8°C) (Food Spectrum, 2002). Jol et al. (2005) claimed that 20% of domestic and commercial refrigerators operated at a temperature of >10°C. Audits International found that 48% of product temperatures in retail refrigerators were >5.0°C and 17% were >8.3°C.

Derens et al. (2004) and Derens-Bertheau et al. (2015) conducted surveys in 2002 and 2012 which measured temperatures in retail display cabinets in supermarkets in France. These showed mean temperatures (standard deviation) of 4.2ºC (2.4ºC), 3.1ºC (2.6 ºC), 3.4ºC (1.8ºC) and 2.8 (1.2ºC) for yoghurt, prepared meals, meat and sliced cooked ham respectively.

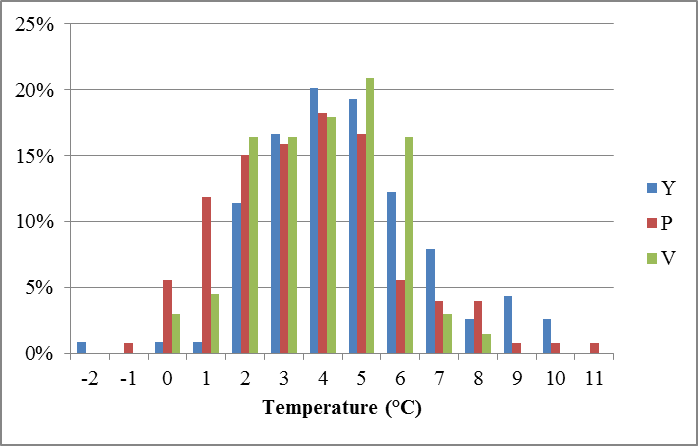


Figure . Temperatures in 307 supermarket retail display cabinets measured in France in 2002. In the legend Y = yoghurt, P = prepared meal and V = meat.

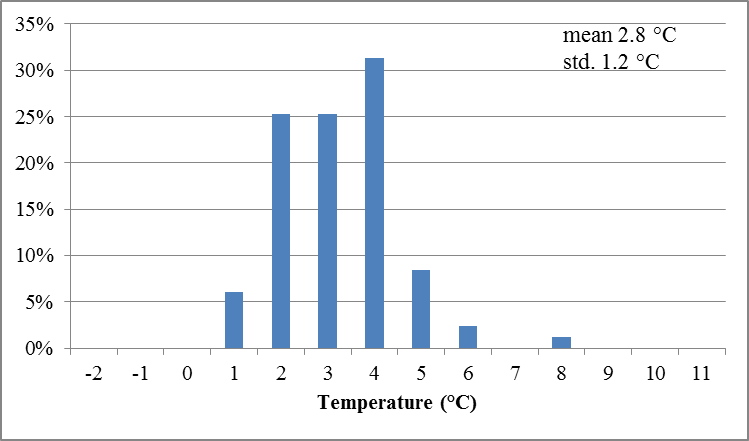


Figure . Temperatures in 83 supermarket retail display cabinets measured in France in 2012 which displayed sliced cooked ham.

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## Centralised air distribution

Conventional refrigeration systems provide cooling to refrigerated cabinets by direct expansion of a refrigerant in an evaporator inside the refrigerated appliances.

Yu et al. (2009) tested a cabinet with a centralised air supply. In this system, the cabinet was supplied with air from an air handling unit via a duct. In a full scale system which has been installed in a number of UK supermarkets in 2017, the air handling system supplies an aisle of cabinets.

The removal of the evaporator from the cabinet offers potential advantages. Fitting the evaporator inside the cabinet can cause constraints on its size. Therefore, the evaporator is often smaller than ideal. This can lead to low evaporating temperatures and difficulties with defrosting. It is not possible to allow all of the useful part of the evaporator to extend to the edges of the cabinet due to space requirements for end turns, expansion valves and space requirements for servicing and fitting. This can lead to air flow and poor temperature at the edges of the cabinet. The air ducted system cabinets also do not have fans as these are replaced by one large centralised fan.

Yu et al. (2009) found that the centralised air distribution system could easily control the air curtain velocity, decrease the frost magnitude, increase the defrost cycle from 6 h to 9 h, and reduce the maximum product temperature rise in the defrost cycle from 3.0ºC to 2.0ºC.

Another advantage of this system is that the evaporator condensate drain is external from the cabinet. Removal of the drains from the shop floor has considerable benefits in regard to water leakage and maintenance.

Often HVAC systems utilise free cooling by taking air from outside and using it to cool buildings. A similar system can be used with centralised air distribution systems if the outside ambient air temperature is below the evaporator return temperature for the refrigerated cabinets. Centralised systems also contain the refrigerant in one area and so it is possible that direct emissions can be reduced as it is simpler to prevent leaks and to identify leaks if they should occur. In addition, a low GWP refrigerant such as a hydrocarbon can be used in the centralised plant.

The practicality of this arrangement needs to be considered. The ducting will take up more space than the refrigerant piping. Noise and condensation from the ducting will need to be considered. Yu et al. (2009) found that balancing the cabinets so that all cabinets, whether at the beginning or end of the duct run, get the same air flow is critical.

### Case studies

A major retailer (Asda) has introduced a centralised air distribution system (Mistral Air System) into their stores. They claim a reduction in both running and maintenance costs. Asda claim a reduction in energy by up to 15% and a reduction in charge of 40% (Gaved, 2016).

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## Defrost drain traps

Defrost drains should always have a trap (Stoecker, 1998). This is common practice and traps should be fitted to all cabinets. Problems do occur when the traps or drains become blocked and water backs up and spills from the cabinet. This may have some impact on energy use if water in the cabinet base affects air flow in the cabinet. However, there is no information available on the impact of blocked traps/drains. As it is common practice to fit traps, this technology has not been assessed for carbon saving.

The failure of drains to cope with the resulting defrost is a major cause of aggravation to store management – wet, slippery floors, etc. The absence of frequent access to the major underfloor clay drainage pipes along a run (ideally one access per cabinet) is a major cause of this. Because of this lack of frequent access to “trunk” drainage, long runs in plastic pipe occur under cabinets. Frequently these fail to run continuously downhill. They are liable to become iced up or otherwise blocked.

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## Defrosts

Most chilled cabinets operate on passive or ‘off-cycle’ defrosts, and therefore almost all defrosting technologies are only appropriate for frozen food cabinets.

### Electric defrost

Electric defrost heaters use a significant amount of energy. Due to the inefficiency of getting heat from the defrost rods to all of the iced fins, much of this energy goes into the cabinet rather than into melting the ice. The ice build-up on the evaporator is rarely evenly distributed and so a great deal of energy is used to heat the evaporator block. Lawrence and Evans (2008) found electric defrosts to be only 15% efficient in a 2.5 m frozen food well display cabinet at climate class 3 (temperature of 25 °C and relative humidity of 60%).

### Hot/cool gas defrost

Hot gas defrosts use the compressor discharge to defrost the evaporators. Refrigerant gas is introduced downstream of the TEV (thermostatic expansion valve). This is more efficient thermally than electric heaters as heat is added inside the evaporator tubes rather than conducted from outside. Hot gas defrosts are also more thermodynamically efficient than electric heaters as the COP of the refrigeration system during defrost will be more than 1.

Hot gas defrost results in lower overall surface temperatures within the evaporator. Temperatures will increase less, meaning that less refrigeration energy is used to re-cool the cabinet. Potentially, this can mean that the set-point temperature could be increased as there would be less temperature increase during defrost. Due to the lower surface temperatures there is less likelihood of steam being created which then condenses elsewhere.

Cole (1989) showed that typical hot gas defrosts are only about 20% efficient (this is the proportion of energy that actually goes into melting the ice) which is slightly more efficient than the data for electric defrosts (see above). 60% of the energy escapes into the ambient air / environment and 20% to heat the metal. It was stated that the maximum theoretical efficiency is 60 to 70%.

The cool-gas defrost is essentially the same, except that the gas used for defrosting is obtained from the receiver, rather than from the compressor discharge. Cool-gas reduces thermal expansion that occurs when subjecting a cold suction line to elevated defrost temperatures. In addition, using saturated vapour will provide a quicker defrost cycle than when using superheated discharge vapour.

Although gas defrost uses less energy than electric defrost, gas defrost requires valving that increases head pressure and consequently requires a higher refrigerant charge. California Utilities Statewide Codes and Standards Team (2013) assumed an increased charge size up to 10% for a system utilising hot gas defrosts. Gas defrost was also expected to increase the potential for leaks due to the need for additional piping and valves and the thermal shock caused by the rapid change in temperature. California Utilities Statewide Codes and Standards Team (2013) assumed that prohibiting hot gas defrost would reduce refrigerant leak rates by 5%. In the climate zone of Sacramento, they estimated an annual refrigerant saving of 30 to 46 kg by removing hot gas defrosts, equating to a greenhouse gas saving of 119 to 182 MTCO2eq per year. The greenhouse gas associated with the extra energy used by the electric defrost was much lower, equating to 11 MTCO2eq per year, providing a net benefit in greenhouse emissions by removing hot gas defrosts.

Compressor discharge pressure has to be artificially raised during defrost to ensure enough hot/cool gas is available; this increases plant power consumption and reduces efficiency. This technology is therefore not compatible with floating head pressure technology.

### Reverse cycle defrost

For a hot gas defrost to work, the compressor needs to run whilst the cabinet is defrosting. For pack systems this is not an issue, as only one or perhaps a few of the many cabinets will be defrosted at a time. For integral cabinets this requires a reverse cycle defrost.

The refrigeration system is reversed by the use of valves. The evaporator now becomes a condenser and the heat of condensation melts the ice. This method results in a faster defrost than electric. On systems with a distributor, the distributor creates too high a pressure drop, therefore an auxiliary side connector must be used to allow the defrost gas to bypass the distributor. The advantages and disadvantages of this system are similar to those of a hot gas defrost system.

### Warm liquid defrost

Warm liquid defrosting (WLD) uses warm liquid from the system condenser to defrost the evaporator coils. Among the advantages is the potential for a simpler overall system design eliminating hot gas or saturated vapour lines and some of the associated valves.

There is a concern that not all the liquid used for coil defrosting and subsequently expanded into the suction line will be evaporated by the superheated vapour from the operating display cases (not under defrost) under all conditions (Baxter and Mei, 2002). This could cause liquid to return to the compressor. Another concern is that at very low liquid temperatures (below about 10-15°C) there is not enough heat available in the condenser liquid to effectively defrost the evaporators. This could happen at low ambient temperatures, though it can be mitigated by increased condensing pressure. However, in this case there is an energy penalty.

### Heat bank defrost

Waste heat is always created during refrigeration. The problem is that this waste heat is created when the evaporator is cooling and is not available when the evaporator is defrosting. A heat bank can be used to capture the waste heat whilst the evaporator is cooling and then discharge it into the evaporator during defrost.

The Kramer-Trenton Company patented a heat bank defrost (Thermobank method), where the discharge of the compressor heats a water store (Dossat, 1991). During a defrost the heat from the water bank is used to re-evaporate the refrigerant condensed in the defrosting evaporator.

A thermal energy storage based novel reverse-cycle defrosting (NRCD) method was developed by Dong et al. (2011) for air source heat pumps. The results indicated that during defrosting the NRCD method improved discharge and suction pressures and shortened the defrost duration by 60%. In addition, the defrosting energy consumption was reduced by 48.1% compared to the use of a standard reverse cycle defrosting method.

### Thermosiphon defrost

A way to use a thermal heat store to defrost an evaporator has been developed by FrigescoTM (Exeter, UK). This system uses a phase change material (PCM) store that is heated by the cabinet liquid line (an added advantage is that it also subcools the liquid). When the PCM is melted a defrost can be initiated. During a defrost, valves are actuated such that the cold evaporator and the PCM store form a closed loop. The temperature gradient formed between the PCM heat store and the evaporator allows a thermosiphon to exchange heat between the two. Refrigerant boils in the heat exchanger, melts the ice on the evaporator and condenses before returning to the heat store to repeat the process until the heat store is exhausted.

The technology has been used on both frozen cold stores (Campbell, Davies and Thangamani, 2014) and frozen retail cabinets (Foster et al., 2013). In the study on a small walk-in frozen food store room cooled by a conventional direct expansion refrigeration system, the power consumption of the two systems was compared. It was shown that the defrost system reduced the power consumption of the cold store by at least 20% in mid-winter conditions. Tests on a frozen half glass door/well freezer cabinet to the EN23953 test standard found that total (refrigeration and direct electrical) energy consumption was 40% lower with the thermosiphon defrost. This was partly due to elimination of the use of electrical defrost power (which reduced direct electrical power by 39%). Since the system leads to lower liquid refrigerant temperatures at the expansion valve, there is a saving in compressor power for the same refrigeration effect. As the defrost is quicker, the temperature increase inside the cabinet is smaller, reducing the amount of heat which has to be removed after a defrost. The cabinet can also be run at a higher evaporating temperature, further reducing energy consumption. There was a reduced refrigeration load (Фrun reduced by 28%) caused by the defrost heaters not adding heat into the cabinet. Due to the increased evaporating temperature allowed by the reduced temperature deviation during defrosts, the refrigerated energy consumption (REC) was reduced by 41%.

Results from Foster et al. (2015) showed that it was possible to eliminate the use of the electrical defrost heaters during defrost; however, a heater mat was required to allow the water to drain from the cabinet without re-freezing. The electrical defrost power was reduced by approximately 80%. Subcooling of the liquid refrigerant reduced the refrigeration duty by approximately 10%. The technology is still at a development stage and methods need to be employed to make sure all ice is removed from the cabinet between defrosts.

The technology has recently been applied to an air source heat pump. In controlled laboratory conditions, a GlenDimplex Class A 8 kW heat pump fitted with a Frigesco™ defrost system operating under simulated winter conditions showed a 17% improvement in COP and an overall reduction in energy consumption of 14.5% compared with the standard unit with reverse cycle defrost (Davies, 2016).

### Defrost controls (on demand)

Defrosting the evaporator only when necessary can save considerable amounts of energy. Most conventional defrosts are scheduled at pre-set times (every 6-12 hours is typical) and this can result in unnecessary defrosts, excess energy use and increase in product temperatures. Defrost controls minimise the number of defrosts needed by a cabinet. This has the effect of reducing direct energy consumption, and also leads to reduced heat gain in the cabinet.

Defrost on demand has been tested by a number of researchers. The Electric Power and Research Institute (California, USA), in partnership with Johnson Controls/Encore, has developed a new demand defrost controller for supermarket refrigerated display cases (Khattar, 1998). Field demonstrations were carried out in supermarkets in Minnesota, New Jersey and Florida. At the Minnesota store, the time between defrosts increased by a factor of four in the winter. The total time in defrost dropped by as much as 66% and on an annual basis by 34%. In New Jersey the time between defrosts increased from one day to the maximum limit of three days, which reduced defrost heater operation by 63% on an annual basis. Analysis showed energy savings of 25,000 kWh per year, increasing to 38,000 kWh per year if indirect savings in compressor use are included.

Tassou, Datta and Marriott (2001) showed that, using relative humidity as a control parameter, the defrost frequency can be reduced considerably without affecting cabinet performance and product integrity. The results from field measurements and laboratory tests showed that the defrost frequency of four cycles per day, normally employed for medium temperature multideck display cabinets in the UK, is only required in extreme conditions (ambient above 22ºC and 60% RH). As this occurs rarely, defrosts can be reduced based on environmental conditions.

Lawrence and Evans (2008) tested an algorithm that detects the need for a defrost from the pattern of refrigerant flow (or evaporator exit superheat). The results were compared with those from a conventional 3 defrosts per day cabinet setup. The tests were deliberately performed in higher temperatures and greater humidity than would normally prevail in UK supermarkets. Even so, it was found that the mean time for ‘defrost required signals’ being given was 38.8 h. The energy used per year for a single 2.5 m frozen well cabinet with a defrost every 38.8 h was 538 kWh whereas the energy used with a defrost every 8 h was 1960 kWh. This means that the 38.8 h defrost represents energy savings of 72.5%. In a supermarket with 40 cabinets, the annual savings would therefore be 56,880 kWh. With an electricity conversion factor of 0.41 (2016 UK Government figures), this equates to a saving of 23.4 tonnes of carbon dioxide emissions.

Practical application of defrost on demand systems has been varied with some systems showing no benefits and others savings of 40% of the defrost energy.

### Ultrasonic defrosting of evaporators

Ultrasonic vibrations can be used to release frost crystals from evaporators. The technology is not suitable for removing the basic ice layer but frost crystals and branches on the ice layer can be removed. The frost falls off the evaporator and collects beneath it. As the basic ice layer cannot be removed the technology needs to be applied throughout the refrigeration cycle to be effective. In trials a vibration frequency of about 28.64 kHz was used with an average power of approximately 30 W (Wang et al., 2012).

No practical information is available on the savings that can be achieved using the technology.

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## Diagonal compact fans

A problem with evaporator fans in retail display cabinets is the amount of space allowed for the fans. This means that they are often placed at a non-ideal orientation to allow them to fit into the tight space and this restricts the air flow. This restriction builds up pressure, which axial fans do not work well against.

According to ebm-papst (2015) the operating point of the diagonal compact fan, compared to the axial compact fan, lies in a higher pressure range. This can increase air pressure and increase air flow rate which has potential benefits for cooling in small spaces such as retail display cabinets. An efficiency of 85% is claimed.

**Reference:**

ebm-papst Inc. “The theory and practice of small fans - What is the correct type of fan: Axial, radial, diagonal?” http://www.ebmpapst.us/media/content/downloads\_1/support\_1/tech\_articles/Theory\_1.pdf (accessed 2015).

## Distributed refrigeration system

Most of the major retailers use a centralised refrigeration plant for the majority of their retail display. Distributed systems consist of several smaller parallel compressor racks distributed throughout the store. These packs are located close to the display cabinets or cold stores. A typical store might have 5-6 units.

A centralised system distributes refrigerant through the entire refrigerated area and is often sited a long distance from the cabinets on the roof or in a dedicated plant room. As each of the distributed systems is smaller and quieter, they can often be placed closer to the particular cabinets they serve. The reduction in pipe lengths reduces energy losses in the system.

Having shorter pipe runs, less refrigerant, being connected to fewer cabinets and using smaller and fewer compressors, can make maintenance simpler for each distributed system. A major advantage of a distributed system is that refrigerant leaks are confined to individual smaller systems rather than one centralised system. Therefore leaks can be determined sooner and have more localised effects. Any system failure will be confined to a smaller number of cabinets. Another important advantage is that a lower system refrigerant charge allows the possibility of using flammable gases which are lower GWP and more efficient.

According to Emerson Climate Technologies (2010), distributed systems have a one-third lower leakage rate than centralised systems because they use less pipe work and typically comprise a small factory-built system using hermetic scroll compressors. Overall the refrigerant charge is 75% of the equivalent centralised system. Therefore it would be expected that loss of refrigerant would be less than that of a centralised system.

In a centralised system the cabinets will generally run on only two evaporating temperatures, frozen and chilled. However, the chilled and frozen cabinets will not all operate at the same cabinet temperature. Chilled cabinets may be operated at 8ºC (produce), 3ºC (dairy), and 2ºC (dairy). Freezer cabinets may be operated at -21ºC (ice cream), and -18ºC (frozen but not ice cream). A distributed system allows these cabinets to be grouped together and run at different evaporating temperatures. In a centralised system, the suction pressure will be the lowest of all the cabinets, and a centralised system will allow the warmer cabinets to run at a suction pressure, potentially saving energy.

**Reference:**

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## Doors on cabinets

Doors can be fitted to open fronted chillers to improve product shelf-life or quality, through the stabilisation of temperature, or to reduce energy consumption. (Markusson and Rolfsman, 2013, Insight, 2011).

Whilst environmentally conscious shoppers have been advocates of doors on refrigerated cabinets in supermarkets for a considerable time, research into the use of doors has presented conflicting arguments regarding their benefits and challenges; as with many technologies, appropriate application is key to realising the benefits.

Initially the uptake of technologies with doors was slow due to a perception that it would reduce sales:

• Asda claimed that early trials had led to a dip in sales as a result of the added inconvenience to shoppers (Jamieson, 2008).

• Pearce (2009) stated ‘We are having to balance energy savings with customer concerns.’

• A Tesco store development manager was quoted as saying ‘there was a stigma on doors being a barrier to purchase and a fear that shoppers would migrate to the competition.’ (Insight, 2011).

However, the Insight article went on to say that although the doors slowed re-stocking during trials, the warmer aisles, better preservation of the food and energy savings meant that the door installations were considered to have been a success in the majority of cases. However, doors were later removed from snacking aisles of high footfall stores.

Fricke and Becker (Fricke and Becker, 2010a) reported that doors had no impact on sales following a study in the Midwestern United States. Two similarly situated supermarkets were chosen (one with and one without doors) to ensure that climate, weather, time-of-year and economic conditions of the shoppers were comparable. A ‘before and after’ comparison of product sales was performed in each of the two stores. An existing display case line-up was identified in each store, and the sales data of the products from that display case line-up were collected for a period of approximately two months. The existing display case line-up in each store was then replaced with a new display case line-up. Each new case line-up was then stocked with the same products, in the same location within the new case, as they appeared in the old case line-up. The sales data of these products from each new display case line-up were then collected for a period of approximately two months. The results showed a circa 2% drop in sales as a result of fitting glass doors compared to refurbishment of old open cabinets but, since the difference was not statistically significant, the conclusion drawn was that doors did not impact sales.

Customers were interviewed at the stores of three leading supermarkets in the Cheltenham and Gloucester area (South West England) over a one week period during February 2013 (Milnes, 2013; The Grocery Trader, 2013, ACR News, 2013) Half of those customers questioned stated that they would prefer to shop from cabinets without glass doors but it was the under 40s who most strongly believed it would be more difficult to shop from cabinets with glass doors; 42.5% of all surveyed thought it would be difficult or very difficult to shop from cabinets with glass doors. For the under 40s this figure rose to 56%. This could be an indication that speed and convenience is a strong driver for younger shoppers; of those surveyed 43% shopped from cabinets with glass doors once a month or less. 88% of those surveyed rated “clear display of merchandise” as important and unsolicited comments regarding condensation on glass doors obscuring visibility of merchandise were received.

### Technology variants

Chiller / freezer

Glass doors and lids are already routinely fitted to freezer cabinets but merchandisers have resisted moves to fit them to chilled cabinets.

Retrofit or new design

Cabinets designed to be glass door cabinets can have smaller evaporators and air circulation systems, reducing their overall capital cost. However, the lowest cost option for a supermarket which already has open cabinets is typically the retrofit option where glass doors are fitted to existing open cabinets. It is possible to significantly raise the evaporating temperature after the doors have been fitted.

Door + air curtain or door only

Air curtains behind the door improve efficiency when the door is opened. Where retrofitted to existing open cabinets there is already an air curtain in place behind the door; this improves performance of the cabinet whilst the door is open. Some cabinets with glass doors, especially where frequent door openings are expected, are also designed with an air curtain which may be activated or accelerated as a result of a door opening.

Doors with and without trim heaters

The requirement for a trim heater depends on the store’s ambient conditions and the temperature of the door glass. Trim heaters are common on frozen cabinets but are not always used on chilled cabinets. On demand trim heaters controlled by ambient humidity are preferable.

Sliding or hinged

Doors are an obstacle and ease of use is important. Shoppers need to be able to open the door and hold a basket at the same time, shop staff need to be able to restock the cabinets, doors need to self-close when shoppers forget and fire regulations must be adhered to so door opening into aisles must not cause a restriction – often requiring wider aisles - and the hinge or runner must be robust enough to withstand frequent use. Various hinge designs and sliding arrangements have been proposed by manufacturers. It may be possible to change the design of the cabinet to re-stock from behind.

Transparent material

Perspex doors offer a lower cost solution but are perceived to be a lower quality and less reliable than glass doors. Either single or double glazing can be used. Frequency of opening or capital cost payback tends to be the determining factor between glass or plastic doors.

Sealed to frame or loose fitting

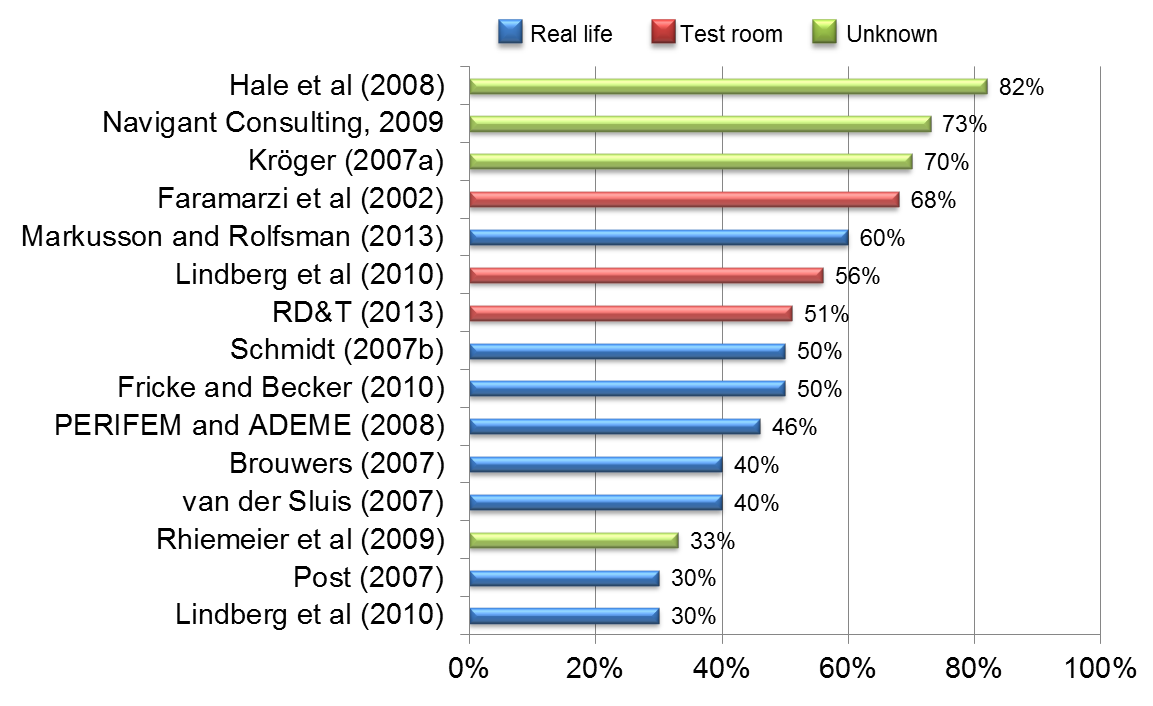
For best results, the door should seal the front of the cabinet but this can be difficult to achieve, particularly on a retrofit case. If the doors are sealed they can be too tight to open due to suction, as the cold air that enters during the door opening cools.

### Energy saving potential

Savings claimed as a result of fitting doors to display cabinets vary. This is often due to how the savings are measured, how the test was carried out (e.g. real life or to a test standard) and whether any increased direct demand (e.g. additional lighting, anti-sweat heaters) at the cabinet was factored in. Figure 4 presents a summary of savings reported. Faramarzi, Coburn and Sarhadian, (2002) found that fitting doors reduced the refrigeration load by 68%. Fricke and Becker (2010b) also reported energy savings of 23% during an in-store trial.

Manufacturers of doors claim to save 35 to 45% of energy (Delta Refrigeration Services Ltd.) and the Co-operative supermarket claims to be saving £50 million per year from doors fitted in just 100 stores (The Guardian, 2012) while Tesco claims savings of £4 million per year from 350 convenience stores with doors on its refrigerated cabinets.

Infiltration accounts for 73% of the refrigeration load in open refrigerated display cabinets according to Faramarzi (1999), whilst ASHRAE (2010) states that it is 81% and 78% for meat and dairy cases respectively. The infiltration load is dependent on the temperature and humidity difference between the cabinet and the ambient. It is these infiltration loads that the doors seek to reduce.

Citing a piece of work by van der Sluis, (2007), Tassou (2011) claimed that glass doors could be expected to save 50% of the energy consumed by chilled multideck cabinets. However, the original report from van der Sluis only claimed savings of 40% (or 56% at the refrigeration compressor) during a study which fitted glass doors to a cabinet in the Netherlands.

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A mathematical model produced by Markusson and Rolfsman (2013) predicted that if doors were fitted to all cabinets in a supermarkets, the refrigeration load would reduce by 30%. They found that they could increase evaporating temperature by 6 to 7°C (from -8 to -1ºC) when doors were fitted and that refrigeration system related energy savings of 60% were theoretically achievable if the refrigeration system was re-optimised. This may, however, not be practical if the uplift in temperature causes an adverse effect on other equipment attached to the same refrigeration system.

Figure . Savings reported

Rolfsman et al. (2014) confirmed the earlier theoretical study under real conditions in a supermarket. It was concluded that with doors and with an increased temperature, it was possible to run at a higher brine inlet temperature (from -8 °C to +2 °C). Frost-free operation also led to improved temperature quality of the provisions, which increased the COP of the installation if the system was optimised. They showed the doors decreased cooling demand by 50%. Adaption of the refrigeration to the lower cooling demand and part-load conditions allowed a reduction in electric energy demand of more than 75 %.

Given that the anti-sweat heaters and additional lighting will have contributed to increasing the refrigeration load, figures from Faramarzi, Coburn, and Sarhadian (2002) imply that infiltration was almost entirely prevented for the tests carried out (the door was not opened); the Markusson and Rolfsman (2013) figures appear more plausible for normal usage.

Since the open fronted refrigerated cabinets both cool and dehumidify the store, fitting doors can lead to increased shopper comfort. However, in summer it will also lead to a reduced cooling load which will need to be covered by the store air conditioning system which may need to be up-rated.

The additional power consumed by the HVAC will offset some of the savings of glass doors but HVAC systems are typically a more efficient means of removing heat from the store than the display cabinets due to their warmer evaporating temperature and resulting greater COP. However, if the reduction in cooling load at the cabinet is 68% and the energy savings are 23% (Faramarzi, Coburn, and Sarhadian, 2002), the COP of the HVAC system would need to be three times that of the chiller plant (which is unlikely).

Laboratory tests conducted to BS EN ISO 23953 at climate class 3 (25°C and 60% RH) showed a 44% reduction in TEC/TDA (total energy consumption / total display area) for M0 cabinets and 41% for M1 cabinets (Evans, 2013). The glass door cabinets in this study did not have any trim or anti-condensation heaters fitted. The evaporating temperature is not known for these tests; the TEC/TDA calculation will have favoured warmer evaporating temperatures so some of the savings may be a function of a warmer evaporating temperature.

Fricke and Becker (2010a) calculated mean electrical energy consumption of new doored and new open display case line-ups using ARI Standard 1200. They found that the doored cabinets consumed 72% less compressor and 20% less fan power. However lighting power increased by a factor of 2.3 and anti-sweat heaters were required which used the equivalent of 37% of the compressor power of the open cabinet. This led to a reduction of total energy for chilled doored cabinets of 18%.

### Costs

Costs vary considerably depending on the type of technology with retrofit plastic doors being cheapest and purpose built, double glazed slide and hinge doors being the most expensive. Quotes provided to the authors in 2014 suggest the following costs for fitting doors to typical 8 foot refrigerated cabinets (Table 2).

Table . Costs for installing doors on cabinets.

|  |  |  |
| --- | --- | --- |
|  | **Parts cost** | **Installed cost** |
| Low cost glass doors | £750 | £1012 |
| Quality or ‘top end’ doors | £1500 | £2025 |

### Paybacks

Many of the trials described above were carried out in a laboratory or on a limited number of cabinets. To extrapolate savings across a supermarket using these data has some limitations. A French study (PERIFEM and ADEME, 2008) extrapolated savings of between 38 and 50% across different sized supermarkets in France. The savings predicted are shown in Table 3.

Table . Financial savings predicted by fitting doors to cabinets.

|  |  |  |
| --- | --- | --- |
| **Size of supermarket (m2)** | **Annual savings by adding doors (excluding installation)** | |
|  | kWh | € |
| 18,000 | 4,500,000 | 300,000 |
| 5,500 | 800,000 | 60,000 |
| 2,500 | 700,000 | 50,000 |

A further study by the Swedish Energy Ministry indicates that the payback period for fitting glass doors to open fronted cabinets was about 16 months. It is not clear if they considered the optimisation of the refrigeration system in their calculation. Several factors are often not taken into account when calculating the costs of changing from open fronted to glass doored cabinets. With open fronted cabinets a proportion of the store cooling is carried out by cold air falling from the open fronted cabinets and usually heat is required to counteract the cold aisles. With closed door cabinets there is less cold air spilled into the aisles and therefore there will be an increased net cooling demand in summer and a reduced net heating demand in winter. In addition, changes to the refrigeration system may be required as the refrigeration duty is likely to be reduced significantly. PECI (2011) created a model of a supermarket to examine the full costs of retrofitting glass doors. The total cost of all labour and materials and for any changes to the refrigeration system were $566.17/ft of case (approximately £1,122/m). The savings varied according to the exact cabinet type but generally provided a payback of approximately 1.4-1.6 years when including the savings from reduced refrigeration energy and reduced gas usage from less store heating.

The PECI study did not take into account any ongoing maintenance for the doors. Inevitably doors or lids on cabinets will require some maintenance. It would seem unlikely that the annual maintenance costs would exceed the installation cost (stated by PECI to be approximately £196/m). Even if the doors were reinstalled each year this would equate to only about 25% of the savings achieved from the cabinets themselves and the reduced energy for air conditioning.

During retrofit operations there is also a possibility to make further energy saving adjustments to cabinets (Navigant Consulting, 2013). For example LED lighting could be fitted (potentially with occupancy sensors). In addition, once the cabinet is enclosed it may be possible to reduce the fan power and move to EC (electronically commutated) motors. However, care should be taken to ensure that any adjustments do not have a negative impact on food temperatures. Other savings from optimising controls, repairing refrigerant leaks and optimising pipe work and expansion valves can also be implemented. To save electrical energy (costs) for retrofitted doors, it is necessary to optimise the refrigeration system for new conditions.

### Infrastructure alterations

If the potential to lower the energy is to be fully utilised, the existing system must be customised for the lower cooling demand (Markusson and Rolfsman, 2013). With doors, the demand on the compressor(s) will reduce by 30-50% and the evaporating temperature can increase by 6 to 7°C. This makes it imperative to introduce good part load efficiency.

Glass door cabinets have an overall reduction in duty compared to open cabinets and refrigeration packs can therefore be installed for a reduced capacity. Fricke and Becker (2010b) claim that it is possible for them to be 15% smaller. However, as the cabinet load increases so greatly (above that of its closed state) during busy periods in the store, a greater turn down ratio is required and this can increase pack complexity.

Cold supermarket aisles are a result (mainly) of spilled cold air and, in part, radiative cooling from the cold face of products facing the customer. Several studies have been carried out and published (Perez, 1985) (Foster and Quarini, 2001) which present a range of methods to reduce the cooling of the aisles, which are often implemented in stores. One recent investigation (Pursglove, 2013) measured temperatures in the chilled aisles of eight supermarkets (one from each of the major chains) between 6.9 and 20.2°C, the average among the stores being 14.6°C, indicating that no significant improvements had been made in store since the work by Foster and Quarini (2001). The article claims that UK guidelines recommend that supermarkets maintain a temperature of 19 -21°C in the winter and 21-23°C in the summer but most supermarkets set themselves cooler targets. ASHRAE HVAC Applications (2011) states 24°C as the design condition for a supermarket retail environment.

This loss of cooling from the cabinet to the store needs to be considered in the stores HVAC. For stores with remote cabinets, although a more comfortable environment will be apparent in winter, it is possible that air conditioning may be required during warm summer periods.

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## Dual port TEV

If a thermostatic expansion valve (TEV) is sized for a normal stable load, it may be undersized during pull down, therefore reducing the maximum capacity available. This can lead to product temperatures increasing after defrosts, door openings and loadings. If the TEV is sized for the peak load, then it will not work well at normal load, as it will never find a proper balance and will be constantly over and undercharging the evaporator (commonly known as ‘hunting’).

Dual-port valves (Sporlan, 2006) are designed to cover a larger range of capacities to satisfy both peak load and part load conditions. This is achieved by control of two independent flow ports; a larger port for periods of high load and a smaller port for periods of normal load. The TEV capacity is doubled when the larger port is fully open (Figure 5).

The consequence of a slow pull down after defrosts is that the maximum temperature of the product will rise higher than it would with a faster pull down. Due to the ability to have a larger capacity during pull down it may be possible to run at a higher evaporating temperature or set-point and still maintain the same maximum product temperature, saving energy. However, the benefits will only be seen if the heat loads on a cabinet vary greatly and this is not common in most UK supermarkets. It should also be noted that electronically controlled valves are less severely affected by hunting phenomena.

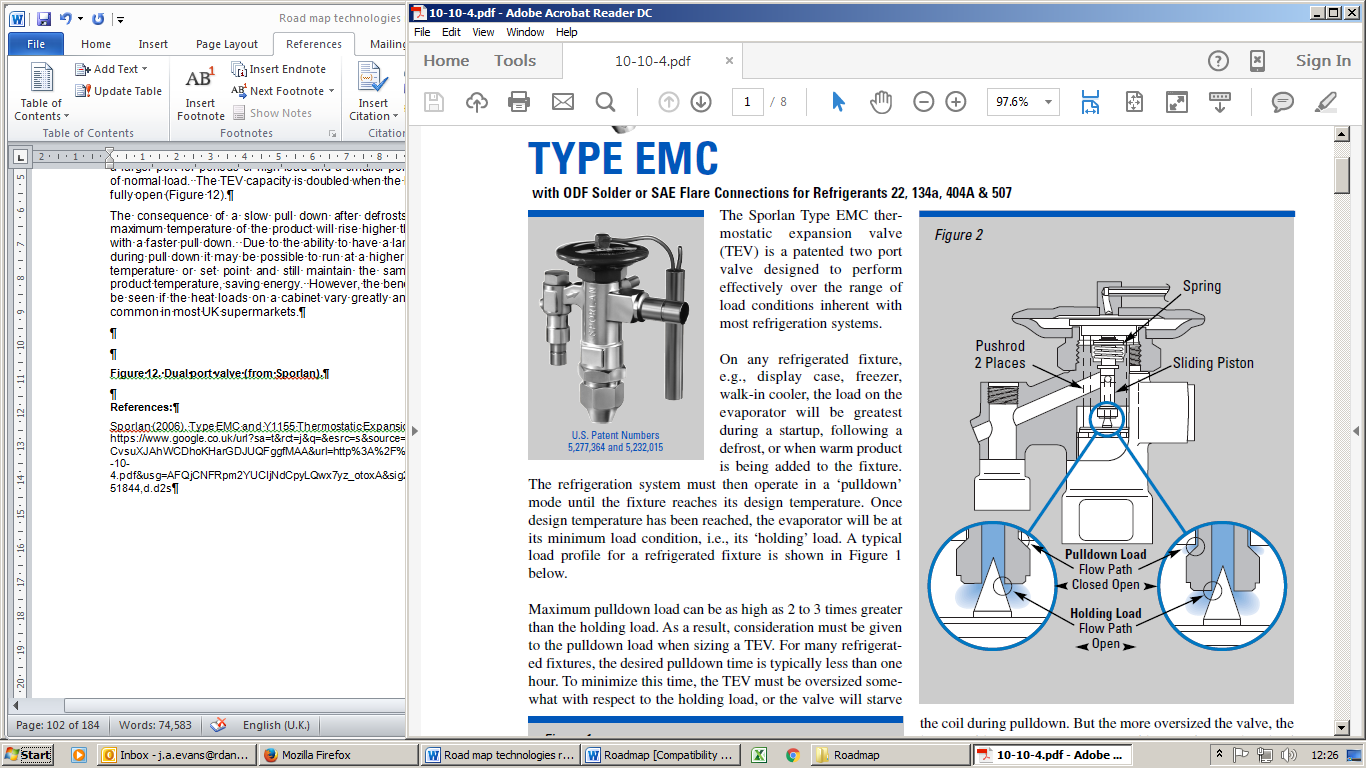


Figure . Dual port valve (from Sporlan).

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## Dynamic demand

Dynamic demand is typically a method by which the demand for electricity is reduced when the frequency of the electrical supply drops. When frequency drops, it means that energy generation is struggling to keep up with demand, so dynamic demand helps by effectively shedding load. By reducing the demand, it can avoid some of the use of higher carbon emission types of generation such as coal or oil-fired power stations which make up a large part of the generation backup when demand is high. In periods when generation exceeds demand, frequency will rise above its nominal average. At these times, energy-using systems which have storage capacity can be run longer to build up reserve capacity. For example, producing colder temperatures in refrigerated display cabinets (e.g. Pedersen et al., 2014) and cold stores can shift load while providing a buffer which can be used to balance off-cycling during peak times.

### Energy Saving Potential

Dynamic demand shifts energy consumption rather than reducing it. However, it can result in savings in emissions of associated CO2e as these vary greatly with the type of generation being used. During low demand periods, electricity is typically sourced from a low carbon mix of generators which includes various renewables. As demand increases, the more carbon-intensive types of generation are increasingly employed. For example, it is reported that in the UK in 2008, carbon intensity peaked at around 600 gCO2.kWh-1, which was almost 50% greater than during low demand periods on the same day (Hart-Davis, 2013).

Use of dynamic demand can also reduce energy costs, as energy used outside of peak demand periods is likely to be cheaper and penalty charges for high consumption can be avoided. In addition, larger energy users have options to contract with their electricity suppliers to allow certain systems to be switched off when appropriate to do so and when demand on the grid is high. Sophisticated versions of such contracts allow for trading of energy savings across markets for different types of generation.

### Current technology

Originally aimed more at domestic appliances such as refrigerators and freezers, dynamic demand has also been applied to industrial and business applications, including some in the supermarket sector. The technology can be relatively simple and cheap in its more basic formats, relying largely on unsophisticated sensing of frequency and simple control of equipment run-times. However, more complex formats are required for some applications, with a good example being the refrigeration systems responsible for maintaining sensitive food display temperatures.

Large scale upward or downward regulation of power supply and consumption using supermarket display cabinets and cold stores requires aggregation of multiple supermarkets (Pedersen et al., 2014). Smart meters and equipment controls installed at each supermarket are linked to centralised monitoring and control, offering flexible integration into the power grid. The control system must ensure that food temperatures are not allowed to rise to levels which promote bacterial growth or other quality deterioration, and for chilled foods it is also important that lower limits are not exceeded when the systems are forced to run longer (Hovgaard, Larsen, and Jorgensen, 2011). This could result in partial freezing and quality loss. Technologies such as phase change materials (PCMs) could be used to help to stabilise temperatures (Alzuwaid et al., 2015). A review by Joybari et al. (2015) considered how PCMs could be used for heat storage at condenser side or cold storage at evaporator side of domestic refrigerators.

A related consideration is whether there is sufficient scope to reduce temperatures from their normal operating levels, and whether the refrigeration system has enough spare duty to achieve the reduction.

### Cost

Capital costs are not well reported, and in any case may be included in contracts with third parties offering control and trading services such as those described in the next section.

### Case studies

Use of demand side management has been reported for supermarket chains. For example, the Guardian (2011) described the trialling of smart metering and demand management for heating and ventilating systems by J Sainsbury plc, a trial which has since been rolled out to multiple stores.

Smart grid systems and trading of energy savings are also available, such as the widely reported ‘Tradenergy’ system from Danfoss and Reactive Technologies (Anon, 2014).

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## Economisers

Economisers are mechanical devices intended to reduce energy consumption. In terms of refrigeration, an economiser uses an intermediate pressure on a single compressor to improve efficiency.

Traditionally economisers have been used on screw compressors but they can also be used on scroll or centrifugal compressors. These machines act like a two stage booster compression system. However, they only use one compressor with two inputs (the main suction and an economiser port at an intermediate pressure). This allows the benefits of a booster system with only one compressor.

Jousson (1988) simulated a twin-screw compressor with an economiser. The COP improvement at low pressure ratios was very small, in the magnitude of a couple of percentage points, but became significant at high pressure ratios. At a pressure ratio of 12.0, the improvement amounted to 16%.

There are two types of economiser setups; one uses a heat exchanger and the other a flash vessel.

In a heat exchanger system part of the refrigerant, typically 10-20%, is evaporated at an intermediate pressure from the economiser port. This evaporated refrigerant is passed through a heat exchanger to subcool the liquid refrigerant between the condenser and the expansion valve. This increases the capacity of the system by around 10% and reduces flashing. There is consequently a small increase in the power input caused by the transport of an additional mass flow rate; the increase in the system efficiency can be explained by considering that the additional compression work is taking place at a higher suction pressure, therefore with a higher efficiency. The heat exchanger can either be shell and tube or plate submerged coil. With the submerged coil, the liquid refrigerant passes through the coil in a submerged tank of boiling refrigerant.

With a flash economiser, all the liquid from the condenser goes through an expansion valve and into a flash vessel at an intermediate pressure (the pressure of the economiser port on the compressor). The flash gas is taken into the economiser port, whereas the saturated liquid is passed through another expansion valve into the evaporator.

According to Bellstedt (2015), at a condensing temperature of 23ºC and an evaporating temperature of -25ºC, economisers provide a 7% improvement in COP. At a condensing temperature of 23ºC and an evaporating temperature of -10ºC, economisers provide a 4% improvement in COP.

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## Ejectors

Two-phase ejectors for the purpose of expansion work recovery can theoretically be used with any refrigerant. However, their main use has been in CO2 systems. This is due to the large pressure differences in CO2 refrigeration systems and the resulting throttling losses, which are particularly high. Using an ejector enables kinetic energy to be recovered during the expansion process, which can then be used in a variety of different forms to increase the system COP. Refrigerant ejectors are inexpensive and reliable devices with no moving parts and just 3 connections which need to be made.

Ejectors have been used in different parts of the system. The most common ejector cycle is to use the ejector as a fluid driven pump to increase the refrigerant pressure at the compressor inlet to a level above the evaporation pressure. Other uses include utilisation of the ejector as a pump to provide liquid overfeed of the evaporator of the system. A typical simple ejector cycle, here for transcritical CO2, is shown in Figure 6 (Zhang and Tian, 2014). The idea of this standard ejector cycle dates back to a patent by Gay (1931). More complex versions of this cycle have been proposed and investigated. Li and Groll (2004)found that the COP of the ejector expansion transcritical CO2 cycle could be improved by more than 16% over the basic transcritical CO2 cycle for typical air conditioning operation conditions.Kornhauser (1990) showed a theoretical improvement of up to 21% in the COP using an ejector instead of a throttling valve in an R12 refrigeration cycle. However, only 3.8% COP improvement was measured by Menegay and Kornhauser (1996).

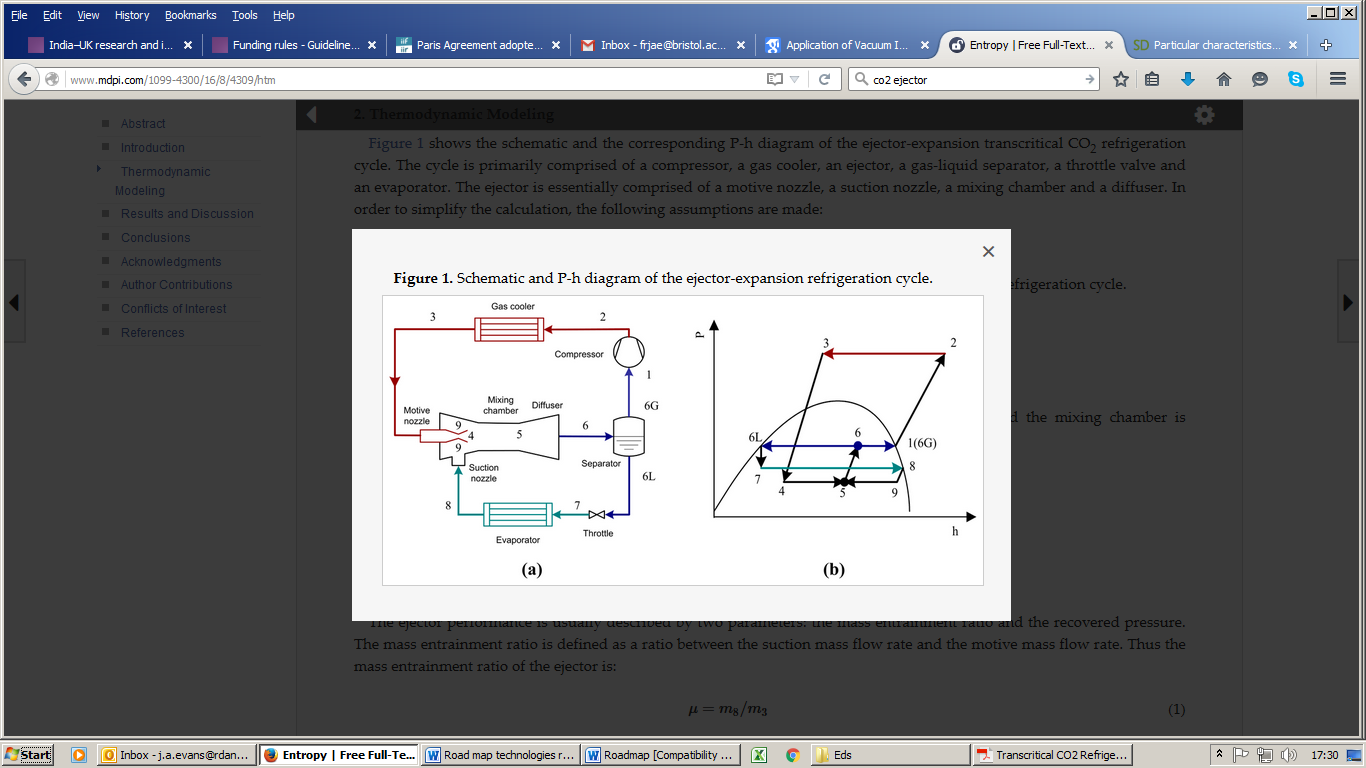


Figure . Typical simple ejector cycle (from Zhang and Tian, 2014).

A CO2 ejector cycle was installed in a supermarket (Migros) in Bulle, Switzerland, by Frigo Consulting Ltd in July 2013 (Frigo-Consulting Ltd, 2014). The system was a transcritical booster system (Figure 7). One of the MT compressors was shifted to become a parallel compressor and a low pressure receiver was added. They claimed a 15% increase in efficiency.

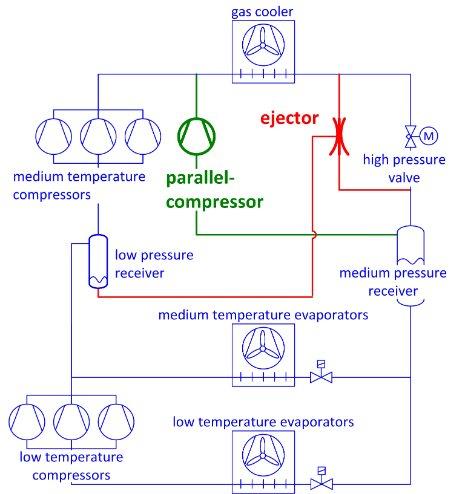


Figure . Ejector used to pump fluid from the low to high pressure receivers (from Frigo-Consulting Ltd., 2014).

Nekså, Walnum, and Hafner (2010) reported that the test result by Ozaki, Takeuchi and Hirata (2004) illustrates that using an ejector in car air-conditioning can give approximately 20% COP improvement. A simulation validated by experimental results (from a mobile air conditioning system) by Elbel and Hrnjak (2004) reveals that the use of an ejector in a CO2 transcritical systems yielded COP improvements on the order of 25% over a standard CO2 system at conditions relevant for air conditioning systems. Their study also showed that the use of an internal heat exchanger greatly affects the amount of expansion work that can be recovered by the ejector.

Hafner, Försterling and Banasiak (2014) created simulation models of a one-stage and multi-ejector system representing a supermarket test facility. The average daily predicted values for the relative COP increase due to the ejector system for cooling varied between about 5% and 55% for Trondheim, Norway in the summer and spring, respectively. Pottker (2012) also showed that the COP of the ejector system increased between 8.2% and 14.8% when compared to a conventional expansion valve system operating with R410A. The two major mechanisms of improvement of the ejector system were quantified separately: COP gains of between 1.9% and 8.4% were solely due to the work recovery; while liquid-feeding the evaporator alone was responsible for 4.9% to 9.0% of COP gain. Overall ejector efficiencies from 12.2% to 19.2% were achieved.

Ersoy and Sag (2014) compared a conventional and ejector R134a system experimentally under the same external conditions. Depending on the operating conditions, it was found that the work recovery in the ejector was between 14% and 17% lower. It was also found that the refrigeration system with an ejector as the expander exhibited a COP that was 6.2 to 14.5% higher than that of the conventional system.

A recent review paper by Elbel and Lawrence (2016) provides additional insight on ejector design and applications.

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## Electronic expansion valves

Electronic expansion valves (EEVs) are typically used to replace thermostatic expansion valves (TEVs). The benefits of EEVs over the TEVs include (Carel, 2006; Danfoss, 2007):

* A wider operating pressure range, allowing the possibility of lowering head pressures.
* Ease of commissioning.
* Data from controllers can be used as early detection of faults such as icing of evaporators, enabling preventative maintenance.
* Some EEVs are bi-directional and can replace the two valves required in reversible heat pumps.
* Some EEVs have a seal gasket on the orifice and can be used as a solenoid valve.
* TEVs can cause hunting.

There are two main designs of EEV; those which vary their orifice opening in an analogue manner to continuously regulate refrigerant flow, such as the Carel E2V (Carel, 2006), and those which adopt a pulse width modulation strategy to deliver refrigerant to the evaporator in short bursts over an orifice of a fixed size, such as the Danfoss AdapKool range (Danfoss, 2007).

There are two common control strategies; temperature-only based on temperature probes at the evaporator inlet and outlet (Danfoss, 2007); and temperature/pressure which measures both pressure and temperature at the evaporator exit to more accurately determine the actual superheat (Carel, 2006).

It is widely reported in the industry that savings provided by EEVs result from tighter superheat control, enabling warmer evaporating pressures. Tassou and Al-Nizari (1993) showed that from a cold start (when the refrigeration system is first switched on, after a long period of shut-down) a TEV oscillated between 1 and 12ºC for a considerable time. An EEV in the same conditions oscillated between 2 and 12ºC prior to settling down to smaller oscillations of within ±1°C from the set value.

One of the major benefits of EEVs to energy saving is the ability to drop the head pressure. This could result in savings, but only in the cooler months of the year (Lazzarin, Nardotto, and Norro, 2009). A bypass capillary or parallel TEV with control to select a TEV with a larger orifice when head pressure drops could provide the same end result (but may be more expensive).

In normal operation, the savings realised by EEVs are largely a result of reducing the system head pressure below that which would be possible with a TEV (Lazzarin, Nardotto, and Norro, 2009); such savings would not be realised in test room conditions where a relatively high condensing pressure is maintained in order to comply with the test standard (BSI, 2012) and a fixed condensing temperature is assumed in the final calculation of TEC (Total Energy Consumption).

Maintenance and commissioning benefits of the electronic valves can also be valuable to the end user in practice. With an electronic valve, the settings can be controlled and altered automatically and remotely from the valve. The valve parameters can even be monitored and used to generate early warnings of service or maintenance issues with the cabinet or plant (Carel, 2006).

### Energy Saving Potential

Lazzarin, Nardotto, and Norro (2009) reported savings of 22% and 18.5% from two case studies drawn out of a one year field trial of expansion valves on supermarket chilled and frozen display cabinets. However, these savings were due to the reduction in minimum head pressure from 16 to 12 bar. Carel marketing material (Carel, 2006) reports savings of 15 to 20% (together with floating condensing pressure control) with peaks up to 30% at some times of the year (based on case studies with commercial refrigeration units and in a telecom control room).

### Cost

For their payback analysis Lazzarin, Nardotto and Norro (2009) estimated the uplift in cost of fitting an EEV over a TEV as €350 per retail display cabinet. For larger plants, the Carbon Trust quotes a cost of £2000 + £1000 installation on a 100 kW chiller.

### Case studies

Lazzarin, Nardotto and Norro (2008) evaluated the efficiency benefits of EEVs over TEVs by altering the cabinets and refrigeration plant in a supermarket (near the North Coast of Italy) to cyclically operate for 24h with TEV then 24h with EEV. The energy and refrigerant flow parameters were all logged for one year. The dataset was then extended using computer based modelling to forecast savings in other areas of Italy and it was found that savings of 18.5 to 22% were likely, depending on the location (and resulting ambient temperatures).

The comparative study between EEV and TEV in supermarket refrigeration display cases by Tahir and Bansal (2005) revealed that EEV provided better evaporator pressure and cabinet product temperature, and less ice accumulation prior to each defrost period. This resulted in air curtain velocity being consistently higher (by about 16% than TEV), while decreasing both the defrost frequency and time.

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## Energy efficient fan motors

In the past shaded pole single-phase (SP) alternating current (AC) motors were the norm. Small SP motors have efficiencies between 19 and 25% (Walker et al., 2004) while permanent split capacitor motors may reach 50% at full-load.

Electronically commutated motors (ECMs) convert the AC input voltage to a DC voltage for the motor. Goetzler et al. (2013) quote a typical maximum attainable motor efficiency of 70% for a brushless direct current (BLDC) motor. A BLDC motor maybe the same as an EC motor, but may also be one with a DC supply. Faramarzi et al. (2004) state an approximate efficiency of 80% for a 1/3 hp dual shaft ECM. NCI/PNNL (2011) quote state-of-the-art EC motors as 66% efficient.

EC motors are so efficient that they run at lower temperatures, extending the life of the motor. They are also quieter.

ECMs maintain their efficiency at low speed. By combining an EC motor with a suitable controller, the motor can operate at a reduced speed with little reduction in efficiency, reducing power input. As ECMs can be operated at different speeds, one motor can replace many AC motors, reducing inventory.

According to Becker and Fricke (2016), a permanent magnet synchronous (PMS) AC motor that can directly use grid-supplied AC current without the need to rectify to DC has recently been commercialised. Field demonstrations showed PMS motor technology is approximately 34% more energy efficient than existing EC motors and nearly 79% more energy efficient than shaded-pole motors. In addition, the new motor exhibits a power factor of approximately 0.83, which is on average 40% greater than that of existing evaporator fan motors.

### Condenser fans

Condenser fans are used to cool the condenser by passing ambient air through it. For remote plants these will be either in a plant room or outside. For integral cabinets, these will be part of the cabinet refrigeration system. ECM fans for integral cabinets will reduce the heat load to the store. This would be beneficial in a store where heat needs to be extracted. However, in a store which is heated with electrically resistive heating, there would be no benefit in more efficient fans

However, stores are normally heated by gas which is more efficient regarding CO2 emissions and cheaper than electric heating.

Variable speed condenser fan motors provide the option to reduce speed, reducing motor power, or increase speed, reducing condensing temperature (increasing COP). An algorithm can be set to allow the fans to operate at the optimum speed.

Westphalen et al. (1996) state that 3% of the energy consumption of an average size supermarket is from the condenser fans. These are most likely the remote condenser fans, not including the ones contained in integral cabinets. This amounts to 7% of the refrigeration energy consumption. They quote percentage savings and payback for changing to condenser fan ECMs for different commercial refrigeration applications. For beverage coolers, savings of 4.5% of the total refrigeration electricity consumption and a 4.4 year payback were shown. For reach-in freezers, the total refrigeration electricity consumption and payback were 2.7% and 2.2 years, reach-in refrigerators 3.3% and 2.0 years, ice machines 5.4% and 2.2 years, and refrigerated vending machines 3% and 8 years.

### Evaporator fans

Evaporator fans are used to pass air through the evaporator and also to provide convection within the cabinet and an air curtain for open cabinets. The electric motors are within the cold space; therefore, all of the energy of the motor is a heat gain on the cabinet. This means that a more energy efficient motor will save energy in both the motor itself and the refrigeration compressor.

SP pole motors are the simplest and least expensive type of motor for this application, hence why they were previously the most common. At the size used in evaporator fans, they are typically between 19 and 25% efficient (Walker et al., 2004).

Westphalen et al. (1996) quote percentage savings and payback for changing to evaporator fan ECMs for different commercial refrigeration applications. For beverage coolers, savings of 29% of the total refrigeration electricity consumption and a 1.4 year payback were shown; for reach-in freezers 2.3% and 2.6 years, reach-in refrigerators 7% and 2.1 years, and refrigerated vending machines 14% and 1.8 years.

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### Evaporative condensers

Evaporative condensers are specifically designed to be simultaneously exposed to water spray and fan assisted air flow. Heat is rejected at the dew point temperature of the air (rather than at the dry bulb temperature). This reduces condensing temperature and therefore saves energy.

Evaporative condensers have a greater benefit where ambient temperatures are high, leading to a high condensing temperature/pressure. They work better in hot dry climates where the difference between wet bulb and dry bulb temperature is high.

German et al. (2012) show that the evaporative condenser performs better than a similarly rated air-cooled condenser at dry bulb temperatures above 35ºC in a dry climate. Varzaly (2012) calculated that, compared with an air cooled condenser, an evaporative condenser at design conditions of 38°C dry bulb or higher and 27°C wet bulb (location in the Miami USA area) reduced the compression ratio of the low temperature system by 18% and the medium temperature system by 23%. The extra cost of the evaporative condenser was somewhat offset by the reduced cost of the compressors. The overall equipment costs were typically increased by 5 to 10%. The return on investment for the evaporative condenser system was less than 2 years. Yanyuan and Guilong (2008) found that the rate of heat rejection, the COP and the refrigerating capacity of evaporative refrigeration systems were 23.9%, 14.3% and 30% greater respectively than those of air-cooled refrigeration systems when the evaporation temperature was -24°C.

Clark and Gillies (2014) stated that evaporative condensers are more expensive both to install and operate than air cooled alternatives below approximately 500 kW duty. For larger systems evaporative condensers offer a lower first cost and good refrigeration plant efficiency at peak ambient conditions.

Evaporative condensers have higher maintenance costs than conventional condensers and require any water to be dosed to eliminate *Legionella* bacteria. They use a lot of water, and this water must be periodically bled off into the sewer to prevent the accumulation of excessive mineral deposits, such as magnesium, silica and calcium. These minerals build up on the condenser coils, impairing heat transfer and thus efficiency. On average, half of the water used by a typical supermarket that has these devices is used in the evaporative condensers, usually amounting to more than 1.5 million gallons (5,700 m3) per year per store, depending on local climate and the size of the supermarket (Alliance for Water Efficiency).

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## Expansion machines (e.g. turbines, not including vortex tubes)

Expanders are widely used in the refrigeration of gases in industrial processes such as the extraction of ethane and natural gas liquids from natural gas and the liquefaction of gases (such as oxygen, nitrogen). They tend to be used in cryogenic temperature processes such as the air cycle. The work created in the expansion process can be usefully used, e.g. to power a generator or to provide work directly to the compressor. Furthermore, the isentropic expansion in the expander drastically lowers the temperature of the gas in comparison to an isenthalpic throttling process. However, these processes are single-phase gas processes. The vast majority of refrigeration systems are two-phase and therefore the expander needs to be able to handle two-phase flow that is generated when the high pressure liquid partially flashes into a gas during the pressure reduction. A two-phase expander is thus required to replace the isenthalpic expansion process of a throttling expansion valve with an isentropic expansion process.

The thermodynamic cycle efficiency of transcritical CO2 systems is lower than that of conventional vapour compression systems. However, recovering the losses that occur during the expansion process of the refrigerant between the exit of the high-pressure gas cooler and the evaporator offers the ability to substantially improve efficiency. In theory, recovery of energy lost in the expansion process of a vapour compression cycle is of interest for any refrigerant. However, due to the large losses of expansion when using CO2, attributable to the high differences between high and low side pressures during operation, a work recovery device is particularly important (Westphalen and Dieckmann, 2004).

Most refrigerant expanders are based on scroll or vane technologies. Westphalen and Dieckmann (2004) developed a scroll expander design for use in CO2 air-conditioning cycles operating in high ambient conditions. They estimated the overall efficiency of the expander was 72% and would reduce power input by 20% by providing some of the work of compression (using an integrated compressor/expander unit). Fukuta et al. (2009) developed a rotary vane-type expander which was small-size, light-weight, and had a simple structure. They measured a maximum total efficiency of 60% for the expander obtained at the rotational speed of 2000 rpm.

A free piston expander utilizing the full-pressure principle was successfully introduced by Heyl et al. (1998) and Nickl et al. (2005). Their device was a combined compressor-expander unit which was installed in addition to a main compressor. Due to expansion work recovery and intercooling of the suction gas entering the second compression stage, a theoretical COP improvement of up to 45% was reported.

There is no published information on commercial implementation of this technology. However, there are several research works that have evaluated the use and efficiency of turbo expanders. Brasz (1995) assessed a refrigeration system of a capacity of 350 to 3,500 kW, employing a relatively high-pressure refrigerant such as R22 or R134a, and a centrifugal or screw compressor driven by a two-pole induction motor (3000 to 3600 rpm) with a turbine efficiency estimated at about 60%. Depending on operating conditions, the turbine reduced the motor load by 6-15% compared to the system with a throttling expansion valve. Efficient energy recovery could also be achieved if the turbine expander was employed in a refrigeration system below 350 kW capacity that has a screw compressor or other type of rotary compressor. Kohsokabe et al. (2006) studied a transcritical CO2 refrigeration cycle for a small capacity air-cooled chiller with a scroll type expander and a rolling-piston type rotary sub-compressor connected with a shaft. In experiments, the expander-compressor unit was shown to be stable and to improve the COP of the CO2 chiller cycle. The test results indicated that the COP improvement of the cycle was more than 30%, while the total efficiency of the expander-compressor unit was 57%.

Robinson and Groll (1996) showed through numerical modelling that under certain conditions a refrigeration cycle based on R22 will perform better than a CO2 refrigeration cycle that uses a valve for expansion. However, a CO2 cycle that employs an expansion turbine under the same conditions performs as well as, or better than, the R22 baseline cycle.

Practical problems that are encountered with compact expander units that are integrated with the compressor are related to parasitic heat transfer losses between the hot and the cold side, leakage and oil management issues in machines that share a common shaft, and off-design operation due to the coupled design of a fixed compression volume with a fixed expansion volume.

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## Fan motor outside of cabinet

Evaporator fan motors are normally mounted within the cold space of the cabinet; therefore all of the energy of the motor is a heat gain on the cabinet. Mounting the electric motor outside of the cabinet but the fan inside the cabinet, by means of a shaft passing through the insulation of the cabinet allows the heat to be dissipated outside of the cabinet. Servicing of external motors may also be simpler. However, a shaft going through the insulation could cause air and moisture ingress problems.

In most cabinets the heat load from the fans will be approximately 5% of the total heat load (ASHRAE, 2001). Therefore mounting the motors outside of the cabinet would reduce the refrigeration energy consumption by 5%. The direct energy used by the motors may however be higher when mounted outside the cabinet as the warmer air passing over the fan motor may reduce the efficiency of operation.

**Reference:**

ASHRAE Refrigeration Handbook 2001.

## Flooded evaporators (added to R744)

To get the maximum efficiency of an evaporator, it can be operated with a liquid overfeed (flooded). Unlike in a direct expansion system, the fluid at the exit of the evaporator is not superheated, it is a two-phase mixture with a small volume of liquid mixed in with the gas. The main advantage of operating an evaporator in a flooded state is that the temperature difference between the air temperature and the evaporating temperature can be greatly reduced since there is no requirement to heat the suction gas above its evaporating temperature. In medium temperature display cases for supermarkets, where the air temperature is about 2°C, it is common to see an evaporating temperature of -8°C when superheated gas is required, whereas with a pumped CO2 system an evaporating temperature of -2°C can be readily achieved. The main disadvantages of flooded systems are that a method of separating the liquid overfeed from the suction gas is required before it reaches the compressor, and the return of lubricating oil from the evaporators to the compressor, which is usually easy in a DX system, may require some additional complex equipment or maintenance routines.

Flooded systems are common on industrial but not commercial systems. In a single gravity flooded evaporator, liquid from the condenser is metered into the evaporator via a header using a float valve. Boiled vapour returns to the compressor at the top of the header.

The header can be replaced by a surge drum/accumulator which could feed more than one evaporator. The high pressure of CO2 needs to be considered with regard to the surge drum. A pump can be used instead of gravity to transfer liquid to the evaporators.

Flooded evaporators are used on secondary pumped CO2 systems. According to Johansen (2006) it is possible to save up to 30 to 40% with R744 supermarket systems.

Reulens (2009) reported an increase in COP of between 2 and 5.5% for a flooded evaporator freezer system compared to a DX R744 system at 40 and 10ºC ambient temperatures respectively.

**References:**

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## Heat exchanger design

The majority of evaporators and condensers in supermarkets are fin and tube heat exchangers. Evaporators are a significant part of the overall cost of refrigerated cabinets and therefore less efficient evaporators are often used to keep the cost of the cabinet low. The same is the case for condensers on integral cabinets. Space is also an issue for refrigerated cabinets. Display area and loading volume are at a premium, and smaller heat exchangers are often used.

The heat flow to or from the evaporator or condenser is governed by the overall heat transfer coefficient, the surface area of the heat exchanger and the temperature difference between the refrigerant and the heat exchange fluid. In retail cabinets the space available for fitting an evaporator or condenser is limited, so a more effective evaporator or condenser is required to make up for the limited size. This will lead to an increase/decrease in the evaporating/condensing pressure respectively, and will lead to a reduction in the compressor power consumption (Chandrasekharan and Bullard, 2004a).

### Evaporator optimisation

Optimising the design of display cabinet evaporators has been demonstrated to achieve compressor energy savings of 28% (Chandrasekharan and Bullard, 2004b). The main reasons for the better performance were the better frost distribution, higher air-side area due to the larger depth of the coil and higher fin density.

Sarhadian et al. (2004) carried out an experimental evaluation of a number of viable and near-term energy-efficiency solutions to an open vertical refrigerated display case. They improved the evaporator effectiveness as a function of the product of overall heat transfer coefficient and surface area (UA) by 68% over a baseline evaporator by increasing the number of circuits, increasing the number of fins, reducing the tube diameter and enhancing the internal tube surface. Use of a micro-channel LSHX resulted in roughly a 19ºC increase in total system subcooling. The prototype case used a dual port mechanical TXV. This allowed for faster temperature recovery than in the baseline case during post-defrost periods. It also allowed for more effective utilisation of the evaporator by operating at 3ºC superheat, which was 2ºC less than the baseline. The evaporator had an 18% increase in refrigerating effect (mainly due to the increase in UA, but also partly due to the improved suction-liquid heat exchange).

It is worth considering that the optimum evaporator for a horizontal open and a vertical glass door refrigerator maybe different. Getu and Bansal (2007) used a mathematical model of evaporators of low-temperature supermarket display cabinets based on various empirical correlations of heat transfer coefficients and frost properties in a fin-tube heat exchanger in order to predict the performance of the display cabinets under frosted conditions. They showed that the frost thickness and the frost thermal resistance on the horizontal open evaporators were higher than that of vertical glass door evaporators for the measured store relative humidity (33 to 41%) and temperature (24.1 to 26.7ºC). However, the air pressure drop was far higher for vertical glass-door than horizontal open cabinets. This higher pressure drop in the vertical glass door cabinets was mainly due to higher mass flow rate of the air (14.66 kg/s) and smaller dry free-flow area of the evaporators (3.1 m2), as compared to 1.84 kg/s and 4.4 m2 for the horizontal open cabinets.

### Micro-channel heat exchangers

The use and understanding of transport phenomena in micro-channel heat exchangers is still developing and is currently a topic of much research effort (Shiferaw et al., 2009). Micro-channel heat exchangers have small channels between plates, giving them better rates of heat transfer (effectiveness). As these heat exchangers are often less deep in the air flow direction, they offer less resistance to air flow and therefore can reduce fan power. These heat exchangers have been used for some time in the automotive industry for air conditioning due to their compact nature and low weight. In addition to enhanced heat exchange, micro-channel heat exchangers weigh less and by being smaller than conventional heat exchangers can aid reduction in refrigerant charge. Reducing refrigerant charge can potentially enable less safe, more environmental friendly or more efficient refrigerants to be applied.

According to Danfoss (2015) micro-channel heat exchangers improve COP by around 10% and require 30% less refrigerant. However, this percentage of increment in COP is linked to the particular design of the unit and can be lower for an enhanced design of the classical “round-tube-and-fin” coil. Furthermore, the charge reduction can be lower; it mostly depends on the design of the manifold.

Frost formation can block evaporators (especially in freezers) and so gaps between pipes and fins need to be larger on evaporators than on condensers and air conditioning evaporators. This limits the effectiveness of the evaporator. For this reason micro-channel heat exchangers tend to be more suited to condensers and evaporators which do not generate frost. With careful design microchannel evaporators can be used under frosting conditions. However, defrosting issues must be considered (Carlson, Hrnjak and Bullard, 2001). Another issue with micro-channel heat exchangers is refrigerant maldistribution which to date has been an issue preventing their use as evaporators (Kulkarni and Bullard, 2013).

Currently micro-channel heat exchangers are costly to fabricate. The cost of extruding and brazing tubes to the header is higher than that of manufacturing louvered fins (Kulkarni & Bullard, 2013).

### Heat exchange rifling

Rifled-tube heat exchangers were introduced in the early 1980s. Rifling can be used in both condensers and evaporators. Rifling increases the wetted surface area and increases the surface heat transfer coefficient by inducing turbulence. The improvement in heat transfer is due mainly to the 50% to 60% increase in internal surface area over plain tube, thereby reducing the liquid refrigerant film thickness (ACHR, 2000). The tube-side heat transfer coefficient for rifled tubing is as much as two and a half to three times that of plain tube for condensing and evaporating conditions (ACHR, 2000). Rifled tube has a more beneficial effect on evaporators than condensers - the reason for this is twofold—1) the increase in evaporator performance has a larger effect on the system than the same increase in condenser performance and 2) an evaporator coil has a higher percentage of its internal surface area in two-phase flow than a condenser does.

The effect is more pronounced in small-diameter tubes — those smaller than the typical 3/8-in. tubing used in heat exchangers — because more surface area is available to groove for a given volume (Cotton, 2012). A proper heat exchanger design should take into account the slightly higher pressure drop.

Rifled tube is more expensive to produce than plain tube; however, the heat exchanger could be smaller and use less material or the system more efficient and use less energy. Due to a lower overall effect on system performance, condensers are not as likely to justify rifled tubing.

### Enhanced internal heat transfer (micro-fins)

Internal finning can aid evaporation and condensation inside the tubes. One approach is to use small spiral fins (micro-fins). The benefits of this are increased heat transfer, with a consequential increase in pressure drop.

When used with R22, Khanpara (1986) found micro fins increased evaporation/condensation by up to a factor of 3.8 with an increase in pressure drop of up to 1.8 times the original. Ponchner (1995) showed average condensation enhancement factors ranging from 2.0 at a low mass flux to 1.4 at a high mass flux for R134a in an 18° helix angle, 0.375" o.d. micro-finned tube. The micro-fins increased pressure drop by a factor of 1.19 to 1.26. Jiang et al. (2016) showed for the micro-fin tube, the average heat transfer coefficients of R22, R134a, R407C and R410A are 1.86, 1.80, 1.69 and 1.78 times higher than those of the smooth tube. The average pressure drop of R22, R134a, R407C and R410A for the micro-fin tube is 1.42, 1.30, 1.45 and 1.40 times higher when compared with that for the smooth one.

The benefits of increased heat transfer in reducing temperature difference between refrigerant and air needs to be compensated with the negative effect of the pressure drop on compressor power.

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## Heat from light outside cabinet

If lights are within the cold space of the cabinet, the heat generated by the light will be a heat gain on the cabinet. Open fronted cabinet lights are often mounted outside of the cold space for this reason and also because fluorescent lights do not work well at cold temperatures. However, it is difficult to position glass doored cabinet lights outside of the cabinet as light will reflect off the glass and therefore not illuminate the inside of the cabinet.

Two solutions present themselves.

1. To mount the light outside and shine the light inside using some type of reflective duct. The authors are not aware of any work in this area.

2. To mount the light inside and remove the heat.

Olivewood Chil-LED (OliveWood) offers the second of the solutions for LED lights in cold storage. They use a heat pipe to transfer the heat from the LED through the insulation and outside of a cold room. Olivewood claims that 70% of the energy consumption of an LED is lost as heat, although it is not clear how much of this heat is extracted out of the cabinet. ASHRAE (2001) reports that depending on the cabinet type, the heat load from lighting can be between 2 and 10%.

This system is ready to use in cold rooms. However, extra development would be required to adapt it to display cabinets.

**Reference:**

ASHRAE Refrigeration Handbook 2001.

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## Heat pipes

In supermarket cabinets, temperatures of products vary according to their position within the cabinet, with the warmest packs normally at the front of the cabinet, as they are more exposed to the ambient environment. Cabinets must be run to keep the warmest product within microbially safe levels. This means that other products are often over-cooled, increasing the energy consumption of the cabinet. Heat pipes offer a potential way to move cooling to warmer areas. Thus the temperature difference in the cabinet could be reduced, allowing set-point temperatures to be raised to save energy.

Maidment et al. (2005) developed a finite difference model to study the application of a heat pipe in a refrigerated retail display cabinet. The results indicated that heat pipes could provide improved heat transfer, which could contribute to lowering core food temperatures by approximately 2.5 to 3.5 K.

Heat pipes have been theoretically shown to improve the performance of display cases both in terms of maintaining temperatures within a narrower band and saving energy. Practical demonstrations of this by different authors have provided very different results.

Lu et al. (2010) fitted a display cabinet shelf with heat pipes with an inclined condenser in one end and showed a reduction in food core temperatures of approximately 3.0 to 5.5 K. However, this result could not be repeated in work by Foster, Orlandi, and Evans (2014). They showed a reduction of only 0.2 K at the centre of a Tylose test pack in a glass door cabinet using a heat pipe directly underneath the pack. This gave a predicted reduction in energy consumption of 1% based on an increased set-point of 0.2 K. A computer model was used to predict the benefit for a poorly performing open fronted cabinet. This showed a reduction in temperature of 0.7 K and a predicted reduction in energy consumption of 3.5%. The heat pipes were hand-made to order at a cost of £82.60 each for 10 units. The number of heat pipes required is dependent on where warm spots may exist on the shelves. However, if it is assumed that heat pipes are only used where ‘m’ packs are located on shelves (not the well) in an EN23953:2005 standard test (this allows the cabinet to perform better in a test, but not necessarily in a real environment), this would lead to 9 heat pipes at a cost of £743.40. As this is a significant additional cost for the cabinet, this would only be cost-effective if the price of the heat pipe was reduced dramatically due to mass production.

Jouhara et al. (2017) used flat heat pipe technology to produce a heat pipe shelf. The outcome of their tests showed that the heat pipe homogenised the temperature profile of the products and reduced the electrical energy consumption of the cabinet by approximately 12%.

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## Hydrophilic and hydrophobic coating on evaporators

Frost build up on evaporators can block the evaporator, causing reduced heat transfer and poor air flow around the cabinet. Hydrophilic or hydrophobic coatings can be applied to the outside of evaporators to effect the way water condenses, freezes and drains.

Jhee, Lee and Kim (2002) found that hydrophilic surface treatments mainly influence the frosting behaviour, whereas hydrophobic surface treatments have some influence on the defrosting behaviour. For practical use hydrophobic surface treatments were more effective in view of the defrosting efficiency and time. The defrosting efficiencies of hydrophobic heat exchangers were enhanced by about 10.8% whereas hydrophilic surface-treated heat exchangers were enhanced by about 3.5%.

Hydrophilic coatings were shown to reduce wet side pressure drop by all the authors presented below. However, one author showed an increase in dry side pressure drop with Hydrophilic coatings.

Shin et al. (2000) studied fin plates used in refrigerator evaporators. They found that the surface with better hydrophilicity provided a lower water holdup and, in turn, a more efficient defrost cycle. The study on the geometry of hydrophilic surfaces on water holdup revealed that fewer and sharper edges of hydrophilic surfaces result in less water holdup.

Wu and Webb (2001) showed that hydrophilic coatings should not be applied to the surfaces of evaporators that experience freezing conditions. If the evaporator must operate with condensate on the fins, but no frosting, e.g. in air conditioning, a hydrophilic coating is clearly preferred. This is because the hydrophilic coating will yield a lower wet surface pressure drop than an uncoated surface.

Wang et al. (2001) compared the airside performance of fin-and-tube heat exchangers in wet conditions with and without hydrophilic coating. They showed the pressure drops for the hydrophilic coated surface were lower than the corresponding un-coated surfaces. A maximum 40% reduction is observed for plain fin geometry.

Liu and Jacobi (2006) found that hydrophilic coatings hold more water when there is no air flow. However, they hold less water as the air flow is increased compared to non-coated evaporators. The coatings also reduce the wet pressure drop (pressure drop through the coil when it is coated in water) significantly without decreasing the wet sensible heat transfer coefficient.

Truster et al. (2014) also studied the surface frost properties and defrosting effectiveness of a metallic heat transfer surface. They showed that non-coated surfaces accumulated 60-90% more frost mass than the hydrophobic surface during the same allotted time period. They also showed that the hydrophobic surface consistently removed more water than the non-coated surface during defrosting.

The benefits of hydrophilic coatings to heat transfer are unclear, with two authors showing a slight benefit and another showing a reduction in performance. This may well be down to the layout of the heat exchanger and its operating conditions.

Wang et al. (2001) compared the airside performance of fin-and-tube heat exchangers in wet conditions with and without hydrophilic coating. They showed that heat transfer was lower for hydrophilic coating than an untreated evaporator. The degradation of heat transfer performance may be up to 20% for fin pitches of 1.2 mm.

Min et al. (2011) studied the performance of hydrophilic treatments on copper finned tube evaporators. They showed that the hydrophilically treated evaporator tended to yield a greater cooling capacity than an untreated evaporator but the increment was small. This may be partially attributed to the limited amount of condensate retained on the fin surface of the evaporator.

Kim and Kang (2003) studied the effects of hydrophilic surface treatment on evaporation heat transfer at the outside wall of horizontal tubes. They found superior evaporation heat transfer performance for hydrophilic coatings, although optimisation of the thickness of the coating was essential.

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## Improved axial fans

Currently most refrigerated display cabinets are fitted with axial fans, the blades of which are made from metal or plastic. Metal blades will have a uniform blade thickness as they are made from pressed aluminium or steel, whereas plastic blades offer the ability to vary blade thickness as they are produced by moulding.

According to Jackman (2007) simple profile-less blades have an efficiency of 50-60%, whereas profiled blades have an efficiency of 60-70%, crescent shaped blades have an efficiency of 70-75% and winglet tipped blades have an efficiency of 75-80%. Axial blades can be better optimised with modern numerical techniques (e.g. neural networks, CFD) to increase their performance and efficiency.

Yang, Ouyang and Du (2007) optimised the design of a low pressure axial fan with skewed blades. The blade was optimised such that the circumferential skewed angle of the optimised blade was 6.1 degrees. The total pressure efficiency was increased by 1.27%, and total pressure rise was increased by 3.56%. The stable operating range of the optimised impeller was extended to more than 30%. Aerodynamic noise was also reduced.

Parker, Sherwin and Hibbs (2005) designed, fabricated and tested improvements to an air conditioner outdoor unit fan system. An improved high efficiency fan design and advanced exhaust diffuser section reduced fan motor power requirements by approximately 49 W (26%) while providing superior air flow.

**References:**

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## Improved cabinet loading

In controlled test room conditions the authors have seen an increase in refrigeration load of up to 60% by altering a cabinet from 6 shelves to 4 with a corresponding increase in the warmest temperature of 2°C. Similar data from an integral multideck cabinet showed an increase of 39% in direct energy when changing from a 5-shelf to a 3-shelf layout and an 8-foot (2.5 m) remote cabinet showed a 31% increase in total energy consumption when going from a cabinet which was empty except for M-packs to a fully loaded cabinet. Therefore cabinet energy consumption and temperature control are strongly influenced by the choice of shelf layout and product loading (which will clearly alter during normal use as customers remove products).

Volume of product loaded into a typical multideck cabinet not only influences the energy consumption but also stored product temperatures. Again, in test room conditions, the authors have seen cabinets well optimised for a minimal loading fail to hold product temperature with a full loading, which implies that more intelligent control of cooling may be advantageous in a store situation. Furthermore, measured energy consumption of an 8’ remote cabinet in a test room situation showed an increase of 30% when the loading was increased from 20% to 75% and control settings were adjusted to ensure similar product temperatures for both cabinet loadings. It is very likely that the most efficient loading for a display cabinet is a function of the air circulation (volume of air circulation and ratio of back panel flow to supply air grille flow) and that control set-points will need to be altered to suit any given combination.

Loading will be in the hands of the shoppers as much as the supermarket. The cabinet needs to work both full and near empty but, because the loading influences the back panel flow and thus the support of the air curtain, energy consumption is inevitably affected.

### Efficiency

Up to 60% variation in refrigeration duty has been observed through shelf positioning and cabinet loading alone; redesigning the cabinet to tolerate variable loading would be beneficial to maintaining product temperatures under any conditions. Efficiency savings would be difficult to quantify as any alteration needs to ensure that the lowest energy condition is achieved as often as possible. Some consideration should be given to how representative any energy test condition is of normal use; it may be noteworthy that the lowest energy shelving configuration is typically that used for testing. Most cabinets would require alterations to their control parameters as the cabinet loading is varied.

## Improved cabinet location

Locating display cabinets in a cool, draught free and shaded position will keep energy consumption low. In a supermarket, display cabinets tend to be shaded from direct sunlight and the effect of radiation from ambient lighting is likely to be lower than that from the cabinets themselves.

Open vertical refrigerated display cabinets in the centre of aisles facing other refrigerated cabinets will generally work much more efficiently than cabinets at ends of aisles or outside of the refrigerated areas. This is because display cabinets cool and dry the ambient air around them, such that they then gain less sensible and latent heat from their surroundings compared to a cabinet outside of the refrigerated aisle. According to Ligthart (2007) the temperature in the chilled aisle is on average 15.5°C as opposed to 21°C elsewhere in the store and RH can drop to 20% when it is 70% in the rest of the store. Foster and Quarini, (2001) measured air temperatures in the aisles of 3 different supermarkets. Average temperatures varied between 12.5°C and 16.1°C.

Draughts have a significant effect on open fronted cabinets as they disrupt the air curtain. It is therefore important to keep them away from doors and ventilation systems. Chen and Yuan (2005) investigated the effect of different air speeds parallel to the length of the display cabinet, ranging from 0.2 to 0.5 m.s-1. The results showed that an increase in longitudinal velocities caused the inside temperature to rise slightly but the impact was very limited. The variations of heat load were also small.

D'Agaro, Cortella, and Croce (2006) used a CFD model to compare the effect of the parallel air flow defined in EN23953 (0.1 to 0.2 m.s−1) with that of no air flow and predicted that ambient air movement affected the cabinet return air temperature and consequently the refrigerating power, which was estimated to increase by about 30% when parallel airflow was applied.

Gaspar, Gonçalves, and Pitarma (2011) showed the total heat transfer rate increased by 5% when the air movement (0.2 m.s-1) direction moved from parallel to oblique and increased by an additional 1% when it changed from oblique to perpendicular. They also showed that when the magnitude of ambient air velocity was increased from 0.2 m s−1 to 0.4 m s−1, the total heat transfer rate increased by 53%.

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## Improved glazing

Refrigerated display cabinets with doors use glass to restrict heat transfer from infiltration, and radiation. Glass doors can be made of single, double or triple glazing with or without a low emissivity coating. Chilled cabinets tend to have single glazing and frozen cabinets double glazing.

The thermal conductivity of the glass, door frame and spacers used to separate double and triple glazed panels as well as the emissivity of the glass all impact on the heat transfer through the glass, represented as the U value.

Low emittance (low-E) glass contains a thin, virtually invisible layer on the surface of the glass which reduces the heat transfer by reflecting radiant heat.

Reducing the conductivity of the gas between the layers of glazing will reduce the heat transfer through it. Traditionally dry air (desiccant inside glazing) or nitrogen has been used as the filler gas. Argon (34% lower thermal conductivity than air) is typically used today and costs are 5% more than for air filled units (double-glazing-info.com, 2008). Krypton (65% lower thermal conductivity than air) is even more expensive and therefore rarely used. Double glazing also exists with a vacuum between the panes, and this allows a smaller gap between the panes. In the case of Pilkington Spacia™ glazing, the overall thickness of the double glazed panel is 6.5 mm compared to a typical pane of 24 mm (Pilkington, 2017).

The layers of double glazed panels must be held apart to provide the cavity. Aluminium is commonly used because of its good structural properties; however, it has a high thermal conductivity. Aluminium has a thermal conductivity of 160 W/m.K, compared to other frame materials, e.g. steel (50 W/m.K), Rigid PVC (0.17 W/m.K) and Hardwood (0.18 W/m.K). This causes a lot of heat transfer at the edges of the glazing. Thermally efficient spacers (warm frame/edge) can reduce overall U values by about 25% (U.S. Department of Energy, 1997). Improving the efficiency of the spacers reduces condensation, which reduces the need for door frame heaters. Warm spacers can be flexible, made from thermoplastic or silicone based materials, plastic/metal hybrid or stainless steel.

The heat loss associated with a thermal bridge is called linear thermal transmittance or psi-value. This is the rate of heat flow per degree per unit length of the thermal bridge that is not accounted for in the U-value of the plain elements. Psi values for different spacers and their effect on a windows U value is given in Table 4 (Fenzi, 2017).

Table . Frame with U = 1.2 W/m2.K and Insulating Glass Unit with U1.1 W/m2.K (940 x 1048 mm)

|  |  |  |
| --- | --- | --- |
| Spacer type | Psi [W/mK] | Overall U Value [W/m2.K] |
| Aluminium | 0.085 | 1.368 |
| Stainless Steel 0.15 | 0.050 | 1.270 |
| Extruded PP with Ferritic Steel foil | 0.044 | 1.254 |
| Extruded PC hybrid spacer with austenitic Steel foil | 0.041 | 1.245 |
| Flexible silicone | 0.035 | 1.229 |

The heat conducted through the glazing will be proportional to the U value (combination of conductivity and surface heat transfer coefficient). Schott (2017) provide double glazed doors for refrigerated cabinets with U values of 1.1 (swing door) and 1.6 W/m2.K (sliding door). Remis quote a U value of 1.2 W/m2.K for their hinged doors on refrigerated cabinets (Remis, 2017).

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## Internet shopping

Grocery shopping from home has been offered in various forms for many years, originally based on telephone, fax and postal ordering. Motivations behind early and often small-scale schemes included increasing sales and profits, supporting local communities or producers, providing a social service and reducing car traffic (Cairns, 1996). The introduction of computer-based internet ordering accelerated and expanded many schemes, to the extent that online ordering and home delivery are now commonly offered nationwide by most of the major supermarket chains.

Although home delivery can offer the potential for reducing customer car journeys, the overall environmental impact is not easy to gauge, with various contributory factors such as distances travelled by customers and delivery vans, type of vehicles, delivery failures and returns, type of goods and whether they are selected in store or at a warehouse (Cullinane, 2009). The complexity of most of these factors and their interactivity makes generalisation of likely benefits difficult to predict (for example see Visser, Nemoto and Browne, 2014 and Wiese, Toporowski and Zielke, 2012, which look at complex factors which influence the impact on urban transport). The methodologies used to estimate or model environmental impact also vary and the differences in the results they give interact with differences due to ‘real’ factors such as geographical location and consumer behaviour (see for example Matthews et al., 2002, who studied the energy implications of online book retailing). Although concentrating on the delivery of small, non-food items purchased online, Edwards, McKinnon and Cullinane (2011) assessed such methodological differences. For the same types of products, these authors also assessed the significant impact of failed deliveries, returns and residual consumer trips, e.g. for browsing before online purchase (Edwards, McKinnon and Cullinane, 2010).

Potential environmental benefits were included in a model of a single dual channel retailer, i.e. simultaneous store and online, retailing products such as electronics, books and groceries (Carrillo, Vakharia and Wang, 2014). The effect on the overall environmental impact of operating the online retail channel was found to be more favourable as the proportion of online sales increased. However, differences between the goods studied were evident, with grocery deliveries being more energy intensive (and potentially somewhat less beneficial) due to the need for refrigeration.

Greenhouse gas (GHG) emissions for apples, tomatoes and yogurt sold in the UK, France and Belgium via hyper and supermarkets, corner shops, open air markets, producer’s basket direct sale, farm shops and e-commerce were studied by Rizet et al., 2010. The emissions were split between those resulting from road or sea transport, energy use in buildings (such as cold stores and the retail premises themselves) and from the consumer’s journey home. For the first of these, not surprisingly, the emissions depended on distance and type of transport in combination with quantities transported. This meant that although emissions were high for items such as apples from New Zealand, there were examples where small quantities were distributed within the sale country, and emissions for these could also be high. Emissions from buildings varied considerably by country due to the different energy sources (France being the lowest due to the high proportion of nuclear power and the UK being the highest due to the low proportion of nuclear power). The journey home was found to assume varying importance depending primarily on whether the shopping journey was dedicated to just the product studied or to a combined shop for multiple products. In assessing emissions related to e-commerce, the authors assumed selection of goods at a regional distribution centre (RDC) followed by home delivery or collection at a central collection point, but did not include in-store selection. The emissions for refrigerated yoghurt retailing in the Paris area were given as an example, which showed that e-commerce was more efficient than the traditional ‘physical’ stores:

**Type of retail operation GHG emitted (gCO2e/kg)**

Hypermarket 170

Supermarket 160

Minimarket 160

E-commerce 110

This was because of several factors; the reduced road transport to the RDC, because physical stores require considerable energy input for refrigeration, heating, lighting etc. which e-commerce does not, and significantly that the ‘last mile’ emissions in grouped delivery rather than multiple consumer trips home were lower. It should be noted, however, that many of these figures are highly dependent on the assumptions used in the study, on the type of product and the location of the retailing operation.

Factors which impact on GHG emissions associated specifically with grocery retailing in Finland were described in detail by Siikavirta et al., 2002. The potential to reduce emissions associated with transport was highlighted, as was the reduced energy required to operate warehouses rather than physical retail stores. An additional benefit was also described – the possibility of reducing waste due to avoidance of over-production, better temperature control and use of technologies such as RFID tracking to assist and speed up distribution, sorting etc.

Considering the case of retailing in Austria, Seebauer et al., 2015 came to different conclusions – that promotion of online retailing is unlikely to have significant impact on carbon emissions and that a more fundamental shift in consumer attitudes is required to move towards lower personal consumption of groceries and other goods.

If the potential environmental benefits of e-commerce are to be realised, reducing consumer car travel is essential and the importance of consumer behaviour must also be included in any analysis as e-commerce does not necessarily fully substitute a home delivery for a consumer shopping trip. A Life Cycle Analysis (LCA) model of various shopping fulfilment methods was developed to include such factors (van Loon et al., 2014). This found that impacts varied depending on whether full substitution of consumer trips was assumed or whether some trips were retained for shopping and related purposes. Impacts also varied between types of supply, e.g. from the retailer to the consumer by van or parcel, retailer sold but shipped directly from the producer, online orders fulfilled by selection in RDCs, dedicated warehouses or stores, collection in store or elsewhere, and through traditional ‘bricks and mortar’ stores. In the full substitution scenario the key contributors to emissions were found (where present) to be consumer transport, parcel based delivery networks, the physical store, and transport between the producer and the distribution centre. The carbon footprints of van home deliveries and consumer pick-up methods were found to be the least, with the others up to 4 times worse. When some (up to 90%) of the consumer trips were retained, the relative impact of the different types of supply shifted, with van delivery from local shops becoming worse than consumer collection from local stores and the same as visiting traditional stores.

### Current technology

As mentioned above, online grocery retailing is increasingly based on internet ordering. The goods ordered can be held in a dedicated home delivery warehouse (the so-called ‘dark store’ option), they can be selected at RDCs, or they can be selected in-store from goods on display or held in in-store cold rooms. In the UK, home delivery by multi-compartment vans is common, with the option of collection by the customer at the store or at a specified collection point. Each of these types of operation has different transport and refrigeration needs, and all could benefit from further study.

### Cost

A comparison of the financial performance of UK online grocery retailing operations during their set-up and early operation was presented by Hackney, Grant and Birtwistle, 2006. One of the aspects considered was the choice of warehouse or store fulfilment of orders. Store models were stated to achieve break-even earlier and incur lower losses before break-even, but to be less profitable than warehouse models beyond break-even. This is evidenced by the most successful UK operation in terms of lowest initial losses and time to break-even at the time, which was Tesco’s store-based model.

The impact of internet shopping on the product quality, particularly for perishable foods (meat, fish, fruit & vegetable etc.), should be taken into consideration. However, few studies have been carried out.

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## Inverter Drives

Inverter drives save energy by reducing the work output of a motor driven system to match the demand through speed reduction. It is often claimed that a 20% reduction in speed of a motor equates to a 50% reduction in energy since the power to drive a motor is proportional to speed cubed (Emerson, 2016). Overall benefits in terms of energy consumption will only be seen if the motor/compressor runs more efficiently at low speeds than it does running at high speeds and cycling on and off.

It is intuitive that reducing the speed of a motor will reduce frictional losses and wear and tear on seals and bearings. However, cooling, and, where applicable, lubrication must be maintained. Cooling is often provided by a fan attached to the motor, and if the speed is reduced, the cooling provided may reduce with speed at a faster rate than the heat generation; this will cause the motor to overheat and fail prematurely. Similarly, where a lubrication mechanism is driven by the motor, lubrication may fail with a reduction in speed.

Retrofitting an inverter drive to an unsuitable motor can lead to (ICE-E, 2012);

1) Bearing failures

2) Increased operating temperatures of the core and windings

3) Increased vibrations (supporting structures and mountings should not have resonant frequencies lower than normal operating frequencies)

4) Increased operating temperature due to reduced heat dissipation to surroundings

5) Harmonics generated through electric supply

### Brushless direct current (BLDC) motors

Motor systems are being produced specifically with variable speed and high efficiency in mind; Brushless DC (BLDC) motors are an example of this. They are most commonly found in small air conditioning units. The BLDC uses permanent magnets attached to the rotor and a BLDC inverter drive to control the motor speed at the stator. Although more efficient than an AC asynchronous motor, they are more expensive and the range of systems available is still too limited to cover all refrigeration applications.

### Application of inverter drives in refrigeration systems

A single inverter drive can be used to control several motors provided they are all required to run together at the same speed.

On a refrigeration system, inverter drives can vary the speed of any induction or synchronous motor; typically this would be the condenser and evaporator fans or the refrigeration compressor. However, not all motors or attached loads can be safely (or usefully) speed controlled and severe damage can result from misapplication of the inverter technology.

There is no single guaranteed answer as to how or where an inverter drive will save money on a refrigeration system. Inverters can save money where a system is operating under part-load if:

• It reduces the operating pressure differential across the compressor

• It replaces and improves upon another capacity control means

• It enables better control of mismatched components

This is with the proviso that the expansion valves (and their control system) are able to respond quickly to refrigerant flow rate or pressure changes caused by the compressor speed regulation.

### Refrigeration compressors

Typically the refrigeration compressor is the greatest energy consumer of any refrigeration system and is likely to be the first place to look for savings. However, not all compressors are suitable for speed control (due to lubrication and cooling methods or valve design) and not all systems will benefit from speed control.

It is essential that any refrigeration system is capable of meeting the cooling load demand; there are two components to this:

1) The evaporating temperature must be low enough

2) The mass flow rate of refrigerant must be adequate

The efficiency of a refrigeration system is further dependent on a minimal temperature difference between the evaporator and condenser.

In order to satisfy the cooling demand and achieve the best efficiency, a fine balance must be found. Every component of the refrigeration system (expansion valves, heat exchangers, compressors, fans and pipework) is sized to achieve a specific duty or narrow range of cooling duties at the design conditions. Any deviation from the design condition could reduce the efficiency of any component or in an extreme case cause the component to operate outside of its specification. Compressor speed control through an inverter driver should therefore only be considered for fine tuning of the plant to meet small variations in demand. The heat load (cooling demand) is unlikely to be constant; at best it will be seasonal. Since the surface area of heat exchangers is fixed, the only variables are the operating temperature differences and air flow rates. Where a significant variation in cooling demand is expected, a multi-compressor system with multiple passes through the evaporator (each with its own expansion valve) should be considered.

Oil return is the greatest concern when speed controlling a compressor. Some compressors use oil pumps (or a distribution system) driven from the compressor and any reduction in compressor speed can prevent adequate lubrication and so reduce compressor life. Furthermore, most refrigeration systems circulate some oil around with the refrigerant. This oil is generally returned in the refrigerant flow but this requires adequate velocities within the pipework; modifications to the pipework may therefore be required if a compressor speed is to be varied. An oil separator fitted in the compressor discharge line and an oil pressure switch to shut-off the compressor in the event of oil logging (excess oil in all the refrigerant lines, usually in the evaporator coil) can go a long way toward preventing problems and should be fitted before a compressor is speed controlled. Consult the compressor manufacturer before speed controlling any compressor. There will generally be a defined range of speeds that the compressor is safe to operate at (typically 30 to 60 Hz).

Compressor valve design will also determine the suitability of a compressor for inverter control. Some compressor valve arrangements are only designed to operate over a small frequency range, or range of gas velocities, and so any significant variation in compressor speed can cause significant inefficiencies or damage to the valves and compressor.

Reducing the compressor speed will reduce the volumetric flow rate of refrigerant at the compressor, reducing its capacity to meet the load demand. This will usually have the effect of raising the suction pressure, causing the refrigerant to evaporate at a warmer temperature. Tight control of evaporating temperature reduces losses in two ways; firstly, compressor power consumption will increase by a few percent for every degree below the design operating temperature the system operates at Secondly is that the lower the evaporating temperature, the more moisture will be removed from the store. Thus, maintaining the evaporating temperature as high as possible is an advantage.

Alternatives to inverter drives as a means of capacity control include:

• Unloading cylinders of the compressor - many reciprocating compressors have this facility

• Using a pack of multiple compressors, any number of which can be turned on or off to match the cooling load

• Hot gas bypass and/or evaporator pressure regulation can be used to maintain evaporating temperature and to accommodate a variable heat load with a fixed speed compressor but will be less efficient than inverter or compressor pack control

Figure 8 shows the amount of energy used by a refrigeration system as a % of full load power when capacity is reduced to 50%. Variable speed is the most favourable but does require an appropriately designed compressor.

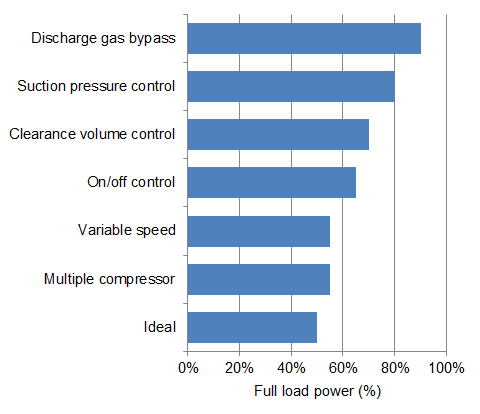


Figure . Comparison of % full load power for different capacity control techniques at 50% load (Qureshi and Tassou, 1996).

### Control

Refrigeration systems generally have multiple separate control systems which, if poorly commissioned, can cause instability, inefficiency and short plant life expectancy.

• The expansion valve may be mechanical or electronic; it will increase or reduce the mass flow rate of refrigerant through it in order to fully utilise the evaporator surface area.

• The cooling is typically controlled at each evaporator by a thermostat (measuring load or cold store air temperature) which shuts off the refrigerant supply to the evaporator using a liquid line solenoid valve when the target temperature is achieved.

• The compressor is usually controlled based on suction pressure. This can be a simple on-off control or a staged control where the compressor has unloading valves or multiple compressors serving the same load.

• The condenser fans will usually be pressure controlled (condensing/head pressure), generally by multiple fans turned on in stages, but the process can be inverter driven.

Since the operating pressures and heat loads on the plant affect all components, the control systems can become unstable as they all change the operating point of other components in an attempt to find their own optimum operation.

### Inverter drive of fans

Both evaporator and condenser AC fan motors can be controlled directly using inverter drives. The most common system uses a single drive to control all condenser fans based on head pressure. Evaporator fans on cabinets are likely to be EC fans and the control of these is dependent on the electronics within the fan itself.

Motor life expectancy is often claimed to be improved through the use of inverter drives but motor manufacturers should be consulted before connecting any motor to an inverter drive as some motor types can be damaged by prolonged operation at a reduced speed due to lack of cooling. Fan blade design may lead to less efficient operation at lower speeds.

### Condenser fan control

The main advantage of an inverter drive over staged control for condenser fans is the better utilisation of the entire surface area of the condenser, since air passes over the entire condenser at low speed rather than just over part of the condenser (as is the case with staged control). This leads to more efficient condensation and improved liquid subcooling. Tighter control is generally achieved with an inverter control compared to staged control. In the event of a fan failure the remaining fans can be speeded up to maintain optimum conditions.

Although energy savings are likely through running all fans more efficiently, savings will not necessarily result. The way to ensure overall energy savings would be to use the inverter drive to reduce the head pressure under normal operation. Savings of a few percent of the compressor power can be expected through a 1K reduction in condensing temperature, so using slightly more power in condenser fans can save energy overall. However, the pressure must be maintained above the minimum operating pressure of the expansion valves or the refrigeration system may not be able to satisfy the cooling demand.

### Evaporator fans

Evaporator fans can also be controlled using an inverter if the motors are suitable. Saving energy on fans inside a cold room or retail cabinet will repay double since any energy put into the fans must also be taken out again by the refrigeration system. Control of evaporator fans can be based on refrigerant superheat at the evaporator exit, air temperature change through the evaporator, or manually adjusted according to the load of the store.

Evaporator fans have two important roles; the first is to circulate adequate air through the evaporator to meet the cooling capacity; the second is to circulate adequate air over the stored product to maintain an even temperature distribution throughout the store and provide a sufficiently high heat transfer coefficient between the stored product and the air.

As the evaporator fan speed is reduced, the “throw” of the fans will be reduced and the velocity of the air over the product will be reduced. Only monitoring of product temperature distribution will show whether air flow rates are adequate.

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## Lighting - cabinets

Studies have shown that lighting accounts for about 15% of the total energy consumed by commercial refrigerators (Southern California Edison Utility Company). In the past, these lights have typically been fluorescent tubes.

Refrigerated display cabinets are fitted with lights to illuminate the food on display. Lighting has both a direct load from the electrical power of the lights as well as an indirect load from the increased compressor power to remove the heat load created by the lights.

The efficiency of a lamp to produce light is called its luminous efficacy (lm/W). It is the ratio of luminous flux (lm), to input power (W). In addition, the quality of colour from the lamp should also be considered. Different technology lamps tend to produce different colours. When comparing fluorescent and LED systems, it has been reported that the lighting within the display case was much more uniform with the LED lighting system compared to the traditional fluorescent system (Rensselaer Polytechnic Institute).

The temperature of the environment where the lamps operate should also be taken into account. The manufacturer’s figures are likely to be given at ambient temperatures which are acceptable for most applications. However, many lamps do not perform the same at the low temperatures in refrigerated cabinets. If using fluorescent lamps, it is important to use lamps which are designed to work at the refrigerated cabinet temperature, or to keep the lamps outside of the cold area. The opposite is true of LED lights, whose efficiency increases with a reduction in temperature. This makes LEDs a natural choice for cold applications.

### Fluorescent lamps

Linear Fluorescent tubes tend to be found in old cabinets within stores which have not had a refit and at the cheaper end of the new cabinet market. These were originally T12s (not sold any more), or, more recently, T8s or T5s (the number after the T refers to their diameter in eighths of an inch and therefore is an easy way to tell them apart). T12 lamps have an efficacy of 78 lm/W, T8 has 92 lm/W and T5 has 103 lm/W (Lighting Solutions, 2009). As T8s are not directly replaceable with T5s without a change of fitting or an adaptor and a change of ballast, the extra efficiency is generally not considered high enough to make the change worthwhile.

Fluorescent lamps are very sensitive to temperature. Peak efficiency is at around 25 to 35°C. However, at -20°C light output can be reduced by 90% and some lamps will not even start. The light output of a T8 linear fluorescent lamp could drop by more than 25% when the ambient temperature reduces from room temperature (25ºC) to refrigeration temperature (7ºC).

Fluorescent lamps require a ballast to provide the correct starting voltage and control the running current. Older T12 fluorescent lamps use magnetic ballasts, whereas newer T8 and T5 lamps use electronic ballasts. Electronic ballasts run at higher frequency, reducing flicker and increasing light output and efficacy. There are two methods of lamp starting, rapid and instant start. Rapid start systems use cathode heating to increase the life by allowing more on-off cycles. The Instant start systems are more energy efficient as they do not heat the cathodes.

The fluorescent lamp and ballast need to be considered together. It is possible to increase the lumens of a lamp by using a different ballast factor (BF). For example a lamp rated at 3000 lm with a ballast with a BF of 0.7 will only produce 70% (2100 lm), with a corresponding reduction in energy usage. A higher BF of 1.2 will give an output of 120% (3600 lm). Fluorescent lamps can be dimmed as low as 1% of their output by using dimming ballasts.

Linear fluorescent tubes spread their light in all directions and require deflectors to aim the light where required. Only about 60% of the light is directed towards the displayed products and the rest is wasted (Raghavan and Narendran, 2002).

### LED lamps

LED lights are fitted to the more expensive new cabinets and are being retrofitted to older cabinets during supermarket re-fits. LED lights operate with a direct current (DC) at about 12 V; therefore require a transformer (driver) to convert the mains (230 V AC) voltage to 12 or 24 V DC. Some lamps have this built in and some require a transformer. Enhanced safety is a prime feature of LED lighting due to the low voltage. Another advantage of LEDs is that they allow the use of motion sensors because they can turn on and off rapidly with no damage, unlike fluorescents.

The US DOE (US Department of Energy, 2014) established the efficacy of warm white colour mixed (CM)-LEDs to be 133 lm/W in 2013. They have a target of 191 lm/W in 2020. They also presented the efficiency of the driver as 85% in 2013, with a 2020 target of 93%.

Philips has a number of case studies where their lights have been installed in supermarket cabinets. They are detailed below.

* Sainsbury’s – Philips LED lighting system was used to light the interior of its freezer cabinets across its stores. In all, 15,000 pieces of Philips Affinium LED freezer lighting modules with Philips Luxeon LEDs were installed, resulting in energy savings of 75% per freezer. Moreover, the improvement in the lighting effect was in excess of 150%. Philips states an LED life of 10 years, compared with that of fluorescent at 6 to 12 months (Neate, 2008)
* Tesco – Philips has deployed Affinium LED freezer modules, replacing the interior fluorescent lighting of Tesco supermarkets’ freezer cabinet estate in more than 750 stores. This reportedly reduced their freezer lighting energy by around 60% whilst at the same time ensuring excellent light quality. They state an LED life of 6 years (however, the same lights have been described as having a life of up to 10 years), fluorescents 6 to 12 months (RAC, 2010).
* Edeka – Philips installed LED lights in their freezer cabinets. They state a 70% reduction in energy and a 2.7 year payback.
* A project was conducted in a participating supermarket in northern California to demonstrate the performance of LED lighting for refrigerated display cases under real world conditions (Diebel et al., 2013). Savings of 46.3% were reported when changing from fluorescent to LED lights (with greater savings if motion sensors were used).
* Heidinger et al. (2014) developed experimental tests in a glass door multideck display case with different types of lighting setup: fluorescent lamps and LEDs, both tested in horizontal (under the shelves) positioning and LEDs in vertical positioning (next to the doors). Experimental results show that LED illumination under the shelves reduces the direct energy consumption by 41% and LEDs in vertical position result in 74% of energy economy. The LED illumination in vertical positioning is the best alternative to save energy and highlight the products exhibited.

### Lamp lifetime

Lamp lifetime due to failure of the lamp is another important consideration. Their life can be seriously reduced by constant turning on and off. Linear fluorescent lamps are quoted to have a useful life of 20,000 to 30,000 hours and LEDs 35,000 to 50,000 hours, according to General Electric Company (2010). They quote LED efficacies between 59-81 lm/W at an operating temperature down to -20ºC and a 78% reduction in power compared to a T8.

### Power factor

The power factor (PF) describes the efficiency with which an electrical appliance uses the current it draws in an alternating current (AC) circuit. Fluorescent lamps have a PF less than 1, perhaps down to 0.9. LED lamps may have PFs as low as 0.5. The consequence of lower PFs is higher current for the same power. This causes greater losses in the transmission lines, which is why the electricity suppliers often charge a surplus when PFs fall below a certain level.

### Costs

The cost of LEDs is higher than fluorescents. According to refrigeration contractors, to upgrade from fluorescents to LEDs costs £48.43 per metre of cabinet length and has a payback 3 year ROI.

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## Lighting (store), impact on cabinet performance

Store lighting has both a direct (energy consumed by the lights) and indirect effect on energy consumption of the retail store. The direct energy consumption of the lights is usually the second largest in a supermarket after the refrigeration system. Skudritis (2015) showed the lighting to be 29% of the total store energy consumption for a supermarket in Latvia.

The indirect effect is caused by the radiant heat from the lighting entering the refrigerated cabinets. Faramarzi (1999) states that radiation accounts for 12% of the heat load of an open fronted chilled cabinet. This includes infra-red radiation from the store surroundings as well as light radiation from the store lights and the cabinet lights. For closed cabinets the heat load from external lighting is likely to be reduced due to the reflective properties of the glass. Faramarzi et al. showed a radiation heat gain of only 1.3% for a glass door cabinet (Faramarzi 2002).

This heat load will have to be removed by the refrigeration system, causing an increase in the energy use of the refrigeration system. Also, in summer the heat load of the lights may need to be removed by the store’s air conditioning system. However, in winter the heat load from the lights may help to heat the store.

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## Liquid pressure amplification (LPA)

Compressors in refrigeration plants tend to operate at a higher delivery pressure than is needed. With liquid pressure amplification (LPA), compressors can run at a lower pressure and save energy.

As stated in the section ‘Reducing head pressure’, reducing head pressure will allow the refrigeration system to run more efficiently (higher COP). However, reducing the head pressure too far can cause too small a pressure difference across the expansion valve, causing it not to work effectively.

LPA consists of a pump at the outlet of the condenser after the liquid receiver. The pump provides an elevated stable pressure before the expansion device, allowing the compressor to float as low as ambient conditions will allow. The pump needs to be able to operate within a refrigeration circuit without leaking. The energy consumed by the pump and that caused by the extra condenser fan flow rate (to reduce head pressure) is far lower than that saved at the compressor.

LPA is best applied in situations where head pressures are allowed to fluctuate. As the pressure at the outlet of the pump is higher than at the exit of the compressor, this allows a percentage of the liquid after the pump to be injected into the superheated gas after the compressor. This allows the superheated gas entering the condenser to be brought to saturation, accelerating condensing and increasing subcooling.

The use of liquid injection in combination with LPA will further increase the energy savings possible due to the desuperheating of the discharge refrigerant gas, which will increase the condenser capacity and thus reduce the difference between the ambient and condensing temperature.

Hadawey et al. (2010) have shown that it is possible to achieve more than 10% in energy savings over and above those that can be attained with floating head pressure by adopting the use of LPA in conjunction with liquid injection. The level of energy savings that can be achieved with LPA, however, is system specific and each application will require careful consideration of the savings against the capital cost of the technology.

Carbon Trust (1996) states that in general the larger the plant, the better the return on investment on LPA. On a 300 kW plant, energy savings of up to 25% have been seen. The capital costs would have been around £25,000, giving a payback of under one year.

In trials of the HY-SAVE® Liquid Pressure Amplification (LPA) system at a Tesco store in Ireland (Hy-Save, 2009), independent energy consultants saw the amount of energy consumed by the refrigeration plant reduce by 24%. Previously at the store, condensing temperatures had to be kept unnecessarily high at around 32°C to combat system flash gas and to keep the stores evaporators operating efficiently. Since LPA® pumps were installed in the system’s liquid line, condensing temperatures have been reduced to 20°C (68°F) whilst delivering vapour-free refrigerant to the store’s display cabinets and cold rooms, maintaining peak efficiency. Flash gas at the evaporators may be a consequence of undersized and poorly insulated liquid lines and can also be overcome with liquid subcooling.

MacWhirter (1999) installed an LPA system on an air cooled water chiller with liquid injection and showed the average power absorbed by the system reduced from 26.16 to 20.91 kW (a reduction of 20.1%).

Information provided by a refrigeration contractor who trialled several types of liquid pumps has not shown these delivered energy savings. The type of pump is important due to cavitation which can limit the operating conditions (head pressure), reducing the potential savings.

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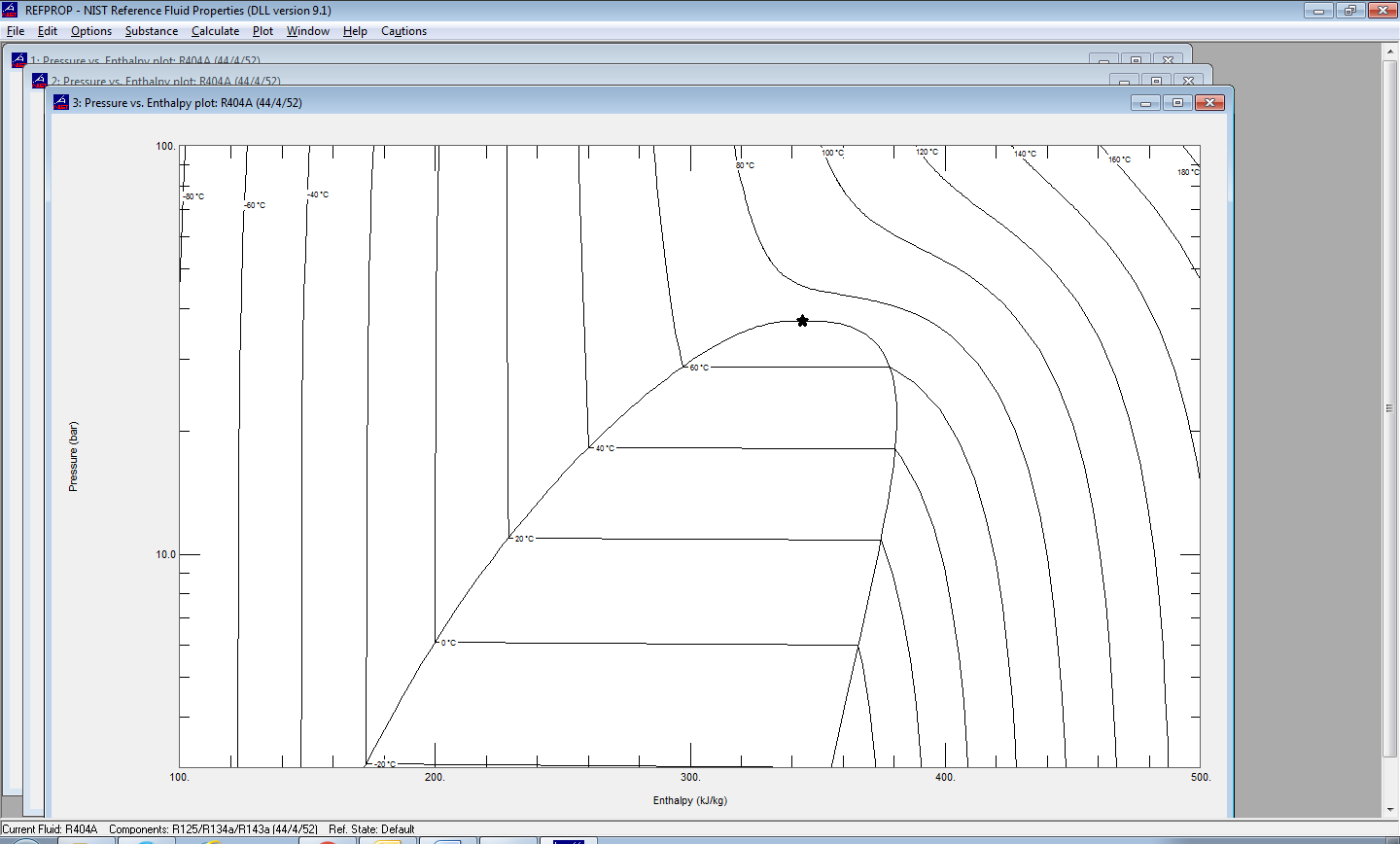
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## Liquid-suction heat exchangers

Suction-liquid heat exchangers (LSHEs) exchange heat between the suction line at the exit of the evaporator and liquid line at the exit of the condenser. They increase refrigeration capacity by subcooling the liquid refrigerant. In addition, they help to ensure that all refrigerant is evaporated before returning to the compressor and prevent flash gas formation at the inlet to the expansion valve. Since most supermarket systems have a substantial amount of non-productive heat gain between the load and the compressor (due to poor insulation and long pipe runs), the LSHE essentially recovers capacity that would otherwise be lost.

The diagram below shows a p-h diagram for R404A refrigerant showing a normal cycle in green. The subcooling of the liquid is shown by the red line extending to the left and is equal in enthalpy difference to the superheating of the vapour represented by the red line extending to the right.



Sub-cooling

Super-heating

**P-h diagram for R404A refrigerant showing a normal cycle in green and the extra red parts show the liquid suction heat exchange.**

The beneficial effects of a LSHE are offset by the increase in temperature and reduction in pressure (caused by pressure drop though the heat exchanger) of the suction gas causing a decrease in the refrigerant density and compressor volumetric efficiency. Therefore their installation is not an energy efficient option for all refrigerants. LSHEs will increase the temperature to the compressor, which reduces motor cooling and increases discharge temperatures.

Klein, Reindl, and Brownell (2000) state that LSHEs with a minimal pressure loss on the low pressure side are useful for systems using R507A, R134a, R12, R404A, R290, R600a, and R410A. However, they are detrimental to system performance in systems using R22, R32, and R717. For R404A, they showed a capacity increase of 50% for a LSHE effectiveness of 0.9 (assuming no pressure drop). However, when the reduced volumetric efficiency of the compressor was taken into account, this dropped to less than 20%.

Mastrullo et al. (2007) state that a LSHE is always beneficial for refrigerants R290, R134a, R413A, R507A, RC318, R417, R236a, R227a, R125, R502, R600 and R600a. It is never beneficial for R717, R32 and R407C. For refrigerants R22, R152a, R410a and R1270, it depends on the evaporating temperature and condensing pressure; the higher the condensing and lower the evaporating temperature, the more beneficial. This assumes maximum effectiveness of the LSHE. They showed that for R134a the COP could increase by up to 20% (condensing temp of 50ºC and evaporating temp of -40ºC). At an evaporating temperature of -10ºC and condensing temperature of 35ºC, the COP increase will be nearer 5%.

Sarhadian et al. (2004) carried out an experimental evaluation of a number of viable and near-term energy-efficiency solutions for an open vertical refrigerated display case. They used a micro-channel LSHE which resulted in roughly an 18ºC increase in total system subcooling which made a partial contribution to the 18% improvement in the refrigeration effect (the majority was due to a high-efficiency evaporator). The measured heat exchanger effectiveness, defined as the ratio of actual to maximum possible heat transfer, was about 70%. Nuriyadia et al. (2015) showed a maximum benefit in capacity/COP of 20% for a refrigerator at -20ºC with a HE effectiveness of about 0.5. However, for a refrigerator at 0ºC there was a 5% reduction in capacity/COP.

Pacific Gas and Electric Company (2011) carried out a feasibility and cost-effectiveness study of high-performance LSHEs on display cases and walk-ins. The results show that the benefit/cost ratio for all system configurations and in all climate zones (in California) for all LSHE types is greater than 1 with the exception of certain MT walk-in LSHEs in small stores. For sizes less than approximately 150 square feet, LSHEs were generally not cost-effective for MT walk-ins. Stakeholders noted that with certain refrigerants (notably R407A), compressors were sensitive to return gas temperatures and that an increase in this as a result of adding LSHEs could potentially cause excessively high compressor temperatures and premature compressor failure for low-temperature suction systems.

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## Loading (food) temperature and duration

Refrigerated display cabinets are designed to keep food cold; they are not designed to remove large amounts of heat from overly warm product. Therefore products should be loaded at the cabinet display temperature.

This may not happen if;

1. a product arrives at the cold store too warm and has not had time to cool down in the cold store
2. the cold store from which products are loaded into the cabinets is warmer than the cabinet temperature
3. product is left too long in ambient conditions waiting to be loaded

1) should not happen if the correct temperature checking of refrigerated products is carried out at delivery. 2) should not happen if the cold stores are set at the correct temperature, are not malfunctioning and are used effectively with a good door management strategy. 3) should not happen if the correct procedures are in place to guarantee that product is not left out and uninsulated for longer than recommended.

The most important consequence of loading warm product is that the product will be above safe storage temperatures until it has cooled down, reducing shelf life. Another important consequence is that if the heat load from the product is higher than the cabinet can extract, then the cabinet will warm, potentially raising other products above their storage temperature. A product left out on a humid day can also have condensation on it, which is detrimental for the look of the product and also may have an impact on the cabinet defrost.

Even if the cabinet is capable of removing all the heat, a refrigerated cabinet is not the most efficient system to do this. Refrigerated cabinets run at lower COPs than the cold stores and are therefore an inefficient way of cooling the product. Effective policies and procedures should limit the temperature of product loaded into cabinets. These policies and procedures should include training of staff.

The time taken to load food into cabinets affects the amount of air which infiltrates the cabinet. This is more significant with closed cabinets but can also have an effect on open fronted cabinets, where the air curtain is disturbed as loading occurs. This will increase energy consumption as the refrigeration system removes the extra heat and will also require extra defrost heat to remove the moisture. Staff training can be used to highlight the importance of keeping doors open for as short a time as possible during loading.

ASHRAE (2006) shows maximum product temperatures rising by 6.3°C, 2.6°C, 1.2°C and 0.5°C due to blocked return air, overloading, cavities and air curtain interference respectively.

Limited information exists on the impact of loading food at the incorrect temperature into retail display cabinets. According to data from IRSTEA (2003), pizzas were found to spend 1.25 hours between being removed from the cold store and loading in the cabinet. The temperatures of the pizzas were as high as 10.5ºC when loaded into the cabinet. The recommended preservation temperature was 4°C. Some of this may be due to the cold store being too warm and some due to the pizzas gaining heat from the ambient environment between cold store and refrigerated cabinets. This means that the refrigerated cabinets have to remove this heat.

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## Magnetic refrigeration

Technologies such as magnetic cooling have potential advantages, such as no refrigerants which may have global warming, flammability or other safety considerations, and potentially higher efficiencies than those of vapour compression technologies. Magnetic refrigeration takes advantage of the magnetocaloric effect; the ability of some metals to change temperature when exposed to a magnetic field. In a magnetic cooling system such materials are known as magnetic refrigerants. Much of the original work and most prototypes developed used gadolinium metal, an expensive rare-earth element, as a magnetic refrigerant. More recent work has looked for new materials that are cheap, have suitable transition temperatures and exhibit a large magnetocaloric effect. Magnetic refrigeration has the prospect of efficient, environmentally friendly and compact cooling for a wide field of applications.

According to Lewis et al. (2007) successful commercialisation will require:

* ‘Magnetic refrigerants’ with a larger magnetocaloric effect to be produced in large quantities.
* Stronger, smaller and cheaper permanent magnets.
* Improvements of the active magnetic regeneration cycle and new cycles.
* Improvements to the engineering design of the systems.

Gschneidner and Pecharsky (2008) ‘boldly’ predicted that production of near room temperature, magnetic refrigeration systems will grow to 1000 units by 2015, by which time they would consider the technology to be commercialised.

Tura and Rowe (2011) described a prototype magnetic refrigerator using gadolinium as the refrigerant. With a total of 110 g of refrigerant, the device produced a maximum temperature span under no thermal load of 29°C, and 10°C under 50 W. The overall COP determined by using the power to the drive motor was between 0.3 and 0.8 under most operating conditions and temperature spans. The maximum COP measured was 1.6 with a span of 2.5 °C. They stated that if the inefficiency of the motor was removed, a maximum COP of 2.2 was achievable, and when the magnet drive losses were excluded, the maximum COP becomes 10.

In 2014, GE Corporate Research demonstrated a concept magnetic cooling engine for individual bottle cooling (Anthony, 2014).

Information from Aprea et al. (2015) stated that COPs comparable to an R134a refrigeration system could be achieved by certain magnetic alloys.

In 2015 Haier, Astronautics Corporation of America and BASF developed a proof of concept wine cooler refrigerated by a magnetocaloric heat pump (BASF, 2015).

In September 2016, Camfridge of the UK and Whirlpool Corporation tested a prototype magnetic cooler the same size as the gas compressor inside a production domestic fridge cabinet (Wilson, 2016).

In October 2016, Cooltech Applications exhibited its magnetic refrigeration system (the MRS400) integrated into a display case, which was previously demonstrated at Carrefour’s head office, at the European trade show Chillventa (Cooltech, 2016).

Some of the recent developments and knowledge that have been highlighted by Wilson (2016);

* Vacuumschmelze are able to produce bulk refrigerant materials with good magnetocaloric properties.
* A compact solution is achieved by maximizing power density.
* Development of a recycled permanent-magnet supply chain will enhance the positive environmental impact of magnetic refrigeration and could also help to reduce costs.

Most of the studies focus on systems with limited cycle frequency in which a fluid transfers heat to and from the magnetocaloric material. A suggested solution for increasing the frequency is use of solid-state magnetic refrigeration in which thermal diodes guide the heat from the cold end to the warm end. Monfared (2017) investigated solid-state magnetocaloric refrigeration systems with Peltier elements as thermal diodes. He concluded that further investigations are required to ascertain if passive thermal diodes can conduct the heat in the desired direction fast enough, considering the practical limitations in manufacturing and material properties.

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## Motor Efficiency Controllers (MECs)

Unlike inverter drives, motor efficiency controllers do not influence the operating speed of the motor; they simply aim to improve the efficiency of the motor by reducing the losses in the motor windings. Because an MEC does not affect the running speed of the motor it will not provide any means of capacity control. However, it can provide energy savings without altering the fine balance of a well optimised and well sized refrigeration system and can be used on systems where variable speed units cannot be employed.

MECs typically cost around a quarter of the price of an inverter but a separate drive must be fitted to each motor.

MECs reduce the power supplied to the motor by monitoring the current and voltage waveforms for slip as the voltage waveform is trimmed by the MEC. The more the voltage can be trimmed, the greater the savings; hence savings are usually only appreciable on motors running at 75% load or less for a significant amount of the time. Peak to peak voltage and frequency are maintained (Figure 9) to ensure that peak torque and operational speed are unaffected by the MEC.

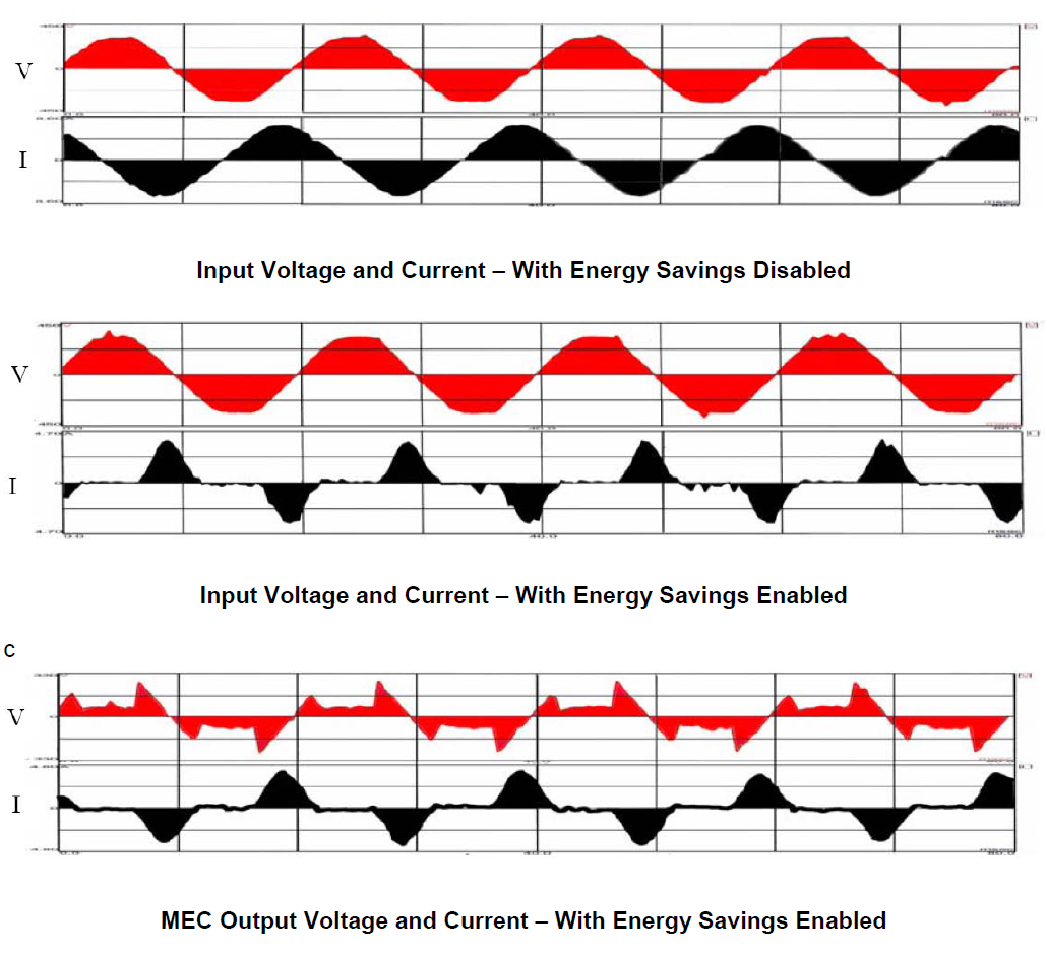


Figure . MEC voltage and current waveforms (Envirostart). Top shows wave forms for direct connection to mains, middle shows the MEC input voltage and current waveforms and the bottom plot shows the wave form provided to the motor.

The energy used and paid for is equal to the integral of the area under the input voltage waveform multiplied by the integral of the area under the input current waveform multiplied by the power factor.

Savings from MECs are limited by the magnitude of the losses within the motor being controlled and are generally greatest in motors which are part-loaded. Case studies by Envirostart Ltd (Birmingham, England), a manufacturer of MECs, have found that savings of 40% are likely on single-phase refrigeration compressors and savings of 15% are likely on three phase compressors.

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## Nanoparticles in refrigerant

Nanofluids are engineered colloidal suspensions of nanoparticles (1-100 nm) in a base fluid. The size of the nanoparticles imparts some unique characteristics to these fluids, including greatly enhanced energy, momentum and mass transfer, as well as reduced tendency for sedimentation and erosion of the containing surfaces. To enhance heat transfer, nanofluids were developed, mainly based on copper and aluminium nanoparticles of above 20 nm size (Eastman et al., 1996). Theoretically, these nanoparticles have a high thermal conductivity and hence should improve the heat transfer near the laminar sublayer (Jana et al., 2007; Lee et al., 2007; Ko et al., 2007). Recent experimental work at NIST (Bello, 2008) with varying concentrations of nanoparticle additives indicates a major opportunity to improve the energy efficiency of large industrial, commercial cooling systems. NIST have shown that dispersing low concentrations of copper oxide particles (30 nm in diameter) in a common polyester lubricant and combining it with R134a improved heat transfer by between 50 and 275%. Success in optimising mixtures of refrigerants, lubricants and nanoparticle additives could be beneficial. High-performance mixtures could be swapped into existing chillers, resulting in immediate energy savings. Due to improved energy efficiency, next-generation equipment would be smaller, requiring fewer raw materials in their manufacture.

Energy savings of between 9.6 and 26% have been reported for a domestic refrigerator (Bi et al., 2008, Bi et al., 2011). The 26% reduction in energy was for R134a/mineral oil with nano-particles compared to R134a/POE oil. The ability to use mineral instead of POE oil was the main reason for the energy savings. The 9.6% in savings were with R600a refrigerant.

Jaiswal (2015) evaluated nanoparticles suspended in R404A. They calculated COP enhancement between 3 and 15% depending on the nanoparticles and their concentration. Higher concentrations gave the highest COPs and copper oxide was the best nanoparticle.

Currently there has been little work on the safety of nanoparticles in refrigeration systems. Although nanoparticle ‘kits’ are sold, there are serious concerns over their safety and their impact on the compressor and expansion valve.

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## Night blinds and covers

Night blinds are an established technology for open multideck cabinets. They are used to provide a temporary physical barrier between the cold air inside the cabinet and the warm ambient air outside whilst access to the cabinet is not required. As such they are only useful when stores are closed. Energy savings are a direct result of the reduced entrainment of warm air by the air curtain and through the reduction of radiative heat gain to product where reflective blinds are used.

Axell and Fahlen (2000) found energy savings of between 25 and 40% due to night curtains. For freezer cabinets Hawkins et al. (1973) reported energy savings of 27 and 29% when night blinds were in use on two freezer cabinets. This equated to overall energy savings of 20% when blinds were used during non-trading hours. This figure seems to suggest that trading hours were very short.

Blinds should be well installed; a poorly fitted blind can result in elevated product temperatures, particularly if food is aligned with the edges of the blind. If the gap at the edges of the blind is sufficiently large, air is lost from the cabinet at the lower part of the gap and air infiltrates in the upper part of the gap. This results in ambient air being drawn into the cabinet in close proximity to the food and a resulting increase in the food temperature in this area.

For well type cabinets, covers can be used to reduce energy consumption and stored product temperature. Unlike the blinds on vertical cabinets, the covers for well cases are typically solid but the covers can be transparent (glass or bubble lid) for use during trading hours. Where the covers prevent access to or visibility of stored product, the level of staff interaction to fit and remove covers is often considered prohibitive to their use.

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## Peltier cooling

Peltier or thermoelectric (TE) devices are lightweight, small, and inexpensive and do not utilise refrigerants. TE devices are limited by their low efficiencies (approximately one third those of a vapour compression system) but do have some advantages in terms of direct emissions, reliability, quiet operation and also may be useful for spot cooling.

There is no published information on the use of Peltier coolers in real life retail display cabinets. It would appear likely that the most suitable application would be to spot cool an area of high temperature within a cabinet. However, good design and optimisation could potentially overcome such issues without the use of a more complex additional technology such as Peltier cooling.

Information presented by Min and Rowe (2000) indicated that the COP of current Peltier coolers is less than 0.5 when operating over a 20 K temperature span. With optimisation, COPs of between 1 and 1.2 were thought possible by the authors.

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## Pipe insulation

Tubing should be insulated on the low pressure side between the evaporator and compressor to minimise condensation (on chiller circuits) or ice build (freezer circuits). There is likely to be some potential benefit in energy consumption due to good insulation as colder, denser gas will enter the suction of the compressor. However, this is likely to be small.

Liquid lines on economised systems should also be insulated as they may form condensation and the benefits of the economiser could be somewhat reduced if un-insulated. Liquid lines which run through hot ceiling voids could also gain heat and lead to flashing so should also be insulated.

It is standard practise to insulate pipework and so there is minimal opportunity for energy savings unless the refrigeration plant insulation has been damaged or compromised.

## Pipe pressure drops minimisation

Reducing the pressure drops between the compressor and condenser and compressor and evaporator will reduce the pressure ratio of the compressor, reducing its energy consumption. A saturated temperature difference in the suction line or liquid line of about 1 K (between compressor and heat exchanger) would account for an energy penalty of about 3% for a chiller system.

However, it is not practical to significantly minimise suction and discharge line pressure drops, since sufficient oil return velocity of between 6-10 m/s should be maintained for all operating conditions and this inevitably imposes a significant pressure drop, especially in systems with long suction line runs such as pack supermarket systems. The competing requirements of maintaining oil return velocity and reducing pressure drop will always lead to a significant pressure drop.

Pressure drops can be reduced by reducing pipe runs, and this gives an obvious benefit to integral systems over remote systems. However, it is essential to consider the energy efficiency of the integral system being used in comparison with a pack system. A simple integral system without capacity control for example may be intrinsically less efficient than a pack system with capacity control.

## Recommissioning

The aim of commissioning new buildings is to ensure that they deliver, if not exceed, the performance and energy savings promised by their design. When applied to existing buildings, commissioning identifies the almost inevitable “drift” from where things should be and puts the building back on course (Mills, 2009).

It has often been found that significant amounts of energy are wasted through poor commissioning (Evans et al., 2013) and it is widely known in the industry that recommissioning (commissioning again) can help (Gaved, 2013). A recommissioning effort will detect and correct any major systemic problems that develop over time.

There is a crossover between service and maintenance and recommissioning. A very thorough servicing programme may incorporate many aspects which are considered under recommissioning. Aspects of recommissioning which are not likely to be included in servicing are optimizing control logic and establishing the most appropriate equipment set-points. It is possible that these have been modified inappropriately at some point or it may be that things have changed and new values are considered more appropriate.

Recommissioning maybe done at different frequencies, either periodically to ensure systems are operating at their designed set-points or after performing significant maintenance, replacement, or upgrades to a store which fundamentally change how a store will perform. The U.S. Department of Energy (2012) recommends recommissioning every three to five years. Recommissioning may be triggered by other things, e.g. energy consumption or higher-than-normal maintenance costs. Monitoring of plants and cabinets provides an early warning of equipment losing efficiency and it may be more prudent to recommission when monitoring indicates there are problems.

A number of refrigeration recommissioning measures are given below:

* Check refrigerated cases set-points. It is possible they have been set lower than needed.
* Clean and calibrate humidity sensors that control anti-sweat heaters.
* Repair or replace gaskets and seals on refrigerated cases (should be covered under maintenance).
* Verify correct charge in refrigeration systems, and repair any refrigerant leaks. Leaks should be covered under F-gas maintenance. However, it is possible that systems have been improperly charged.
* Verify optimal head and suction pressures.
* Verify or establish an effective maintenance protocol,
* Ensure that airflows in refrigerated cases are not blocked by improperly stocked shelves. If they are, adequate measures, e.g. training, monitoring or a physical mechanism to stop this happening, should be put in place.
* Check temperature probe locations. For refrigerant temperatures, they should be in proper contact with the refrigerant pipe and for air probes they need to be in a sensible position for measuring the air temperature.

Training of service contractors is a vital part of ensuring that the plant continues to operate at the best possible efficiency, much of the blame for poor performance being laid at the door of under-skilled, time-poor service staff.

Retro- and recommissioning yield average whole-building energy savings of 16% (not specific to retailing) and a simple payback of 1.1 years, according to Mills (2009).

Some companies offer a recommissioning service paid for as a commission on energy saved, and forecast energy savings of 5 to 10% as a result of recommissioning (Xcel Energy, 2010). Greening Retail (2014) reports average energy savings of 31% over a series of 4 case studies where supermarket installations were recommissioned; details of the individual case studies or significant adjustments made were not provided in the available text.

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## Reducing head pressure

Supermarkets maintain head pressure at a minimum level either to ensure consistent operation of expansion valves, or to enable refrigerant gas to be used to defrost evaporators.

Reducing head pressure will allow the refrigeration system to run more efficiently (higher COP). However, reducing the head pressure too far can cause too small a pressure difference across the expansion valve, causing it not to work effectively. There also needs to be enough pressure difference (between high and low pressure) to cause refrigerant to pass through the heat exchangers at a flow rate which allows adequate heat exchange. However, there is considerable potential to save energy by lowering the minimum head pressure to the lowest level which still works effectively.

Reducing the head pressure of a refrigeration system will save between 2% and 4% for every 1°C reduction in condensing temperature (Carbon Trust, 2011).

Reducing condensing pressure too low will have an effect on cabinet expansion valves. Thermostatic expansion valves (TEVs) operate less well at low pressure differences: they may need to be replaced with Electronic Expansion Valves (EEVs), which will work with lower pressures. However, they are still subject to minimum acceptable pressure differences. Alternatively, liquid pressure amplification (LPA) could be considered to raise liquid line pressures. Additionally, lowering the minimum head pressure may also be incompatible with hot gas defrosting systems as sufficient pressure will not be maintained at all times in order to initiate the defrost.

Modelling results of a supermarket multi-compressor refrigeration system for a period of 24 hours in the summer have shown that energy savings of 22.5% can be achieved by reducing the minimum condensing pressure from 15.1 to 12.0 bar. The reduction that can be achieved is dependent on the size of the condenser compared with its load. Oversized condensers offer greater potential savings. These savings will increase during winter operation where ambient temperatures are lower (Ge and Tassou, 2000). Laboratory testing of a supermarket refrigeration system (Toscano, Walker and Tetreault, 1983) showed the major factor contributing to savings was the use of floating head control. Analysis of the data showed that use of this control, rather than ambient subcooling, produced a 14.6% reduction in compressor energy consumption. The test conditions included winter and spring ambient temperatures ranging from -13 to 21°C.

Floating head pressure occurs automatically by setting the intended head pressure lower. The condenser will then run at as low a condensing pressure as possible, limited only by the minimum setting. The electricity used by condenser fans can be significant and rises fast with fan duty. Electronically commutated motors have a substantial benefit in terms of power use.

A case study at Albertsons grocery stores (US) (Rocky Mountain Power, 2007) showed annual savings of 47,430 kWh per year against an installation cost of $9,878 by using floating head pressure control.

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## Reducing thermal radiation

### Introduction to thermal radiation

Thermal radiation is one of the mechanisms by which the refrigerated cabinets gain heat, along with convection, conduction, infiltration etc. According to Sarhadian et al. (2004) radiation provides 12% of the heat load of an open multideck display case.

The effect of radiation drops when glass doors are fitted, as these provide a barrier to infrared radiation. Faramrarzi et al. (2002) showed 7.5% and 1.3% of the cooling load resulted from radiation for an open and closed door chilled cabinet respectively, with the total heat load lower for the glass door cabinet at only 32% of that for the open cabinet.

The effect of radiation is higher for frozen cabinets as the radiation is affected by temperature difference between the cabinet and the surroundings more strongly than other heat loads. Faramarzi et al. (2000) showed a radiation heat load of 43% for a frozen food coffin cabinet compared to 8 to 10% for open vertical chilled cabinets.

How much of the incident radiation is absorbed or reflected by the cabinet or products within the cabinet is dependent on the emissivity. Low emissivity surfaces, e.g. opaque metal-like surfaces, absorb little thermal infrared radiation and reflect the remainder. The opposite is the case of infrared absorbing surfaces, such as a matt black paint. Unpainted aluminium has an emissivity of approximately 0.1, which means 10% of the radiation is absorbed, and 90% reflected. A matt back surface may have an emissivity of 0.90 which means that 90% of the radiation is absorbed and only 10% reflected.

### Low emissivity packaging

Low emissivity packaging can be an effective means to reduce radiant heat gain to food in exposed positions within a display cabinet. If the food temperature can be reduced, then food safety or longer shelf life can be maintained. Low emissivity packaging is not applied proactively by food manufacturers, although some foods do have reflective packaging.

Davies et al. (2012) examined the potential to use new printing techniques to produce low emissivity packaging. In the work they measured packaging emissivities from 0.79 (waxed paper) to 0.01 (Aluminium foil/plastic laminate). They showed food top surface temperature was reduced by 10.6, 9.9, 9.4 and 6.0 K for packaging emissivity of 0.01, 0.07, 0.28 and 0.44 compared to the standard packaging emissivity of 0.79.

This would enable refrigeration system efficiency improvements that could reduce energy consumption and carbon by 30%. Overall carbon savings were predicted based on energy and the use of packaging with low embodied carbon.

Where warmer evaporating temperatures are enabled by the reduced heat gain to vulnerable products, some further energy savings may result; Granryd (2000) claims that for a -18°C product temperature, an air temperature of -31°C is required. Reducing the emissivity would enable warmer discharge air (and consequently evaporating) temperatures to be used, which would raise the system COP and save more energy.

### Radiant reflectors

Aluminium foil or infrared mirrors impart low surface emissivity and are being used in some open display cabinets for both aesthetic (presentation of product) and heat transfer reasons.

Application is not solely limited to chilled display cases. The greatest impact may be made in the freezer cabinet. Radiation becomes more significant for open freezer cabinets but these are becoming less common in supermarkets (with only open well cabinets currently being used) as full glass door cabinets become increasingly popular.

Hawkins et al. (1973) described radiant heat reflectors for use with open (well type) freezer cases. Radiant heat reflectors would be less practical and/or effective with a vertical cabinet due to the orientation of the exposed cold surfaces. A vertical cabinet faces another cold vertical cabinet and therefore the radiation heat load is less than that of a horizontal cabinet which faces a warm ceiling. There is also space above the horizontal cabinet to fit reflectors, whereas there is little room in an aisle. Hawkins reported 2K reductions in warmest pack temperatures with simple reflectors and up to 5K with corner cube type reflectors. Potential energy savings were not presented but would be estimated to be approximately 5-8% based on raising the evaporating temperature by between 2 and 5 K. The final design presented comprised tiles of corner cube reflectors which served not only to minimise the radiation through the use of low emissivity materials but also to avoid reflecting warm surfaces against cold surfaces. This technology was used in field trials by T. Wall and Sons (Ice Cream) Ltd.; stored product temperature was reduced but energy savings were not reported.

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## Refrigerants - Introduction

As the international scientific consensus calls for limiting the global temperature increase to 2ºC (The Cancun Agreements, United Nations Framework Convention on Climate Change) to prevent undesirable climate effects, the European Council has called for a reduction of greenhouse gas emissions in the EU by 80-95 % by 2050 compared to levels in 1990. In order to achieve this objective at the lowest cost, all sectors and greenhouse gases must contribute, including fluorinated greenhouse gases (F- gases) whose global warming potential (GWP) can be up to 23,000 times more than that of carbon dioxide.

According to the cost-effective pathway to decarbonise the EU economy, emissions of F-gases should be reduced in the order of 79% by 2030 (Chatain, 2014). In total, F-gases account for 2% of all greenhouse gases in the EU today but have a much more potent atmospheric warming potential than CO2.

The European F-gas Regulation (EU, 2014) has been reviewed with the aim of strengthening the core elements of the Regulation. There are 3 main measures:

* Limiting the total amount of the most important F-gases that can be sold in the EU from 2015 onwards and phasing them down in steps to one-fifth of 2014 sales in 2030. This will be the main driver of the move towards more climate-friendly technologies;
* Banning the use of F-gases in many new types of equipment where less harmful alternatives are widely available, such as fridges in homes or supermarkets, air conditioning and foams and aerosols (Table 5);
* Preventing emissions of F-gases from existing equipment by requiring checks, proper servicing and recovery of the gases at the end of the equipment's life.

Table . Dates of the prohibition of F-gases in many new types of equipment.

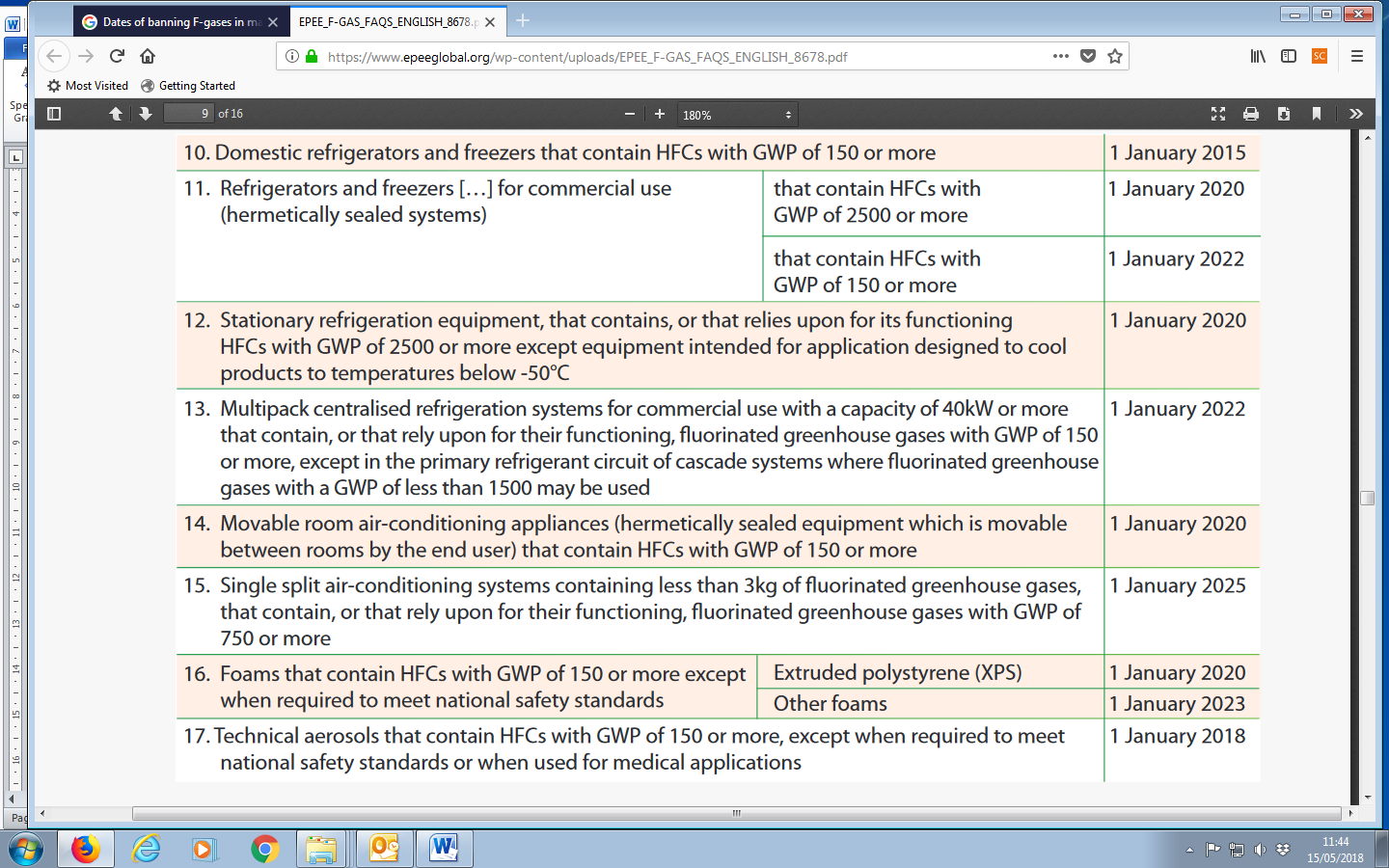


Table . UNEP (2014) classifies refrigerants according to their 100 year GWP

|  |  |  |
| --- | --- | --- |
| **100 Year GWP** | **Classification** | **Current options to fulfil this criteria** |
| < 30 | Ultra-low or Negligible | Natural refrigerants |
| < 100 Very low | < 300 Low | A2L HFOs |
| 300-1000 | Medium | A1 MT HFOs |
| > 1000 | High | A1 LT HFOs, HFC407 and 134a |
| > 3000 | Very high | R404A |

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## Refrigerant – Carbon dioxide (CO2, R744)

Carbon dioxide (R744) is an alternative refrigerant for remote supermarket refrigeration systems. The major benefit over currently used HFCs is that it has a GWP of 1. The major benefit over HCs, (the other alternative to HFCs) is that it is non-flammable.

The drawback of R744 is that the refrigerant has very different properties from other refrigerants. The saturated vapour pressure of R744 at -10ºC is 26 bar, as opposed to 4.3 bar for R404A. This means that pipework and joints need to be able to withstand a much higher pressure. When the saturated vapour temperature is below 31ºC the system will be running subcritically and therefore similar to standard refrigeration systems. However, when the discharge pressure is above about 73 bar, the system will operate transcritically. This means that heat at the high pressure side of the cycle cannot be rejected by condensation in a condenser but instead must be rejected by gas cooling. Whether the system is running sub or transcritically will depend greatly on climatic conditions. This is why R744 systems are more widely used in colder climates, e.g. Scandinavia, and less so in warmer conditions, e.g. Southern Europe. R744 systems can be made to run subcritically by cooling the high pressure side by means other than ambient sensible heat, e.g. evaporative condensers, ground source or by using a cascade system.

When R744 systems run in a pure transcritical cycle (i.e. no let down of pressure into an intermediate pressure vessel), the expansion process is different to a subcritical process. This is because the fluid is not a liquid at the entrance to the expansion valve; instead it is a transcritical fluid.

These issues mean that a non-conventional refrigeration system is required, which is more costly and complex. This causes greater problems with installation and maintenance, as many technicians are not trained for R744 systems. Based on the Australian experience, initial installation costs for CO2 systems will be higher by around 20%, although it is expected that capital equipment costs will decrease once the volume of installed systems grows (Australian Green Cooling Council).

R744 is hazardous to health at reasonably low concentrations, so this also needs to be taken into account if the concentration of CO2 can build up in confined spaces due to a leak. Despite the issues with R744, many retailers believe the benefits far outweigh the drawbacks. Next to Denmark, where large quantities of HFCs are banned, the UK is leading the way in the adoption of CO2 transcritical refrigeration systems (ACR News, 2014). According to natural refrigerants group Shecco, the UK currently has 267 supermarkets running transcritical CO2 systems.

### Cascade

The advantage of a cascade system is that the high pressure side of the R744 circuit can be kept subcritical by using the low pressure side of another refrigerant to cool it. R404A has been used for the high temperature circuit, as it is has been the predominant refrigerant used in supermarkets for many years. As the high temperature circuit is confined to the plant room, leakages of the higher GWP fluid can be kept to a minimum. Campbell, Maidment, and Missenden (2007) describe an R404A and R744 cascade refrigeration system. Their calculations demonstrated considerable energy savings and substantially reduced CO2 emissions (1274 tons CO2 for a 25 kW pack over 10 years) when compared with traditional systems (R404A separate low and medium temperature packs).

However, with future F-gas legislation, another refrigerant is required. Cabello et al. conducted an experimental comparison of a cascade refrigeration facility working with the refrigerant pairs R134a/R744 and R152a/R744. They found that, apart from safety considerations, as R152a is included in the A2 group, the results of the wide range of tests conducted show that no special energy improvement or penalty is achieved. They also concluded that replacement of R134a with R152a is technically and energetically feasible.

### Secondary system

With a secondary system, R744 is pumped as a volatile secondary fluid through the evaporators of the refrigerated appliances where it boils and provides the cooling effect. The return pipe to the receiver carries a mixture of liquid and gas. The high side will primarily use an HFC refrigerant in a conventional circuit.

### Booster systems

A booster system uses two compressors to run both LT and MT packs in the same cycle. Carrier has installed a transcritical CO2 booster system in a Carrefour supermarket in Valencia, Spain, where temperatures average 30ºC in summer (Carrier, 2014). This system uses hydrocarbon subcoolers, economisers and parallel compression to allow the CO2 system to operate efficiently at such high ambient temperatures. After 15 months of operation the system is showing a return on investment of 1.2 years with energy savings of 13% from the refrigeration system.

The Danish supermarket chain Fakta has experienced significant energy savings with their second generation transcritical booster system. Fakta made energy savings of 10% by applying a CO2 system compared to an HFC (Danfoss, 2011). RDM controls have equipped New Zealand’s first transcritical CO2 refrigeration grocery store. The new plant has only been running for a relatively short period, yet early indications are that the system is delivering ‘a double digit percentage reduction in energy use’ compared with conventional systems (Resource Data Management, 2014).

### Benefits

A benefit of the high pressures is that suction lines can be much smaller (at least a 50% reduction in suction pipe diameter), due to the much higher gas density. This causes higher volumetric efficiency of the compressor, allowing it to be 6 to 8 times smaller than those of R22 systems. Heat exchangers can also be smaller or evaporating temperatures increased, for the same reasons. Visser (2002) suggests evaporators may work at 2 K higher than conventional R-404A evaporators. Effects of pressure drops are less significant with R744 systems than conventional systems due to the elevated pressures.

### Costs

Tesco has installed five CO2 systems in China, and the majority of these had up to a 50% cost increase (Gaved, 2013). However, costs have reduced considerably and in 2015 RAC stated that the cost of installing a CO2 transcritical booster system was 4% higher than an efficient HFC system running R407F. With reduced capital costs for R744 systems and new taxes on HFC refrigerants in several countries, the cost benefit of R744 systems compared to HFC refrigerant is likely to increase.

Alessandro, Filho, and Antunes (2012) investigated a cascade cycle (CO2/R404A) system with carbon dioxide for subcritical operation and R404A in the high temperature stage (pump circuit for normal refrigeration and direct expansion for deep-freezing), and also R404A and R22 with direct expansion systems. The cascade cycle was more efficient by around 22.3% in comparison with the R404A system, and 13.7% with the R22 system (both systems operated with frequency inverter and electronic expansion valves). The two racks that make up a cascade system using CO2 on low temperature and R404A on high temperature stages were found to be 18.5% more expensive (based on 2008 values) than single-stage racks using R22 and R404A based on the same cooling capacity. The cost of refrigerants in the CO2/R404A cascade system was $3786 lower than that in the R404A system and $906 lower than that in the R22 system.

Hill Phoenix (2015) assessed the return on investment (ROI) of three CO2 installation examples. They showed an ROI between 0 and 5.6 years. The equipment costs of CO2 were between 10.8 and 24.0% higher than the HFC installation. However, the cost of installation and operating costs were always lower than the HFC.

### Energy

Sawalha (2008) modelled two CO2 systems (centralised with accumulation tank at the medium temperature level and parallel with two separate circuits for low and medium temperature levels) in three different climate conditions, cold, moderate, and hot. The study showed that the CO2 systems, except parallel, are better for cold climates, such as the case of Stockholm. NH3–CO2 cascade systems were better for hot climates, such as the case of Phoenix, USA; in this climate case the CO2 modified centralised system had only 1% higher annual energy consumption than the conventional R404A system and the modifications on the centralised system solution proved to be more important for high ambient temperature operating conditions. This analysis shows that from the energy consumption point of view, the CO2 centralised solution is good for low ambient temperatures and the NH3–CO2 cascade is good for high ambient temperatures. Both systems proved to be good alternatives to an R404A DX system for supermarket refrigeration.

Mazyar and Sawalha (2014) analysed eight supermarkets (five CO2 transcritical and three R404/R410). They showed that while older CO2 systems have lower COPs in comparison with a conventional HFC system, newer systems have comparable or better performance in moderate-cold climates like Sweden.

Yamasaki, Yamanaka, and Matsumo (2012) also evaluated transcritical CO2 systems for many aspects of performance and reliability. For the same level of cooling performance, 20% greater efficiency was realised compared to a conventional R134a cycle. Shilliday et al. (2009) showed that introducing a two-stage compression process to the transcritical R744 cycle increased the COP by up to 12%. Including a 75% effective internal heat exchanger in the two-stage cycle increased the COP of the cycle by around 27% compared with the basic single-stage cycle. Comparing the performance of a two-stage transcritical R744 (with a 75% effective internal heat exchanger) with R404A for use in commercial refrigeration, showed that R404A outperformed the R744 cycle by 10.3 to 28.5%. The basic single-stage R744 cycle was outperformed by R404A when rejecting heat at 40°C by 47.4 to 72.6%.

Karampour and Sawalha (2018) investigated the integrated and state-of-the art features of CO2 transcritical booster systems. The results indicated that two-stage heat recovery, flooded evaporation, parallel compression and integration of air conditioning are the most promising features of the state-of-the-art integrated CO2 system. According to the calculation results, heat recovery in two stages is an energy efficient solution to provide tap water heating and space heating demands. The air conditioning integration into CO2 system is compared with a stand-alone AC system. SEERAC of the CO2 system is comparable to a stand-alone HFC-based AC system. Flooded evaporation and the methods to provide it are discussed. An evaporation temperature increase of 3-4 K in MT and LT levels results in energy savings of about 12% in Stockholm and Barcelona. The refrigeration performance of the state-of-the-art CO2 system was compared to alternative refrigeration system solutions. These included cascade ammonia–CO2 and propane–CO2 solutions, and DX or indirect HFC/HFO solutions. A standard CO2 refrigeration system was considered as the reference system. A comparison of annual energy use (AEU) in Stockholm showed that the state-of-the-art CO2 system was the most energy efficient solution (15% in AEU savings). The AEU comparison in Barcelona indicated that ammonia–CO2 cascade had energy savings of about 20%, and the state-of-the-art CO2 system and R404A DX follow these systems with AEU savings around 15-16%. The R449A DX system is less efficient than the state-of-the-art CO2 system both in warm and cold climates. This shows that the CO2 system is an efficient solution even for warm climates.

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## Refrigerants - HFC retrofit with HFO

Hydrofluoroolefin (HFO) refrigerants are relatively new refrigerants. HFO-1234yf was developed for motor vehicle air conditioning to deal with the EU (MAC) directive. The pure refrigerants are mildly flammable (A2L classification) and are currently being trialled on chiller systems.

Commercially available HFO refrigerants are R1234yf, R1234ze, R450A, R513A and R513B. These refrigerants have been successfully trialled in supermarket applications (see Case Studies below).

R1234ze(E) or “ze” for short and R450A are replacements for R134a and will work at chilled temperatures. R1234ze is classed as mildly flammable. It is not flammable at ambient conditions and needs a lot of ignition energy to ignite it. R450A offers a non-flammable (ASHRAE A1) replacement for R134a. However, the consequence is a much higher GWP than the mildly flammable alternatives. R450A has a GWP of 570 and is composed of 42% R134a and 58% R1234ze.

R448A and R449A are replacements for R404A or R22. They are A1 refrigerants with a medium GWP of 1400. Another replacement for R404A is R455A, which has a much lower GWP of 148. However, it is classified as an A2L.

Table 7 shows an inexhaustive list of HFO based refrigerants that are being used or evaluated for use as retrofit blends for R404A or R134a with comparatively low GWP:

Table . List of HFO based refrigerants

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| R-number | Composition | GWP | Replacement | Safety class |
| 444A | R32/152a/1234ze(E) | 93 | 134a | A2L |
| 445A | R744/134a/1234ze(E) | 120 | 134a | A2L |
| 448A | R32/125/1234yf/134a/1234ze(E) | 1400 | 404A | A1 |
| 449A | R32/125/1234yf/134a | 1400 | 404A | A1 |
| 450A | R134a/1234ze(E) | 570 | 134a | A1 |
| 451A | R1234yf/134a | 140 | 134a | A2L |
| 451B | R1234yf/134a | 150 | 134a | A2L |
| 452A | R32/125/1234yf | 2100 | 404A | A1 |
| 454C | R1234yf/32 | 148 | 22/404A | A2L |
| 455A | R32/1234yf/744 | 148 | 404A | A2L |
| 513A | R1234yf/134a | 631 | 134a | A1 |
| 513B | R1234yf/134a | 596 | 134a | A1 |

### Case studies

R1234ze has been tested by Star in a chiller (ACR News, 2012). They found particularly good performance at part load in lower ambient temperatures. Some of the materials used in joints and seals did not perform well, and special O-rings had to be sourced for some of the system components. They also learned that the lower gas density can cause increased pressure losses which could negatively affect the chiller efficiency.

R1234ze has been considered more applicable for systems with centrifugal compressors. This is because variable-speed centrifugal compressors are not locked in to a fixed swept volume, so the capacity loss associated with this refrigerant in positive displacement machines can be offset by adjusting the compressor speed (Pearson, 2013). The same author stated that, provided components are made available at reasonable cost and the refrigerant is also reasonably priced, R1234ze will be a suitable ultra-low GWP alternative for R134a in variable-speed centrifugal compressors.

R1234ze has also been used by Waitrose in a water chiller to cool the condensers of their integral cabinets (Honeywell, 2012). Frascold has investigated the optimisation of compressors for HFOs.

Yana Motta et al. (2010) investigated R1234ze and R1234yf to replace R134a in an integral vending machine. They showed that comparable performance to R134a could be achieved without significant hardware modifications. Good interaction with POE oils was also demonstrated. R1234ze efficiencies were found to be lower than R134a and R1234yf. These efficiencies were due to system design and different designs could overcome these issues.

Spanish energy consultant Tewis Smart Solutions has reported successful use of the HFO blend R450A as a replacement for R404A in supermarket systems for the Dia Group in Valencia (Cooling Post, 2014). The conversion, aided by Tewis, was reported to be trouble-free with the main change affecting the expansion valves (Cooling Post, 2014).

However, R448A (SolsticeTM N40) has been used in a professional 3 door freezer cabinet (Precision Refrigeration) (ACR News, 2013) and was trialled in a fast food restaurant in London. Precision found that R448A offered better refrigeration performance than R404A with a lower energy consumption. R448A also exhibits lower recovery time after defrost and significantly better pull-down time alongside a reduction of about 65% in GWP over R404A

Makhnatch et al. (2017) carried out a retrofit of R449A into an existing R404A medium temperature indirect supermarket refrigeration system (secondary fluid temperature at the evaporator outlet between −9 and −4 °C). It was demonstrated that with a slight expansion device adjustment and 4% increase of refrigerant charge, R449A could be used in this refrigeration system designed for R404A because of its suitable thermodynamic properties and acceptable maximum discharge temperature. At a secondary fluid temperature at the condenser inlet of 30 °C, the COP of R449A nearly matches that of R404A (both were between 1.9 and 2.2), despite having approximately 13% lower cooling capacity.

R455A has been used by fresh fruit exporter Ortolan in two of its cold rooms (Honeywell, 2018). R455A had advantages over its alternatives. For example, a leak of R744 could damage apples, and R455A has lower flammability than R290 and a GWP much lower than R448A.

### Not yet commercially available

Anther HFO blend which is currently undergoing testing is SolsticeTM N20. It is a blend containing R1234ze. SolsticeTM N20 has a lower capacity than R404A and therefore the refrigeration system may need to be redesigned. It is an A1 refrigerant with a GWP of approximately 1000.

Mexichem are evaluating other blends of refrigerants, such as LTR4X – a non-flammable blend of R32 (28%), R125 (25%), R134a (16%) and R1234ze (31%). It has a GWP of 1300 and is intended as an alternative for R22, R407 and R404A DX applications (Low and Cooper, 2014).

The only work that can be found on these new refrigerants is from Yana Motta et al. (2012) where experimental measurements were performed in their laboratories. These are detailed in Table 8. The refrigerants were found to have the same or better COPs than R404A. Efficiencies relative to R404A were higher at higher evaporating temperatures.

No case studies using these refrigerants in a commercial supermarket have been identified.

### Costs

HFO Blends were initially more expensive than the HFCs they replace. This is due to the increased cost to produce the base HFOs R1234yf and R1234ze. It was reported that R1234yf is likely to remain more expensive than R134a, even at a mature production scale (Low and Cooper, 2014). However, with the reduction in quota at the time of finalising this document (May 2018), R448A and R449A were cheaper than R404A, but still more expensive than R134a. R1234ze was cheaper than R134a. Also, many countries in Europe have or are considering taxes on HFCs and financial incentives to move to lower GWP refrigerants.

Table 8 shows reported capacity and COP for different HFO-based R404A replacement refrigerants.

Table . Capacity and COP of HFO refrigerants at different conditions compared to R404A as well as GWP and ASHRAE class

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| Refrigerant | Condensing temperature | Evaporating temperature | Capacity | COP | Reference | GWP | ASHRAE Class |
| R448A | 35ºC (ambient) | 2ºC (cold volume) | +2% | +5% | Yana Motta, Spatz, and Vera Becerra, 2012 | 1400 | A1 |
| R448A | 35ºC (ambient) | -26ºC (cold volume) | 0% | +9% | Yana Motta, Spatz, and Vera Becerra, 2012 | 1400 | A1 |
| R448A | 45ºC (condensing) | -10ºC (evaporating) | +8% | +15% | Honeywell | 1400 | A1 |
| R448A | 45ºC (condensing) | -30ºC (evaporating) | +0% | +13% | Honeywell | 1400 | A1 |
| R448A | unknown | freezer | similar | +9% | Sethi et al., 2016 | 1400 | A1 |
| R455A | unknown | freezer | similar | +6% | Sethi et al., 2016 | 150 | A2L |
| N-20 | 35ºC (ambient) | 2ºC (cold volume) | -18% | 0% | Yana Motta, Spatz, and Vera Becerra, 2012 | ~1000 | A1 |
| N-20 | 35ºC (ambient) | -26ºC (cold volume) | -20% | +2% | Yana Motta, Spatz, and Vera Becerra, 2012 | ~1000 | A1 |
| R455A (L40) | 35ºC (ambient) | 2ºC (cold volume) | -4% | +2% | Yana Motta, Spatz, and Vera Becerra, 2012 | 148 | A2L |
| R455A (L40) | 35ºC (ambient) | -26ºC (cold volume) | -3% | +6% | Yana Motta, Spatz, and Vera Becerra, 2012 | 148 | A2L |

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## Refrigerants - HFC retrofit with hydrocarbons

Hydrocarbon (HC) refrigerants are natural, lower toxicity refrigerants that have no ozone depleting properties and negligible global warming potential.

Hydrocarbon refrigerants have an A3 safety classification (listed in EN378). Due to the flammability of HC refrigerants, the quantity of refrigerant which can be used in a refrigeration system is limited such that they are unable to be used in distributed systems where refrigerant is transported from a central condensing unit to multiple evaporators through piping. This limit is generally less than 1.5 kg for General Occupancy Class A (EN 378)

For distributed systems, secondary circuits must be used to supply liquid-cooled condensing units or cooling coils directly. They can, however, be used in integral cabinets as long as charge restrictions are observed.

Isobutane (R600a) has a lower volumetric refrigerating effect (VRE) than R134a. It is however compatible with most common materials and lubricants.

Propane (R290) and propylene (often called propene) (R1270) are similar to R404A in terms of saturated vapour pressure and VRE and therefore can be used as a retrofit. However, this is in principle not advised due to the existing HFC equipment not being designed or constructed with regards to flammability mitigation measures.

HC refrigerants increase the energy efficiency of the individual systems, typically between 10 and 20% (King and Garvey, 2011). R1270 was also shown to reduce electrical consumption of the compressor by over 10% compared to R404A.

Arnemann et al. (2012) carried out experiments with a scroll compressor. They showed R290 and R1270 have a higher COP than R404A at higher pressure ratios. At low pressure ratios, R290 does not show such an advantage, while R1270 always has a better COP than R404A. At an evaporating temperature of -10ºC, R290 has a better COP than R404A, between 0 and 29% for a condensing temperature of 30 and 60ºC, respectively. At the same conditions, R1270 has a better COP of between 8 and 15%.

Arnemann et al. (2012) also stated that R290 has a similar operating envelope to R404A, though R290 also has high discharge temperatures. Reliability tests with R290 and a standard POE oil used for HFCs have shown some concerns. The cooling capacity of R290 is lower than R404A, especially at low condensing temperatures.

Pederson (2012) showed a 7.9% and 8.3% increase in COP for R1270 and R290 over R404A at an evaporating temperature of -10C and condensing temperature of 35ºC. The increase in COP of these refrigerants rises to a maximum of 10% at a condensing temperature of 45ºC. R290 has a GWP of 5 and R1270 a GWP of 1.8.

It is technically feasible to remove R404A or R134a from existing systems and replace them with R290, R1270 or blends containing HC, as appropriate. However, it is highly likely that the resultant system will not comply with safety rules related to the application of HC refrigerants because the refrigerant quantity is unlikely to comply with charge limits and the electrical equipment will not be suitably protected. There may be a case for a controlled conversion from HFCs to HCs where the system efficiency can be improved. In this case, it is essential that suitable safety measures are undertaken. Such guidelines have been published in, for example, GIZ (2010, Chapter 6) and GIZ (2014).

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## Refrigerants - HFC retrofits with lower GWP HFCs

There are a number of low GWP HFCs, which can be retrofitted in systems using R404A and R134a. Those with a high and medium GWP (300 to 3000) typically have an A1 safety classification, which means they have lower-toxicity and are non-flammable. A number of other blends have an A2L safety classification, having lower toxicity and lower flammability. Those with A1 classification may be applicable to direct distributed, remote and integral systems, both low and high temperature, whereas those of A2L classification are less likely to be applicable to direct distributed and remote systems.

There are numerous blends for the replacement of direct systems, but none can replicate the pressure-temperature relationship, volumetric refrigerating capacity and efficiency of R404A and R134a exactly. Furthermore, most blends have two or more components, resulting in significant temperature glides, so suitability in heat exchanger designs can be of concern. When considering class A2L blends, it is unlikely that the resultant system will comply with safety rules related to the application of flammable refrigerants because the refrigerant quantity may not to comply with charge limits and the electrical equipment will not be suitably protected. There may be a case for a controlled conversion from HFCs to A2L refrigerants where the system efficiency can be improved and there is a major reduction in GWP. In this case, it is essential that suitable safety measures are put in place. Such guidelines have been published for HCs in, for example, GIZ (2010, Chapter 6) and GIZ (2014) and in principle these can equally be applied to A2L refrigerants if the appropriate values are used.

Table 9 is a not an exhaustive list of refrigerants that are being used or evaluated for use as retrofit blends for R404A with comparatively lower GWP:

Table . HFC Refrigerants that are being used or evaluated for use as retrofit blends for R404A with comparatively lower GWP.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **R-number** | **Composition** | **GWP** | **Replacing** | **Safety class** |
| 407A | R-32/125/134a | 2100 | 404A | A1 |
| 407B | R-32/125/134a | 2800 | 404A | A1 |
| 407C | R-32/125/134a | 1700 | 404A | A1 |
| 407D | R-32/125/134a | 1600 | 404A | A1 |
| 407E | R-32/125/134a | 1500 | 404A | A1 |
| 407F | R-32/125/134a | 1800 | 404A | A1 |
| 407H | R-32/125/134a | 1495 | 404A/507 | A1 |

R407A (GWP of 2100) has similar properties to R22. R407A is typically used in low and medium temperature refrigeration applications. R407F has a pressure profile which better matches R22 than R407A and can be used in low and medium temperature refrigeration applications. It has a higher capacity than R407A at low temperatures. It has a lower GWP than R404A but higher than R134a. However, recently these refrigerants are becoming less popular choices than R448A and R449A.

### R407 Case studies

Danfoss (2013) states COP improvement of 5 and 7% for R407A and R407F respectively at an evaporating temperature of -10ºC for reciprocating compressors.

Yana et al. (2012) performed tests using a commercially available condensing unit and an evaporator for a walk-in freezer/cooler. They showed benefits in COP between 0 and 3% for R407A and 5 and 6% for R407F with the largest benefits being at the lowest cold store temperatures.

Milnes (2011) states that the energy efficiency can be improved by 7% to 12% because the new refrigerants (R407A or R407F) have superior efficiency characteristics to R404A. A few minor design changes may be required (e.g. changes to expansion valves) but the cost of such changes is small. These savings are approximately twice those reported by Yana et al. (2012) and Danfoss (2013). Milnes (2011) goes on to say that ‘the retrofit programme should also include a thorough check of all components and plant recommissioning. There are many examples of where this process has uncovered previous problems and led to overall energy savings well above the 7% to 12% target’.

Fricke, Abdelaziz, and Vineyard (2013) reported that ‘R407A showed a 29% emission reduction compared to R404A in the multiplex DX system. Energy consumption was shown to be roughly the same (slightly higher and slightly lower for different stores). R407A is an attractive drop-in replacement for R404A and requires minimum system modifications’.

Table 10 shows the increase or decrease in both capacity and COP for R407A and R407F at different conditions.

Table . Capacity and COP of different HFC refrigerants at different conditions compared to R404A

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **Refrigerant** | **Condensing temperature** | **Evaporating temperature** | **Capacity** | **COP** | **Reference** |
| R407A | 45ºC | -10ºC | -5% | +5% | Danfoss, 2013 |
| R407F | 45ºC | -10ºC | +3% | +7% | Danfoss, 2013 |
| R407A | 35ºC (ambient) | 2ºC (inside cold volume) | -8% | 0 | Yana et al., 2012 |
| R407A | 35ºC (ambient) | -26ºC (inside cold volume) | -12% | +3% |
| R407F | 35ºC (ambient) | 2ºC (inside cold volume) | +4% | +5% |
| R407F | 35ºC (ambient) | -26ºC (inside cold volume) | -2% | +6% |

### Costs

According to the Öko Institut (2017) prices for R134a, R404A, R407C and R410A at the level of other equipment manufacturers (OEMs) remained reasonably constant between 2014 and the last quarter of 2016. The biggest rise was for R404A which rose about 25% over this period. Between then and the second quarter of 2017, all these refrigerants have risen significantly in price, with R407C and R410A rising the least (30%) and R404A rising the most (130%).

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## Secondary systems

Secondary loop systems use a contained primary refrigeration system (usually in a plant room) to cool a pumped secondary fluid. These secondary fluids are generally brines or glycol based fluids but can be CO2 or ice slurries.

As the primary refrigeration plant is contained in a plant room, there is less restriction on choice of refrigerant (flammable and/or toxic refrigerants can be considered) and also refrigerant leakage is lower due to reduced refrigerant piping lengths and number of connecting joints. Also, leaks can be detected in areas of easier access, i.e. plant room rather than a stocked cabinet. Therefore, CO2 emissions associated with refrigerant leakage can be reduced. Kauffeld (2007) suggests reductions in refrigerant charge of up to 80 to 90% are possible and with higher quality factory assembly of the primary refrigeration system, there is a lower risk of leaks. Baxter (2003) reported a similar level of reduction in refrigerant charge requirement of about 85 to 90% of that needed for a conventional direct expansion system.

DelVentura (2008) claimed leakage from a traditional DX system reduced from 30% to less than 5% when using a secondary system. The reductions in direct CO2 emissions were dependent on the refrigerant being used. Pajani et al. (2004) estimated that leaks could reduce by 96% (from 250 to 10 kg/yr). This is equivalent to a 3% leak rate (compared to 16% with DX) with an 80% lower charge.

As secondary fluids can be either single- or two-phase, some of the advantages and disadvantages may differ depending on which is chosen. Wang et al. (2010) stated that the advantages of two-phase secondary refrigerants include lower pump power, smaller pipe sizes and excellent heat transfer properties. However, the disadvantages of two-phase secondary refrigerants are higher initial costs and higher pressures in the case of CO2. Despite the overall reduction in direct emissions through the use of a secondary fluid, the additional heat exchange process and pumping energy can reduce the system efficiency and the direct electricity consumption can be greater for the same cooling load served by a DX system. Devotta and Sicars (2005) state that the relative energy consumption of indirect (secondary) systems compared to DX systems can increase by up to 15%. However, some research has highlighted that this is not always the case. Wang et al. (2010) reviewed the case for a range of low GWP alternatives to HFCs and concluded that there was a strong case for a small refrigerant charge and a secondary loop. They also concluded that with newly available technologies and materials, there was a good economic case for secondary loop systems which would bring environmental benefits and energy savings (however, the magnitude of savings was not stated and the results to support the statement were not found within the paper).

Hagland Stignor et al. (2007) carried out a theoretical case study comparing five different cases of indirect cooling systems in supermarkets. They found the following:

* Energy use as well as cost for an indirect cooling system could be reduced considerably by replacing a conventional cooling-coil by a more efficient unit.
* Replacing propylene glycol, 39%w, by Temper -20 in a system with traditional conventional cooling-coils results in lower energy use, but a system with more efficient heat exchangers adapted to the laminar flow regime is less sensitive to the selection of a secondary refrigerant.
* Excessively high values of the liquid flow rate in an indirect cooling system result in a high cost of investment, since larger dimensions of the components must be selected.

Work by Hrnjak (2000) suggested that the performance of the heat exchanger would not be impeded by using a secondary coolant to replace direct expansion of refrigerant. When comparing potassium formate (as a secondary coolant) with R404A (in direct expansion) in a low temperature cabinet, it was shown that the performance of the heat exchanger with the secondary coolant exceeded that with R404A. It was therefore likely that heat exchanger size or design would not require significant alterations to accommodate secondary coolants. In the study the expansion valve and distributor were removed to convert the heat exchanger from R404A to potassium formate.

The corrosiveness of the secondary refrigerant needs to be taken into account, especially if the secondary refrigerant will be retrofitted to existing pipework and heat exchangers. Typical glycols have a low, organic salts have a moderate, and inorganic salts have a high level of corrosivity. According to Hillerns (2001), even polyethylene glycol will corrode all relevant refrigeration materials at a rate greater than 0.1 mm/yr. However, when inhibitors are applied, both potassium acetate and formate have less than 0.1 mm/yr corrosion on all relevant refrigeration materials except soft solder.

At Loblaws supermarket in Canada, a secondary system was expected to produce energy savings of 18% in refrigeration and heating and a 73% reduction in CO2e emissions (Pajani et al., 2004). The supermarket used two refrigeration loops, with potassium formate brine in the low temperature loop at -25ºC and propylene glycol brine at -5ºC for the medium temperature loop. A third loop using ethylene glycol was used to recover heat from the condensers for space heating and hot water and heat rejection of surplus heat to the ambient air.

Haaf and Heinbokel (2002) compared three system solutions; the first was a two-stage NH3 indirect system with brine at the medium and low temperature levels. The second was a single-stage propane indirect system with brine at the medium and low temperature levels. The third was a CO2 cascade system with NH3 or R404A in the high stage. The field study comparisons were made with a conventional DX R404A system. NH3 was found to be 20-30% more expensive in terms of investment costs and had 10-20% higher energy consumption. The propane system had 15-25% higher investment costs and 5-20% higher energy consumption. The cascade system had investment costs and energy consumption almost the same as DX R404A.

Field trials by the California Energy Commission reported energy savings of 4.9% over a nine-month period for a secondary loop system versus a baseline DX system (DelVentura et al., 2008). The study compared a baseline system with a distributed DX of refrigerant with a glycol secondary loop system; the energy savings were largely attributed to the improved performance of the refrigerated cabinets enabling an elevated evaporation pressure in the refrigeration system despite the need for the intermediate heat exchanger to cool the glycol.

Sánchez et al. (2017) experimentally analysed the energy impact of conversion of a direct HFC134a/CO2 cascade refrigeration system to an indirect HFC134a-secondary fluid / CO2 cascade for commercial applications. Their results showed an increase in the energy consumption of the whole system between 7.6 to 14.0% when using propylene-glycol/water and between −0.3 to 11.1% when using Temper −20® as secondary fluid.

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## Shelf risers and weir plates

Risers are simple strips of plastic (usually clear) fixed to the front of shelves. They typically extend around 50 mm above the shelf and have the purpose of retaining product and reducing entrainment. Weir plates are also strips of plastic (usually in the order of 100 mm high) but can also be glass that fit on the front of the well between the air curtain return grille and the ambient. They have the added benefit that they create a well of cold air in which the base packs are positioned.

They are suitable for use on, and likely to enable energy savings in, any open fronted refrigerated display cabinet where they are not currently fitted. The effect of fitting a riser and weir plates is typically an improvement in temperature of product sitting on the well (particularly front product), and a reduction in energy.

Risers and weir plates are an established and simple technology. They can be fabricated inexpensively and are already in use on many display cabinets.

The major drawback is that they can impede customers buying product and the retailer loading product. For this reason they are sometimes removed from the cabinet. They also tend to be damaged or knocked off and are often not replaced.

Carbon Trust (2012) suggests energy savings between 2 and 4% by fitting risers.

## Short air curtains

In order to overcome the problem of back panel flow supplying the coldest air directly to where it is needed least, a design was patented by Linde for chute shelves (Schuster and Krieger, 2007). The chute shelf delivers cold air from the underside of each shelf front to reduce dependency on back panel flow whilst enabling a slower air curtain over the front of the case.

The authors reported on a patented design for an open vertical cabinet for merchandising meat, where precise temperature control was required. The cabinet used a chute shelf design to provide a short curtain for each tier and a conventional vertical air curtain over the entire open front. The cabinet did not achieve the target 2 K temperature variation in stored product temperature under the EN ISO 23953 (2005) test, but did achieve a 3 K range, half that of the standard cabinet which used back panel flow to stabilise its air curtain.

Unsupported (no back panel flow) short air curtains can remain stable for their entire height with slower discharge velocities than unsupported taller curtains (Hayes and Stoeker, 1969b) and so have a lower entrainment rate, making them more efficient. It therefore follows that if back panel flow is to be designed out of conventional cabinets, a series of short curtains would provide the way forward.

Research by Hammond, Quarini and Foster (2011) concluded that it was advantageous from an energy efficiency viewpoint to use multiple short air curtains rather than one single long curtain to seal the open front of a vertical multideck display cabinet. Previous research by Axell (2002), Schuster and Krieger (2007) and Stribling et al. (1995) also implied there was practical scope to do this. Axell (2002) investigated curtain height and its effect on the air curtain and concluded ‘...for the same height/width ratio, a short air curtain is more efficient than a high air curtain’.

Shorter curtains were reported to eliminate the need for back panel flow which enables tighter temperature control independent of cabinet loading (ACR News, 2013) (The Grocery Trader, 2013). Axell (2002) made similar observations: ‘…the arrangement of blowing cold air from the rear duct is not the optimal way of cooling product in an open display case, as the cold air emerging from the rear duct impinges on the coldest of the products on the shelf. Back panel flow is unreliable and is not suited to varying product loading patterns’.

It is well reported that doors are not a one-size-fits-all solution. Adande Refrigeration has developed a cabinet based on short air curtains called ‘Aircell’ which is being proposed as a viable alternative to doors. Aircell works by dividing the case’s merchandising envelope into separate air flow managed cells with short, low pressure air columns. Each cell has its own air curtain which is more efficient than the full height air curtain on a conventional multideck case. The net result is less pressure on the air curtain of each cell and a substantial reduction in cold air spillage from the case (Wood, 2013). Adande claims 30% less energy consumption compared with conventional open front, refrigerated multi deck display cases (Adande, 2015). Independent tests have shown that temperature fluctuations of just 2.8K can be achieved in an Aircell cabinet (ACR News, 2013, Food and Drink International, 2013).

### Efficiency

Aircell has been presented in the trade press as an alternative to doors (ACR News, 2013). It promises similar energy savings (circa 30%) to fitting doors and tighter temperature control than conventional cabinets. Aircell leaves the cabinet open and so does not present a barrier to sales and does not require additional trim heating. In 2016 Adande received a grant to productionise the technology with the Bond Group.

### Costs

Aircell is still under some level of development and Adande claims that the cost for fitting the technology (to a new cabinet) is not expected to be any more expensive than fitting glass doors to a cabinet.

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## Store dehumidification

Entrainment or infiltration of ambient air into a display cabinet through the open front of the cabinet accounts for around 78-81% of the total heat load on the cabinet (ASHRAE, 2002). For 25°C 60% RH ambient (class 3 BS EN ISO 23953 conditions) around half of the infiltration load is latent. The condensation and subsequent freezing of the moisture removed from the ambient air results in frosting of the evaporator coil. Frosting then reduces the performance of the cabinet by reducing the effectiveness of the evaporator (frost forms a thermally insulating layer) and by reducing the volume of the air circulation (due to increased pressure against fans) which weakens the air curtain and so increases the rate of infiltration of ambient air into the cabinet.

Dehumidification of the ambient air reduces the latent heat load portion of the infiltration load. Work by Howell (Howell, 1993a, Howell, 1993b and Howell et al., 1999) has shown that even after the energy required to dehumidify the ambient air is considered, the energy saving is around 5% for every 5% reduction in store humidity (based on reduction in store humidity from 55% to 35 at 14 to 7ºC dew point temperature). In the UK dew point varies from about 1ºC in winter to 12ºC in summer. It is therefore applicable to dehumidify UK stores in the summer, but in winter the humidity is already quite low according to thermal comfort requirements.

Orphelin, Marchio and D’Alanzo (1999) estimated that in a typical French supermarket store, if RH was controlled to 40%, an annual reduction of 4% would be made on the refrigeration power and 0.1% on the total store power. At an RH of 45%, an annual reduction of only 2% would be made on the refrigeration power but a more significant 0.8% on the total store power. This shows that there is an optimum RH where the benefits to refrigeration power start to be outweighed by the increase in dehumidification power.

When determining the ideal store conditions, the comfort of the shopper must be considered. It is likely that drier, warmer air will be more acceptable to shoppers than cooler air (Pursglove, 2013; Tassou and Xiang, 2003; Ndoye, Mousset, Carlier, and Arroyo, 2011). Low humidity can also lead to dry skin, irritated sinuses and respiratory tract, and itchy eyes. Over time, low humidity can dry out and inflame the mucous membrane lining your respiratory tract, although it is unlikely that the time spent shopping in a supermarket store will lead to such severe effects. Reducing the dew point to below 2°C can also result in eye irritation (ASHRAE, 2013).

It is not typical to dehumidify stores to reduce energy consumption of the refrigerated cabinets. Some stores have cold air recovery from the “cold aisles” where the refrigerated cabinets operate. These systems will increase shopper comfort but will also increase the cooling load on the display cabinets as a result of raising the aisle (and so also the entrained air) temperature.

Fricke and Sharma (2011) investigated the potential energy savings associated with reducing the relative humidity in the vicinity of refrigerated display cases in supermarkets, as compared to the widely accepted current practice of maintaining a relatively high and uniform humidity level throughout the entire supermarket. They showed that for medium temperature refrigeration systems, refrigeration system energy use decreases anywhere from 15% to 22% when the relative humidity is reduced from 55% to 35%. For low temperature refrigeration systems, refrigeration system energy use decreases anywhere from 0% to 17% when the relative humidity is reduced from 55% to 35%.

Adaptive defrosting (or at least a less frequent defrost regime) may be required for the display cabinets in order to fully realise the potential benefits of store dehumidification (Tassou and Datta, 1999).

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## Store temperature control

Store temperature control is as much about distributing heat within the store as it is about the heating or cooling systems used to maintain the desired conditions. In areas of stores with refrigerated cabinets, the aisle temperatures are notoriously cool. Around the bakery area, temperatures can be too hot.

Current technology involves cold air retrieval, HVAC systems, mechanical A/C and gas fired heaters.

Thermal comfort is a human-experienced sensation that occurs as a consequence of a scenario of environmental and personal variables. The environmental variables such as air temperature, mean radiant temperature, relative air velocity and ambient water vapour pressure are within the control of the HVAC system of a store. However, human, personal variables such as activity level and clothing are not (ASHRAE, 2013).

Cold aisles near refrigerated cabinets are the most common complaint from shoppers but most remedies for this would result in increased energy consumption of the store. Fitting doors on cabinets does, however, allow both the energy consumption of the cabinets to be reduced and improve customer comfort. Lindberg et al. (2008) showed that retrofitting glass doors on vertical display cabinets reduced the temperature gradient between ankle and head in front of a dairy cabinet by 2 K.

The cold air spillage can be removed using a simple ventilation system, often referred to as cold air retrieval, heated floors or diffusers in the ceiling to blow warm air into the aisles (Foster and Quarini, 2001). Many stores already implement such systems but often fail (perhaps deliberately to strike a balance with efficiency) to maintain the perception of thermal comfort amongst shoppers.

Where integral refrigerated cabinets are used, water-cooled condensers can enable excess heat to be removed from the store or used to satisfy a heating demand elsewhere. Some manufacturers have also offered cabinets with dual condensers, one to reject heat to the store and one to reject heat to a water loop. The refrigerated cabinets then become a significant part of the store energy management.

Allowing a broader tolerance on the acceptable temperature band can offer some savings. The cost for implementing this is minimal as it involves only a simple change to the store control system.

There is little that the store can do to reduce the effect of the mean radiant temperature on the shoppers; glass doors could go some way to helping but only where the glass is suitably reflective to radiant heat.

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## Strip curtains

Strip curtains provide less of an obstacle to sales than a glass door but can be awkward where larger products are being accessed.

Strip curtains are more commonly found on cold store doors than open fronted display cabinets. They consist of clear plastic strips hung over the opening of the cabinet to provide a separation between the cold and warm air whilst only presenting a partial reduction in visibility.

Strip curtains are faster and cheaper to install than glass doors and so offer a faster payback but they tend to be less transparent and require more maintenance to keep them clean and tidy. Grades of PVC can vary and it is important that suitable UV stabilisers are added to prevent a loss of transparency of the strip curtain and furthermore that the PVC grade is suited to the temperature of application.

Redwood, a manufacturer of strip curtains, claims their curtains will reduce energy use by between 18 and 60%; based on their own research and customer feedback (Redwood Strip Curtains Ltd). The cost for a typical 2.4m x 1.7 m strip curtain for a chilled multideck cabinet would be £156 at 2017 prices.

Condensation on the strips may cause a problem; however, the authors present no evidence for this.

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## Suction pressure control

The required suction pressure is not a constant; it will depend on the load. The suction pressure would normally be set to allow adequate cooling at peak load. During the winter and overnight a higher suction pressure may be acceptable.

Compressor power is directly related to pressure ratio, so increasing suction pressure will reduce pressure ratio and therefore compressor power. Increasing the suction pressure will also increase duty. According to typical compressor data from Bitzer (compressor 4NES-14Y-40P on R404A), raising the evaporating temperature by 4 to 5ºC (typical with suction pressure optimisation) will increase COP by 11% (chiller) and 13% (freezer).

The suction pressure obtained is varied by altering the number of compressors in the pack that are operating. The suction pressure prevailing is compared with the suction pressure set-point to control the number of compressors in operation. It is possible to vary the set-point according to external factors, such as the external ambient temperature. This can save significant amounts of energy.

Sometimes, for convenience, cabinets are connected to systems that are aimed at higher temperatures than that for which the system has been set up. To enable this to be done, mechanical evaporator pressure regulators (EPRs) are used to maintain the higher set pressure to the refrigerated cabinets. The pressure drop across the EPR represents a cost. It will be understood that the suction pressure of the compressor needs to be below the pressure of the EPR.

One way of implementing suction pressure control is to use an electronic EPR that varies the pressure by using a stepper motor. The suction pressure will be controlled to maintain cabinet temperature rather than fixed suction pressure. This varies the evaporator temperature in the cabinet. The different evaporator temperature feeds through to providing a different amount of cooling to the cargo in the cabinet.

This type of suction pressure control means that the evaporating pressure (and, therefore temperature) varies independently for each cabinet. The suction pressure provided to the system as a whole (from the refrigeration pack) has to be lower than that required by the “most needy” cabinet. To ensure that this is so, it is done separately for each load (cabinet). Therefore suction pressure will vary to maintain a constant cabinet temperature. It is necessary for the pack controller to be in communication with the cabinets and to adjust the suction set-point dynamically in operation. This lower pressure will mitigate against overall refrigeration system efficiency.

Another way of implementing suction pressure control is to vary the pressure set-point at the compressor pack dynamically without altering valves on the cabinets. The input for the change in the pack pressure set-point can be ambient temperature in the shop or, better, information about the state (i.e. refrigerating, defrosting, recovering from defrost etc.) of and “hunger” of each of the cabinets. The temperatures for each of the cabinets is then controlled by a solenoid valve switching the refrigerant flow on/or off to the evaporator.

Lawrence and Gibson (2009) state that it is important that all cabinets work properly. If there is a problem with one or two cabinets, they should be fixed.

The means by which suction pressure control operates to reduce energy use is by running fewer compressors for shorter periods of the day. This is best done by acting on the pack controller directly.

It is possible to purchase a kit that converts a mechanical expansion valve to an electronic valve with no need to braze in new valves, reducing installation labour and component cost. Nevertheless, control circuits for the new valve will be required.

Lawrence et al. (1998) analysed the potential savings of suction pressure optimisation for three climatic conditions, Barcelona (Spain), Birmingham (UK) and Bergen (Norway). They predicted savings from 8.6% for frozen packs in Bergen to 14.7% for chilled packs in Birmingham.

Lawrence and Gibson (2009) state that typically suction pressure control saves 15% of pack energy.

A study conducted by a supermarket chain (Parker Hannifin Corporation, 2010) showed an 11.4% reduction in energy for the low temperature pack and 1.5% reduction for the medium temperature pack when suction pressure was controlled.

A study by California Utilities Statewide Codes and Standards Team, 2013 showed that floating suction pressure was considered cost-effective (based on a Life Cycle Costing Methodology and not just financial cost) for all system configurations and in all climate zones.

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## Tangential fans

Air flow in retail display cabinets is usually provided by axial fans interspersed across the length of the cabinet. The diameter of the fan is usually bigger than the height of the evaporator, so the fan is angled downwards to allow it to fit. This leads to a very uneven flow onto and through the evaporator.

The small ratio of impeller diameter (30 to 65 mm) to impeller length means that tangential fans are an easy fit within retail display cabinets. The impeller length can cover the full length of the cabinet e.g. 2.5 m and the impeller diameter the height of the evaporator. This leads to a very linear even flow onto the evaporator.

It is claimed that tangential fans can provide more even air flow in cabinets and have been shown to produce overall energy savings of 2% (Faramarzi et al., 2000). The savings are relatively small and are probably related to the slight increase in evaporating temperature that can be achieved if air flow is more uniform.

The use of tangential fans in place of axial fans could reduce the number of motors required and thereby reduce capital cost. A disadvantage of tangential fans may be that they are more difficult to clean in comparison to axial flow fans.

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## Thermostatic flow control (TFC)

Thermostatic Flow Control (TFC) is a new, flooded evaporator technology, applicable for all sizes of plants and all refrigerants, except those with big temperature glide (Zimmermann, 2007). The flow to the evaporator is controlled by the receiver pressure, which is controlled by a heat exchanger – without any moveable parts.

TFC has two-step throttling separated by a receiver with integrated suction gas heat exchanger (lower diagram in Figure 10). The pressure drop between the condenser and evaporator is shared between the two steps, and the receiver pressure is controlled by the difference between gas supplied and gas removed. If the heat exchanger removes more gas than supplied by the first throttling step then the receiver pressure goes down – otherwise the receiver pressure goes up. According to the patent, it is possible to ensure the evaporator is always flooded, though the authors could find no independent evidence for this.

The primary benefits of this system are the improved system efficiency as in any flooded evaporator system (see section on flooded evaporators). The secondary benefit is that this is done without either a pump or a gravity head and float valve allowing it to be economic for smaller systems (Zimmerman claims down to 100 W).

Flooded evaporators allow evaporating temperature to be raised compared to DX evaporators due to more effective evaporation. This has an added advantage where product dehydration is concerned. A higher evaporating temperature means a higher dew point in the cold area. TFC is used by Norbake (Norbake Services Ltd, 2015) for its dough controllers and humidity in the cabinets is claimed to be 92 to 93%. They quote energy savings of 18% compared to traditional units.

Zimmermann (2004) also claims that the system has been used in specialised plug-in freezers in supermarkets and to have saved 16% of the energy compared to the original capillary DX system. The study also claimed that milk cooling tanks have shown energy savings of 17%.

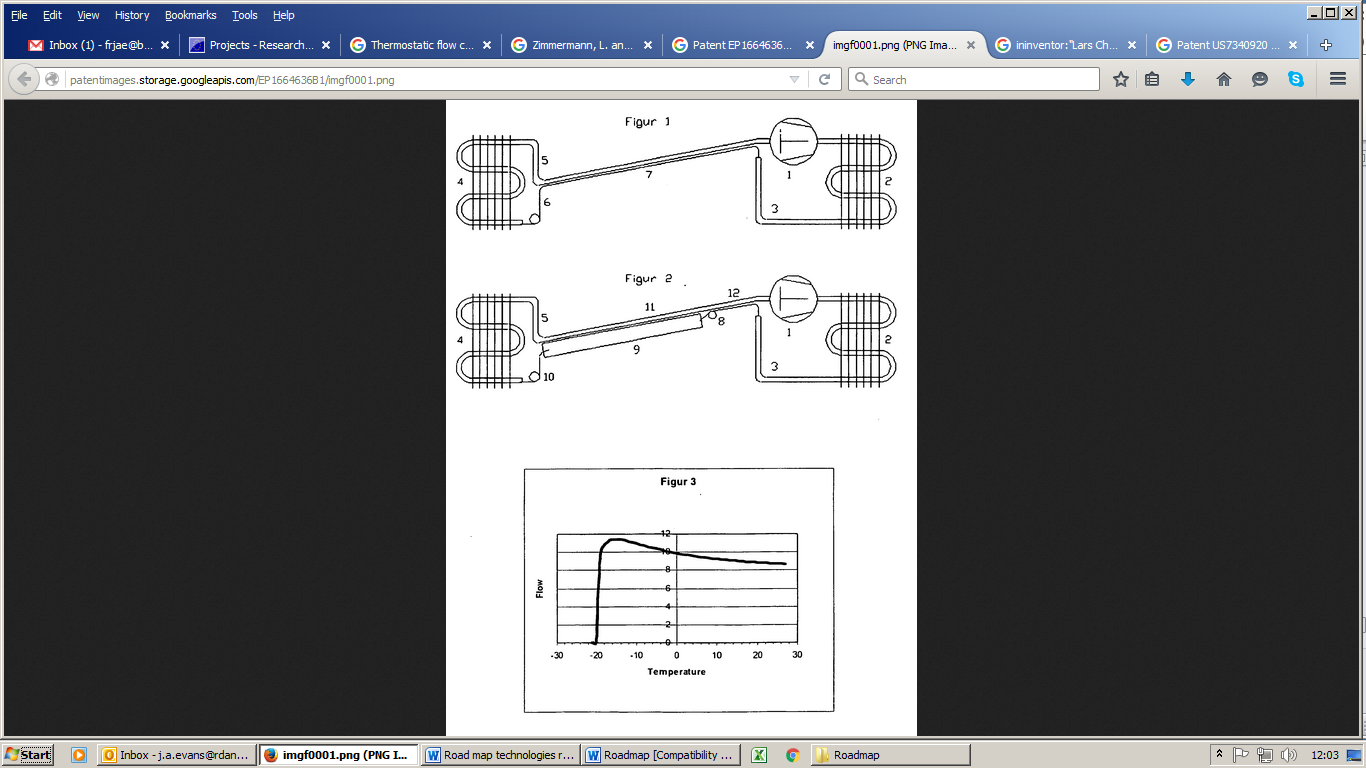


Figure . TFC concept from Zimmermann, 2007 (upper diagram conventional capillary system, lower diagram TFC system).

**Compressor (1), condenser (2), evaporator (4), receiver (9), and with capillary throttling (8) between condenser and receiver, and with capillary throttling (10) between receiver and evaporator characterised by thermal contact (11) between suction line and receiver, and orientated so that suction gas passes from receiver bottom towards receiver top.**

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## Training and maintenance

Poor cabinet loading, excessively cold set-points and misuse of features such as night blinds or night set back all lead to increased energy consumption and risk damage to (or reduced quality of) the stored product.

Maintenance staff can cause a significant increase in energy usage through misguided adjustment of set-points or poor commissioning. Every degree warmer the condensing temperature is set, or degree cooler the evaporating temperature is set, will result in a 2 to 4% increase in energy consumption (Action Energy, 2003). Similar losses occur as a result of dirt/debris build up on condensers as this also increases the condensing temperature. Fennelly (2014) states that dirty condensers are the biggest single reason for non-scheduled service calls.

Electric defrosts on frozen cabinets are only about 15% efficient (85% overhead) (Lawrence and Evans, 2008) and so significant energy wastage can result from an overly cautious defrost schedule. Conversely, excessive frosting of the evaporator reduces the efficiency of the system and can, by reducing air flow within the cabinet, lead to poor product temperature control and increased heat loads through infiltration.

On cabinets with doors, the seals should be inspected, cleaned and replaced where necessary. A leaking door seal will increase the heat load on a cabinet, increasing the energy consumption and leading to poor stored product temperature control.

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## Trigeneration

Trigeneration (otherwise known as Combined Cooling Heat and Power, CCHP) is the production of cooling, heat and electrical power in one combined process. Typically this is based on oil or gas fired generators which produce electricity, and in the process generate heat. The heat can be used directly e.g. for hot water or space heating, but it can also be used as required by an absorption chiller (or more rarely an adsorption chiller or a desiccant cooling system) to produce cooling. The proportions of electrical power, heat and cooling can be varied depending on demand. Sources of waste heat and renewables such as biogas can also be usefully exploited in trigeneration systems.

There has been considerable research on trigeneration systems and their performance, summarised in several comprehensive review papers (Cho, Smith and Mago, 2014; Jradi and Riffat, 2014; Liu, Shi and Fang, 2014; Deng, Wang and Han, 2011; Wu and Wang, 2006). The primary generators include internal combustion engines and gas turbines and, with continuing research, fuel cells, Rankine cycles and Stirling engines also show promise. The strategy chosen for operation of trigeneration systems is critical to their efficiency (see, for example, Jradi and Riffat, 2014), with options being to follow thermal load, follow electrical load or to optimise a combined strategy.

The use of trigeneration specifically for supermarket applications has also been researched. Maidment et al. (1999) theoretically assessed the performance of a CHP system used to supply heat and power to a relatively small UK supermarket. In the standard application, a vapour compression system was used to supply display cabinet cooling but the addition of absorption cooling driven by waste heat from the CHP system was also assessed. The absorption cooler supplied glycol at -10˚C, and this was used to cool the chilled display cabinets. Primary energy savings of 20% and a payback period of 6 years were estimated. Extending the use of the absorption system to supply the frozen cabinets was found to be impractical. In further work (Maidment and Tozer, 2002), various CCHP configurations and CHP engine sizes were modelled and compared with conventional supermarket energy performance. The optimum configuration was found to be a lithium bromide / water absorption system for chilled water between 7 and 14ºC (the cabinets cooled by cascade vapour compression system in the cabinet), and variation in payback versus engine size for the various configurations was presented for the supermarket considered.

Marimon et al. (2011) compared various configurations of ammonia / water absorption based CCHP systems in supermarket service, and found that all had payback periods of less than 6 years. As in other studies, the authors stressed the impact on such analyses of the price differential between electricity and gas and of energy subsidies.

Micro gas turbine (MGT) based trigeneration systems for supermarkets were modelled by Sugiartha et al. (2009), who found energy and emissions benefits compared with conventional supermarket systems. Operation in a full electrical output mode was found to be preferable to a heat-load following strategy. Payback periods were shown to reduce as the price differential between electricity and gas increased. For an MGT setup with an absorption COP of 0.5 operated on full electrical mode, a payback period of 5.7 years was found.

Several more recent studies have looked at integrating trigeneration with carbon dioxide (CO2) refrigeration systems. Suamir, Tassou and Marriott (2012) proposed using the cooling generated by the trigeneration system to condense the CO2 refrigerant in a cascade arrangement. This ensured that the CO2 refrigerant was maintained in subcritical conditions at all times. Using an electric to gas price ratio of 3.6, a payback period of just over 3 years was found compared to conventional systems, with energy savings of 30% and greenhouse gas emission savings of 43%. Options for such systems were further explored in Suamir and Tassou (2013). The alternative of using the absorption cooling for space cooling was found by Ge, Tassou and Suamir (2013) to also show promise in an MGT-based CHP plant integrated with CO2 refrigeration.

Availability and uptake of trigeneration systems has improved in recent years, but further development and experience in operation are still required before more widespread adoption.

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## Two stage compression

Operating over a large pressure ratio (high condensing temperature and low evaporating temperature) can require an intermediate pressure step to avoid discharge temperatures becoming too high. High discharge temperatures can cause both the refrigerant and oil to decompose.

Two-stage compression can be carried out by separate compressors, or by using a scroll compressor with an economiser (see economisers).

Inter-cooling between the stages allows the compressor discharge temperature to be reduced, also improving the system COP.

This can be done either by an intermediate receiver (flash vessel) or a heat exchanger or direct injection.

By separating flash gas from an intermediate receiver and injecting it between the compressor stages, the discharge temperature can be reduced. The liquid from the intermediate receiver that enters the evaporator is at a lower enthalpy and vapour quality than it would have been if expanded in one stage, therefore increasing the capacity of the compressor and also the COP. In addition, compression in the high-stage compressors occurs closer to the two-phase region with lower specific work required.

Alternatively, liquid from the intermediate pressure can be expanded through a heat exchanger which subcools the high pressure liquid from the condenser, therefore increasing the capacity of the compressor and also the COP. The expanded refrigerant from the heat exchanger is injected between the compressor stages, reducing discharge temperature.

A fraction of the refrigerant flow at the condenser outlet can be used to achieve major desuperheating between the compressor stages without any subcooling. Torella et al. (2010) conducted a second law analysis based on experimental data of a two-stage vapour compression (R404A) facility driven by a compound compressor for medium and low-capacity refrigeration applications (between –36ºC and –20ºC evaporating temperature). They showed that subcooling between the stages gave the best COP. Direct liquid injection between the stages gave a lower COP than with no arrangements at intermediate pressure. Depending on the refrigerant used, energy savings of the order of 10% might be achieved in conventional systems by subcooling (Kauffeld, 2016).

For supermarkets the main benefits of two-stage systems are for R744 (where most of the research has been concentrated). However, when working below evaporating temperatures of about -30ºC, discharge temperatures can also be high for R404A systems; therefore two stage systems have the same advantages as for R744 systems and are especially beneficial in ammonia systems due to the high discharge temperatures.

### R744

High discharge temperatures can be a problem with transcritical R744 cycles. For freezer racks, compressor exit temperatures of 120–150ºC will be encountered even under isentropic compression conditions (Srinivasan, 2011). Girotto, Minetto and Neksa (2004) reported discharge temperatures as high as 200ºC for single-stage compression, and with two-stage compression with inter-stage cooling discharge temperatures below 140°C (which is commonly considered an acceptable value for the CO2 compressor). According to Srinivasan (2011) the benefits that are derived as a result are 1) operating each stage with a high volumetric efficiency, 2) reduction of leakage across the piston rings, 3) increase in isentropic efficiency of each stage, 4) reduction of discharge gas temperature, 5) abatement of problems associated with lubricating oils and 6) reduction in compressor work. These benefits should be balanced against 1) an increase in the number of cylinders and the motors, and 2) the need for an additional heat exchanger for inter-stage gas cooling and associated increase in costs.

Much of the research is theoretical and is very dependent on refrigeration system design. Almeida and Barbosa (2011) conducted a theoretical analysis of a two-stage transcritical cooling cycle using R744 as a refrigerant. They predicted that performance of the two-stage cycle was superior to that of the single-stage system by 15%. When the intercooler operated at about the geometric mean of the evaporating and condensing pressures, compared to the single-stage cycle, the coefficient of performance (COP) of the two-stage cycle is improved. Huff, Hwang and Radermacher (2014) theoretically investigated three two-stage cycle options for the CO2 cycle. They claimed that a two–stage split cycle outperformed all other options and showed a 38-63% performance improvement over the basic single-stage cycle. Sawalha (2008) used a computer simulation model of a centralised CO2 transcritical system with accumulation tank. Using two-stage compression in the centralised system solution instead of single-stage resulted in a total COP that was about 5–22% higher than that of the reference centralised system.

Many supermarket R744 systems are booster systems (see Refrigerant – R744). Sawalha et al. (2008) showed that the two stage system can be applied to the high pressure circuit of the booster system, whilst the low pressure circuit remains unchanged.

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## Vacuum insulated panels (VIP)

VIPs consist of an open cell foam slab enclosed in a barrier film (Figure 11) (Brown, Evans, and Swain, 2007). A high vacuum is achieved within the enclosure, maintained by the impermeability of the barrier film and by the presence of a gas absorber (or getter) within the enclosure. The foam slab maintains the physical dimensions of the panel, supporting the barrier film, and reduces convection by the remaining gas molecules and the radiant heat transfer across the panel. The getter absorbs water vapour, atmospheric gasses and gasses emitted by the slab during the life of the panel to maintain the vacuum.

Current technology uses polyurethane foam insulation with a thermal conductivity of between 22 and 35 mW/m.K. VIPs typically have a thermal conductivity of around 3 mW/m.K measured at the centre of a panel. However, the film material does influence the conductivity of the panel as a whole and 5mW/m.K would be more typical when considering the complete panel. Figure 12 compares the thermal conductivity of complete VIP panels to a range of conventional insulations, with data taken from multiple sources to demonstrate a typical performance range for each (Kacimi and Labranque, 2011; Porextherm GmbH, 2009; ASHRAE, 2001; VensilResil Ltd., 2003; TAASI Corporation, 2012; Manin et al., 2003; Domínguez-Muñozet al., 2009; Engineering Toolbox, 2012; Bing, 2006; Nanopore inc., 2008; Nanopore inc., 2012).

Much of the published data (based on overlapping VIPs used to form an insulated box) shows VIPs only realising 2.5 to 3 times better insulation compared with PU (Polyurethane) rather than the expected fivefold benefit (Hammond and Micic, 2013). Hammond and Micic (2013) have shown that VIPs embedded into PU foamed walls can yield 86% of the expected benefit (using manufacturers’ thermal conductivity data); the remaining 14% is equivalent to ~2 mW/m.K variation in thermal conductivity of the PU and VIP and this is within claimed manufacturing tolerances.

Where research has shown VIPs to provide a 50 to 60% reduction in heat gains through the walls, the difference was usually associated with “edge effects” or “thermal bridges” (Brown, Evans, and Swain, 2007) but Kacimi and Labranque (2011) claimed that the metallisation layer on the VIPs was too thin for the thermal conductivity of the barrier film to cause any significant edge effects.

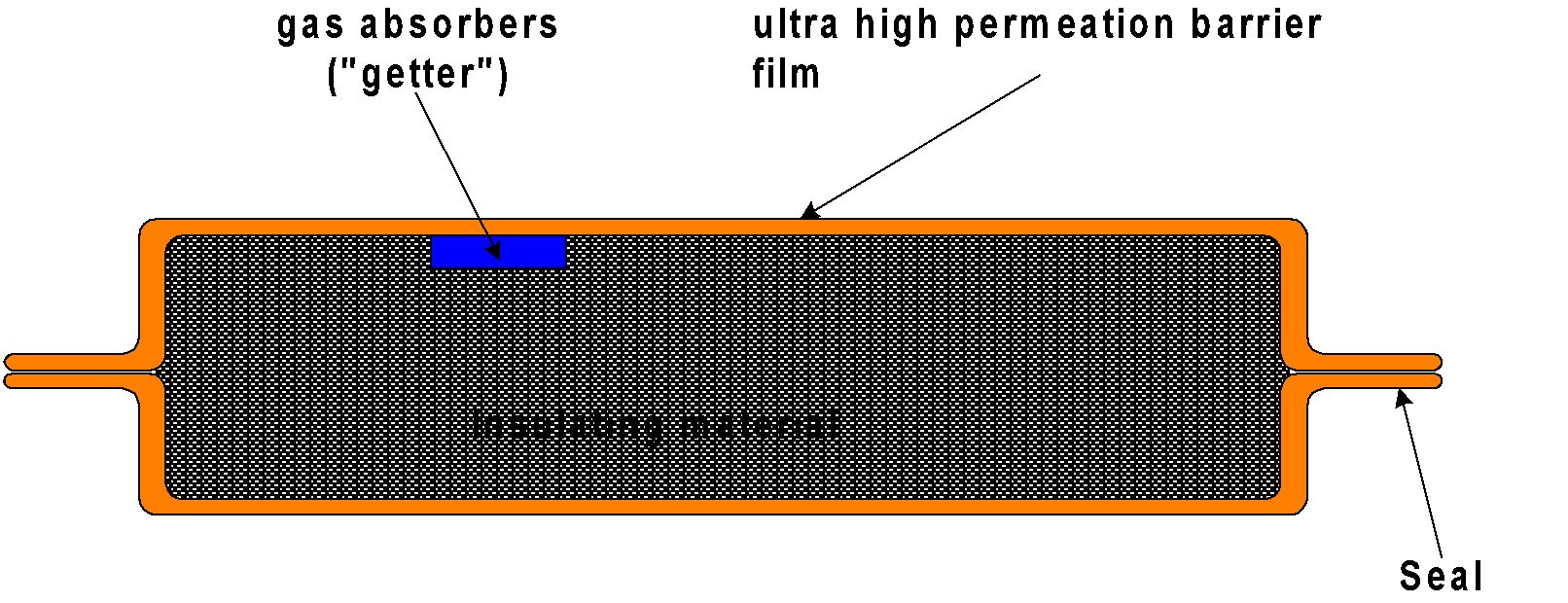


Figure . Schematic of a typical Vacuum Insulating Panel (VIP) (Swain and Brown, 2004).

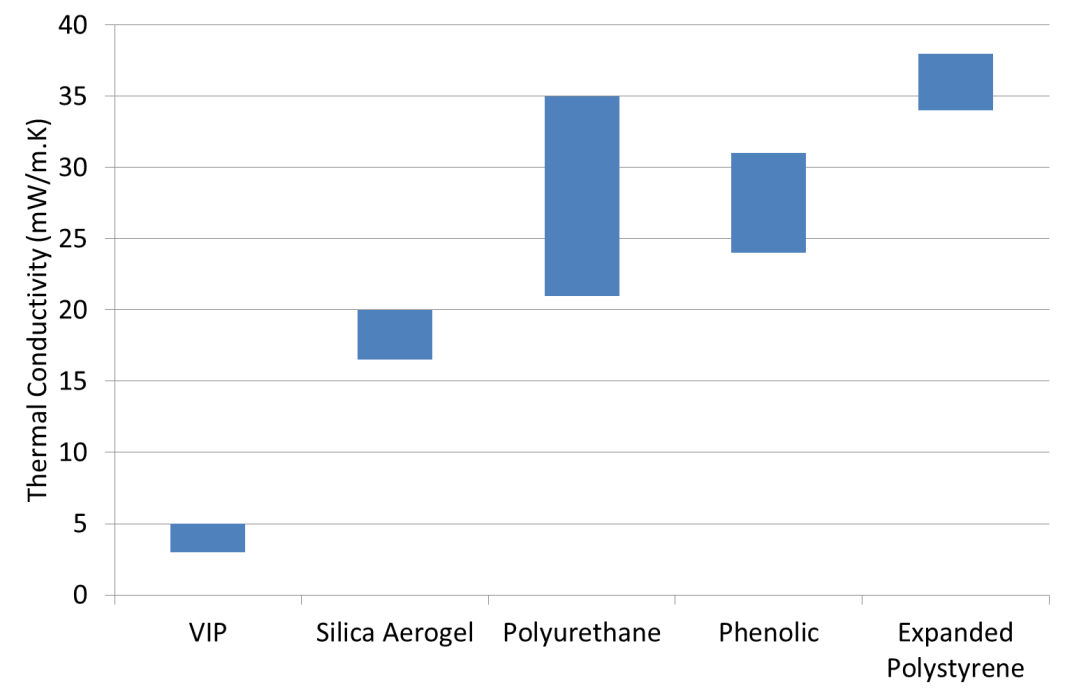


Figure . Ranges of typical thermal conductivities of conventional insulation materials and VIPs. Values based on material manufacturers’ datasheets and other technical manuals.

VIPs are currently integrated into production refrigerating (and other low temperature) appliances, where space and energy efficiency are both of high importance. Typically solid panels of VIP are secured against the outer wall of the insulation cavity before being foamed into position. This reduces heat gain without reducing internal volume. These panels would typically cover around 80% of the surface area of a cabinet wall but would not extend to the edges or corners of any individual wall. The PU foaming process is unaltered as a result of the addition of VIPs except for a reduction in the volume of foam to account for the VIPs.

The current cost of VIPs is more than that for PU foam; according to Alotaibi and Riffat (2013) €7.10 per m2 for PU foam and €168 per m2 for VIPs. However, the energy and/or space savings achievable can still make them an economic option. The main benefits of VIPs are the reduced thermal transmission for the same thickness, the reduced space taken by the insulation for the same thermal transmission or a combination of these two, especially where energy efficiency indexes are calculated based on internal volume and external dimensions are constrained. It is possible that weight of the materials used may also be important to some users, especially in transport applications.

In applications where existing insulation panels can be directly replaced by VIPs, there will be no significant change in fabrication cost. However, if in the application the existing foam insulation is an integral part of the structure, the insertion of VIPs is an extra step in the production process and may require strengthening of the structure in other ways, which will add to the product’s cost. In appliances where the VIP’s barrier film could be vulnerable to damage, there may be an additional cost for protection.

The thermal conductivity of VIPs is around one fifth of that of the polyurethane foam typically used. So for a given thickness of wall, the heat gain through the walls could be reduced by as much as 80%. However, polyurethane has two major benefits which are missing from VIPs; the low cost and the mechanical properties; PU foam can be used to add rigidity to a cabinet whereas VIPs must be protected from indentation, puncture or buckling to avoid damage to the foil coating. Furthermore any panel joints of poor integrity can quickly offset all gains.

Studies based on small thermal transport boxes which utilised slabs of PU foam or VIPs against each wall (Brown, Evans, and Swain, 2007) or in a sandwich layer (Kacimi and Labranque, 2011), both without a high integrity corner joint, only yielded half the improvement expected. Other studies (Hammond and Marques, 2014; Hammond and Micic, 2013) have shown that when used in combination with PU foam, VIPs can perform close to their expected thermal conductivity but only around 80% of the wall area was covered in these studies and panels were only 20 mm thick.

A study by Hammond and Marques (2014) analysed the benefit of embedding VIPs in the PU foam walls of a commercial service refrigerator and freezer product. The payback time for the VIPs would be 7.6 and 3.1 years for the fridge and freezer respectively. Currently 20 mm thick VIP panels are expected to cost £38 / sq.m (€46.11 / sq.m). The production costs of VIPs needs to fall below €25 per square metre (based on the 20 mm thick panel) before they become universally economical for freezer applications. For refrigerators, even at €25 per square metre the payback is only starting to become interesting on the walls adjoining heat sources, such as behind the condenser or compressor (Hammond and Evans, 2014).

For large scale applications like cold stores, energy savings could be made but unless space is of high value, the additional cost of the VIP is unlikely to be justified compared with the cheaper option of adding more PU foam.

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## Water loop systems (plus R1270)

Integral cabinets tend to have low refrigerant leakage compared to centralised plants. In addition, they can be operated safely and efficiently using hydrocarbon refrigerants which cannot be used in central plants because of charge size restrictions. Typical integral cabinets use air cooled condensers, rejecting the condenser heat to the shop floor. Using a large number of integral cabinets in a supermarket therefore results in a large heat gain to the store. The use of water cooled condensers overcomes this problem. Heat is rejected to a water loop system via a plate heat exchanger on the appliance, with the heat then removed from the water by externally located water chillers or dry coolers. In some cases some of the rejected heat is recovered to counteract the cooling effect of the open fronted cabinets.

In a trial in a Waitrose supermarket in the UK, integral cabinets operating on R1270 with water cooled condensers were compared over an 8 week period to a central plant direct expansion system operating on R404A. A reduction in CO2 emissions of 11.0% (based on a 2.8% reduction in electricity and a 57.5% reduction in gas) was measured (King et al., 2010). This was partly due to low condensing temperature and higher COP compared to a conventional direct expansion type supermarket refrigeration system. However, reported trials may not compare an optimised central plant system with an optimised water loop system. The savings claimed may therefore be exaggerated when comparing against an optimised central plant system. However, as many central plant systems are not optimised, the comparison is actually realistic.

King et al. (2010) benchmarked R404A data against R1270. In back-to-back testing there were consistent energy savings of 16% when using R1270 over R404A in the same application (water cooled integral cabinets). This applied to both high temperature and low temperature applications.

Pederson (2012) showed a 7.9% and 8.3% increase in COP for R1270 and R290 over R404A at an evaporating temperature of -10ºC and condensing temperature of 35ºC. The increase in COP of these refrigerants rises to a maximum of 10% at a condensing temperature of 45ºC. R1270 has a GWP of 1.8.

Direct savings are related to the refrigerant used in the integral cabinets and in the water chiller. If a hydrocarbon is used in both the integrals and the water chiller, the direct emissions could be significantly reduced. Stand-alone commercial applications have leakage rates of 1.5% per year (Defra, 2011) and so even if an HFC were used in the cabinets and the water chiller, the direct emissions would be reduced.

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# Application of refrigeration system technologies to a baseline store

## Review of technologies

The refrigeration system technologies reviewed were applied to a baseline store to ascertain their impact to save carbon. Each technology is reviewed and ranked according to the attributes listed in Table 11.

Table . Technology information.

|  |  |
| --- | --- |
| Quality of information | 5 independent peer review papers in general agreement = 5\*  3 independent peer review papers in general agreement =4\*  General agreement between independent reports or 1 peer reviewed publication=3\*  General agreement between web-based and sales literature =2\*  Personal communication only = 1\* |
| Barriers to staff/customers | H=major barrier  M=partial barrier  L=no barrier |
| Availability barriers | H=prototype/demonstrator only  M=limited availability  L=available |
| Limits to commercial maturity | H=lack of maturity  M=intermediate  L=mature |
| Ease of use and installation | H=major issues  M=some issues  L=simple |
| Technology dependence | H=high (i.e., interaction with another technology)  M=some  L=none |
| Maintainability issues | H=major issue  M=some problems  L=no issues |
| Legislative concerns | H=major (issue now)  M=could be an issue in near future  L=no impact |
| Scope of application | Range of applications |
| Energy savings (confidence) | % or actual savings:  High (H)=high confidence based on 4\* or 5\* papers, good agreement of quantified savings in literature  Medium (M)=3\* or fewer papers, lack of agreement in literature but some level of quantification of savings achieved  Low (L)=poor information in literature, little agreement in papers, savings not robustly quantified |
| Direct emissions (confidence) | % emissions from technology (Confidence: High, Medium, Low, H/M/L) |
| Payback time (years) | Time to recover cost of technology. This is equal to the saving in electrical energy per year divided by the cost of the technology. It does not include other ongoing costs, e.g. maintenance |

## Assumptions for the baseline store

An ASDA store at Weston-Super-Mare was used as a baseline store. Table 12 lists the store information. Table 13 shows the cabinet types and length of each type in the store. Figure 13 shows a schematic diagram of the refrigeration system layout. The store refrigeration load was split between low temperature (LT) and medium temperature (MT) packs and condensing units. The packs supplied the refrigerated cabinets and the condensing units supplied the cold rooms (which are outside the scope of this report). The LT cabinets were cooled by 2 packs and the MT cabinets were cooled by 4 packs. The refrigerant used for both MT and LT packs was R404A. The estimated energy used by each cabinet item is shown in Table 14.

Table . Store information

|  |  |
| --- | --- |
| **Store general** | |
| Opening hours | Monday 8 a.m. until Saturday 10 p.m.  Sunday 10 a.m. until 4 p.m. |
| Size (m2) | 6290 (74 x 85 m) (total store) |
| Store lighting | LED lights, Fluorescent tubes (T5 or T8), halogen spotlights, with the number of lights switched on reduced to a third, between 22:00 and 06:00. |
| Store temperature (°C), RH (%) | 19-24 (day) 17-24 (night), no humidity control |
| HVAC | Gas heating, DX air conditioning, air curtains |
| **Cabinets** | |
| Length of chilled cabinets (m) | 200 |
| Length of frozen cabinets (m) | 70 |
| Cabinet temperature control | Produce 8°C (diff = 2K)  Dairy 3°C (diff = 1K)  Meat 2°C (diff = 1K)  Frozen -18°C (diff = 3K)  Ice cream -21°C (diff = 2K) |
| Cabinet lighting | Fluorescent (originally T5), LED (upgraded later) |
| Controls for cabinet lighting | MT/HT cabinet lights on between 06:30 and 23:30  LT always on |
| Cabinet fan motors | EC fans |
| Anti-Sweat Heaters | Only on frozen cabinets, controlled by humidistat (fully on when humidity is > 60%, off when humidity is < 20%, modulates in between) |
| Shelf risers | Standard on chilled – however, not all cabinets have them fitted |
| Defrosts | Freezers 2/day  Chilled (off cycle) 4/day  Terminate on temp (max and min time) |
| Monitoring system | RDM |
| Cabinet loading | Some cabinets have packs blocking air return due to bad loading or customer interaction.  20-minute target for time between cold store and cabinet and 20 minutes between delivery vehicle and cold store |
| **Refrigeration plants** | |
| Type | Mix of packs and condenser units |
| Condensers | Air cooled |
| Condenser fan motors | Replacing SP (30%) with EC motors (70%) |
| Suction-liquid heat exchange | None |
| Floating head pressure control | Head pressure controlled to 10.5 barg (however, head pressure rises above this in summer)  Water spray used for 1-2 weeks / year when operating above design conditions |
| Suction pressure control | LT 0.7 barg  HT 3.5 barg  No floating suction control |
| Liquid pressure amplification | None |
| Pipe insulation | Insulated |
| Pressure drops | Minimised in design |
| Refrigerant | R404A (all refrigerated cabinet packs) |
| Refrigerant charge (kg) | R404A 1023 kg (cabinet packs and integrals)  R407C 35 kg (air conditioning)  R410A 11 kg (air conditioning)  R134a 3 kg (integral cabinets)  R600a 1 kg (professional cabinets)  Total 1073 kg |
| Refrigerant leakage (%/year) | 6% |
| Causes of leakage | Compressor change  Lots of pipework leaks  Leak on condensing unit  Liquid line fracture |

Table . Cabinet types and length of each cabinet in the store.

|  |  |  |
| --- | --- | --- |
|  | Cabinet types | Total length (m) |
| LT remote | FGD | 14.63 |
| HGD/well | 53.64 |
| **TOTAL** | **68.28** |
| MT remote | Roll-in | 48.77 |
| Multideck | 130.45 |
| **TOTAL** | **179.22** |
| LT integrals | HGD | 0.91 |
| MT integrals | Multideck | 24.87 |
| FGD | 5.00 |
| **TOTAL** | **35.66** |
| Hot food |  | 4.88 |
|  | **TOTAL ALL** | **283.16** |

Table . Energy used by cabinets split into component items.

|  |  |
| --- | --- |
| **Item** | **kW** |
| Compressor | 80.31 |
| Condenser fan | 12.04 |
| Evaporator fan | 4.42 |
| Defrost heater | 3.96 |
| Trim heater glass | 9.02 |
| Lights | 3.48 |
| **Total** | **113.23** |

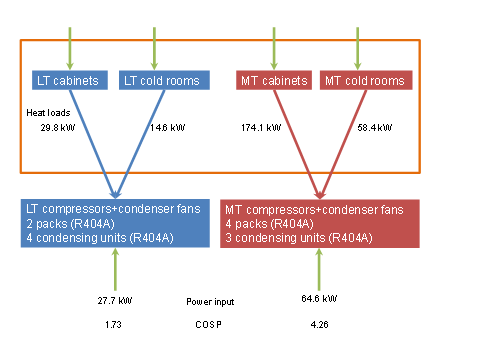


Figure . Schematic of refrigeration plant configuration at baseline store (showing compressor duties and input power).

## Development of the refrigeration system model

A model of the baseline supermarket was developed to determine the impact of new technologies on direct and indirect carbon emissions.

The following sections describe how the model was constructed and the assumptions applied.

### Indirect emissions

The yearly indirect CO2e emissions of the baseline store refrigeration system were calculated by multiplying the yearly energy consumption of the refrigeration system by a UK electricity CO2 conversion factor of 0.46219 (Defra, 2015)

The total energy consumption of the refrigeration systems in the baseline store was broken down into component parts, and the effect of the technologies evaluated on each of these component parts.

The refrigeration system was first divided based on cabinet type categories (in brackets the EN23953 cabinet classification) as shown below (and denoted *cat* in the following equation):

1. Remote chilled multideck (VC2)
2. Remote chilled roll-in (VC3)
3. Remote frozen HGD/well (VF1)
4. Remote FGD (VF4)
5. Integral chilled (VC2+HC1,4)
6. Integral FGD (VF2)
7. Professional (catering) cabinets

Each of the categories was then broken down into individual refrigeration components as shown below (and denoted *com* in the following equation):

1. Compressors
2. Condenser fans
3. Evaporator fans
4. Defrost heaters
5. Trim heaters
6. Lights

The total energy consumption of the refrigeration systems in the baseline store was:

Where:

*P* = total power of the baseline store

= total power for each cabinet category

*Pcom* = power of each of the components

*cat* (subscript) = component categories

The component powers are defined in the following sections.

### Remote cabinets

#### Compressors

The compressor power of each category of remote cabinet was calculated by taking the duty of the refrigerated cabinets in that category and dividing by the COP of the refrigeration compressor packs which fed those cabinets.

The duty of the cabinets supplied by the store refrigeration contractor was for EN23953 climate class 3 conditions (25°C, 60% RH). As the store operated at a lower temperature and humidity, the duty for each cabinet needed to be reduced to reflect real store conditions. Based on work by Mousset and Libsig (2011), the duty for each cabinet was reduced by 40% to reflect store conditions.

The design COPs of the LT and MT refrigeration systems were based on the COP for each refrigeration pack from manufacturers’ data at the store design conditions (condensing temperature of 40ºC). As the store design conditions were different from the real conditions (due to change in ambient temperature and therefore condensing temperature), the COPs were adjusted from the design COP using a coefficient. This coefficient was the ratio of the Carnot COP at design condensing temperature to real condensing temperature. The real condensing temperature was assumed to be the mean yearly ambient temperature in Birmingham plus 10ºC. The mean condensing temperature (23ºC) took into account the current refrigeration systems operation where condensing temperature was not allowed to reduce below 22°C.

#### Condenser fans

Based on design information from the baseline store, the condenser fan motor power was taken to be 3% of the heat rejected by the condenser.

#### Evaporator fans

Based on information provided by the refrigeration contractors responsible for maintenance at the baseline store, the evaporator fan motor powers were 60 W per 2.5 m of frozen cabinet and 38 W per 2.5 m of chilled cabinet.

#### Defrost heaters

Information provided by the refrigeration contractors responsible for maintenance at the baseline store indicated that all chilled cabinets operated using passive (off-cycle) defrosts. Frozen cabinets defrosted for 35 minutes every 12 hours. The power for defrosts per 2.5 m section of cabinet was 2.21 kW for FGD cabinets and 3.10 kW for HGD/well cabinets.

#### Trim heaters

The refrigeration contractors responsible for maintenance at the baseline store provided information showing that chilled cabinets did not have trim heaters. The frozen cabinets had 805 W per 2.5 m section of cabinet and the heaters operated for 40% of the time based on a humidistat control.

#### Lights

The refrigeration contractors responsible for maintenance at the baseline store provided information that showed lighting consumed 44 W per 2.5 m section of cabinet. The assumption was applied to both chilled and frozen cabinets.

### Integral cabinets

The total energy consumption of each of the integral cabinets was either taken from manufacturers’ specifications or estimated based on the category and size of the cabinet.

The proportion of power for each refrigeration component for the chilled VC2 and frozen FGD cabinets was considered the same as for the remote cabinets of the same category. The professional cabinets were all considered to be chilled (only 1 out of 10 was actually frozen). The proportion of power assigned to each component came from test data collected by the authors.

### Validation

The calculated total power of the refrigeration system for the store was compared with the total power of the 9 refrigeration electricity meters in the store. The total estimated power was 8.7% lower than the average electricity meter power over a year. It should be noted that it was not possible to be entirely sure what equipment was connected to each of the electricity meters, and therefore the refrigeration energy from the meters can only be considered an estimate.

### Direct emissions

The quantity and type of refrigerant in each of the remote packs and integral cabinets was provided by the refrigeration contractors responsible for the store maintenance.

The refrigeration systems were defined as:

1. LT remote packs
2. MT remote packs
3. LT integral cabinets
4. MT integral cabinets

The direct emissions were obtained by multiplying the mass of refrigerant in the system by the GWP of the refrigerant and the % leakage rate per year. The % leakage rates of the remote refrigeration plant (MT and LT) were considered as 6.1% per year. This was calculated by taking the mass of refrigerant charged (from the F-gas records) over a 20 month period and adjusting to a 12 month period and dividing this by the total charge of refrigerant in the store.

For the integral cabinets (MT and LT) the leakage rate was assumed to be 1.5% based on data from Defra (2011).

The total direct emissions of the baseline store, *DT* was

Where:

*Ds* = the direct emissions of each of the systems.

### Estimated benefits of technologies

Each of the technologies was assessed for its potential to save direct and indirect emissions from the refrigeration systems. Indirect savings were attributed to each cabinet component for each cabinet category. Savings in direct emissions were attributed to each refrigeration system (remote LT, MT and integral LT, MT).

From this, a set of coefficients was created. These coefficients were used as multipliers for each of the powers. A coefficient of 1 meant no savings and a coefficient of 0 meant 100% savings. For the indirect emissions, PT, the total power of the baseline store with the technology applied was defined by:

Where:

*CT* = the indirect emissions multiplying coefficient.

For the direct emissions the total direct emissions *DT*, of the baseline store with the technology was defined by:

Where:

*CD* was the direct emissions coefficient.

### Uptake of technologies

An assumption was made that technologies could be applied in the shortest reasonable time (i.e. technology can be ordered immediately and there are no delays in its supply). It should be noted that in reality supermarket funding cycles and planned routine updates to supermarkets will control uptake of the technologies.

***Presentation of CO2e savings***

The technologies were presented in graphs showing the CO2e saving potential and as bubble charts showing the carbon savings relative to the take up time and payback benefits of each technology (Figure 14). Where potential savings were varied, minimum and maximum savings for each technology were calculated.

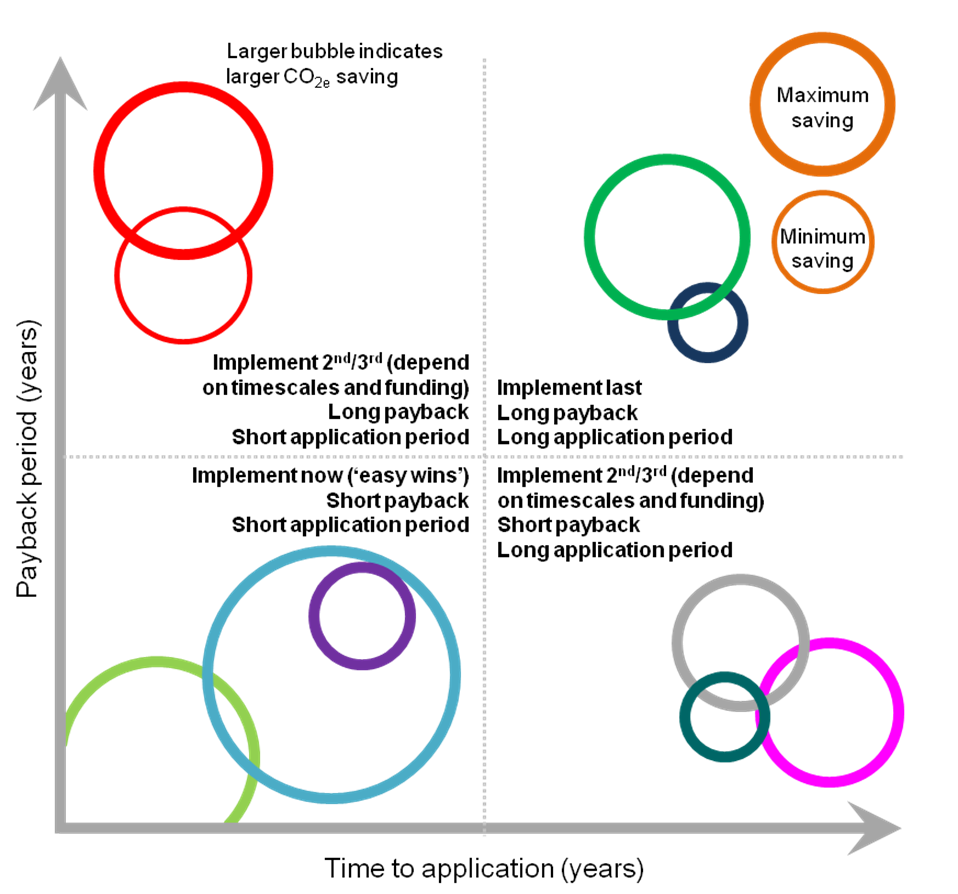


Figure . Bubble map schematic presentation.

**References**

Defra (2011) 2011 Guidelines to Defra / DECC’s GHG conversion factors for company reporting: methodology paper for emission factors. [Online] Available from www.defra.gov.uk [Accessed 14 January 2015].

Defra (2015). Government conversion factors for company reporting. http://www.ukconversionfactorscarbonsmart.co.uk/ [Accessed January 2016].

Mousset, S. and Libsig, M. Energy consumptions of display cabinets in supermarket. ICR 2011, August 21 -26 -Prague, Czech Republic.

# Application of technologies to the baseline supermarket

## Adiabatic condensers

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | Condensers |
| Energy savings (confidence) | 1.2% of total energy (M) |
| Direct emissions (confidence) | None (H) |
| Payback time (years) | > 10 years |

### Direct emission savings assumed

No direct emission savings are assumed.

### Estimated energy savings

The following assumptions are made:

* Air onto a condenser is maintained at 1.5ºC above the wet bulb temperature.
* The condensing temperature is assumed to be the air temperature onto the condenser plus 10ºC.
* The condensing temperature is not allowed to reduce below 22°C.

Using hourly weather data from Bristol for the year 2000, savings of 1.2% of the annual refrigeration energy were calculated. This was based on the improvement in Carnot COP and assumes an evaporating temperature of -20ºC. If floating head pressures were applied, the benefits would be larger. There would also be greater benefits with transcritical CO2 systems.

## Air deflectors/guides

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M (only 1 company supplying) |
| Ease of use and installation | M (important to install properly) |
| Technology independence | H (possibly not applicable with doors) |
| Maintainability | M (may become damaged by customers) |
| Legislative concerns | L |
| Scope of application | All open fronted cabinets |
| Energy savings (confidence) | 23% of compressor energy (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1 year |

### Energy savings assumed

It is assumed that 17% in energy savings (Foster, McAndrew and Evans, 2014) can be achieved on all open fronted cabinets. This equates to 23% of the compressor energy usage. It should be noted that the savings measured were all on integral cabinets, although it is expected that similar savings would be expected on remote cabinets.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Foster, A., McAndrew, P. and Evans, J. Novel aerofoils used for reducing energy consumption and improving temperature performance for multi-deck refrigerated display cabinets. 3rd IIR International Conference on Sustainability and the Cold Chain, London, 2014.

## Anti-fogging glass

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | L |
| Technology independence | M (replaces glass door heaters) |
| Maintainability | M (installation easy, durability poor) |
| Legislative concerns | L |
| Scope of application | Glass door cabinets |
| Energy savings (confidence) | 32-65% of glass heating (L) |
| Direct emissions (confidence) | None (H) |
| Payback time (years) | 1 year |

### Energy savings assumed

Savings have been assumed to apply only to frozen cabinets. Information on actual savings is limited and so the values of 32% to 65% savings on glass door heater energy from Anti-Fog Systems LLC and Dixell (Asia) Co., Ltd., 2010 have been used in calculations.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Anti-Fog Systems LLC. (2014). Refrigeration Applications. Retrieved 10 17, 2014, from http://antifogsystems.com/what-can-we-do-for-you/

Dixell (Asia) Co., Ltd. (2010). Case study of anti-fog film installation to prevent fogging on cabinet glass door.

## Anti-sweat heater control

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M (electrical installation and probes) |
| Technology independence | L (can be applied to any cabinet with anti-condensation heaters) |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | Freezer cabinets |
| Energy savings (confidence) | 0% (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1 year |

### Energy savings assumed

The baseline store had anti-sweat heaters controlled by a humidistat, therefore no savings to the baseline store are applied.

### Direct emission savings assumed

No direct emission savings are assumed.

## Boreholes and ground sink condensers

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | H |
| Technology independence | H |
| Maintainability | M |
| Legislative concerns | M (may be issues with ability to drill boreholes) |
| Scope of application | Only remote cabinets |
| Energy savings (confidence) | 20% (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 2-4 years using Geoscart model |

### Energy savings assumed

R744 systems showed energy savings of 24.6% (Leiper et al., 2014). Kauffeld (2016) suggests a 20% reduction in compressor power. As the baseline store is R404A and the climate of the UK is mild, a value of 20% has been assumed.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Kauffeld, M (2016). Current and future carbon-saving options for retail refrigeration (Chapter 7). In: Sustainable Retail Refrigeration, Ed: Evans, J.A and Foster, A.M. Wiley-Blackwell.

Leiper, A., Skelton, J., Rivers, N., and Zaynulin, D. (2014). Preventing transcritical operation of CO2 refrigeration systems with ground coupling. 3rd IIR International Conference on Sustainability and the Cold Chain. Twickenham, London: Institute of Refrigeration.

## Cabinet air flow

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | H (expertise needed to optimise the curtain) |
| Limits to commercial maturity | M |
| Ease of use and installation | H (unlikely to be a simple retrofit, specially redesigned parts required) |
| Technology independence | M (interaction with fans, doors etc.) |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | Open fronted chilled cabinets |
| Energy savings (confidence) | 3 to 13% of total cabinet energy (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1-2 years |

### Energy savings assumed

3-13% savings in total energy based on the work presented by Stribling et al. (1995 and 1995b) and Foster, Madge and Evans (2005). It has been assumed that the savings only apply to open fronted chilled cabinets as frozen cabinets have doors and so there is less benefit from optimisation of the air curtain (assuming an air curtain is applied). This data is quite old and would be expected to be relevant to changes to old cabinets.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Foster A.M., Madge M., Evans J.A. (2005).The use of CFD to improve the performance of a chilled multideck retail display cabinet. International Journal of Refrigeration 28 (2005) 698–705.

Stribling, D., Tassou, S. A., and Marriott, D. (1995). Optimisation of the Design of Refrigerated Display Cases Using Computational Fluid Dynamics. Proceedings of the Institute of Refrigeration (pp. 96: p 7.1 - 7.10). Institute of Refrigeration.

Stribling, D., Tassou, T., and Marriott, D. (1995 b). The use of CFD in the minimisation of air overspill from refrigerated display cases. Chartered Institute of Building Services Engineers.

## Cabinet lighting controls – dimming/switching using occupancy sensors

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L (if LED)  M (if fluorescent) |
| Technology independence | L |
| Maintainability | M (occupancy sensors could fail) |
| Legislative concerns | L |
| Scope of application | All retail cabinets |
| Energy savings (confidence) | 31 to 43% of lighting (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 3-4 years |

### Energy savings assumed

Reduction of 31 to 43% of lighting load based on Richman and Tuenge (2009) and Diebel, Mort, Thomas, and Park (2013). This assumes that the cabinet lights are not switched off overnight (i.e. 24 hour opening) as in the baseline store. In stores where lights are turned off overnight the savings would be reduced by approximately half.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Diebel, D., Mort, A., Thomas, A., and Park, S. Energy savings from led refrigeration display case lighting. Retrieved 2013, from ADM Energy Research Association: http://admenergy.com/wp-content/uploads/2012/04/039.pdf

Richman, E. E. and Tuenge, J. R. (2009). Demonstration Assessment of Light Emitting Diode (LED) Freezer Case Lighting in Albertsons Grocery in Eugene, OR. Pacific Northwest National Laboratory. Springfield, VA: National Technical Information Service, U.S. Department of Energy.

## Cabinet selection

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | M |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | All refrigerated cabinets |
| Energy savings (confidence) | 10-20% of total (H) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | <1 month if correct cabinets are selected |

### Energy savings assumed

Savings of 10-20% on refrigeration energy based on Evans, Scarcelli, and Swain (2007).

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Evans, J. A., Scarcelli, S., and Swain, M. V. (2007). Temperature and energy performance of refrigerated retail display and commercial catering cabinets under test conditions. International Journal of Refrigeration, 30(3), 398-408.

## Centralised air distribution

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L (assuming ducting does not cause a barrier) |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | H |
| Technology independence | M |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | All chilled cabinets (remotes only) |
| Energy savings (confidence) | 11% (L) |
| Direct emissions (confidence) | 40% reduction (M) |
| Payback time (years) | 4 years |

### Energy savings assumed

Total savings of 11% have been calculated on the remote chilled cabinets based on data provided by Asda.

### Direct emission savings assumed

Savings of 40% are assumed based on a reduction in charge of 40% provided by Asda.

## DC electronically commutated (EC) permanent magnet motors for condenser fans

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L (retrofit available) |
| Technology independence | M |
| Maintainability | L (longer life) |
| Legislative concerns | L |
| Scope of application | All refrigeration systems |
| Energy savings (confidence) | 15% of all cabinet fan power (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 6-7 months on refit, i.e. when fans already due to be changed. |

### Energy savings assumed

Maximum savings of 50% on fan energy, assuming comparison with shaded pole motor. The baseline store was considered to have 70% EC motors and 30% SP motors, therefore providing a 15% reduction in energy. This was considered applicable to both remote and integral condensers.

### Direct emission savings assumed

No direct emission savings are assumed.

## DC electronically commutated (EC) permanent magnet motors for evaporator fans

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | H |
| Ease of use and installation | L (complete retrofits available) |
| Technology independence | L |
| Maintainability | L (longer life) |
| Legislative concerns | L |
| Scope of application | All cabinets |
| Energy savings (confidence) | 0% as already applied to base store (H) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

The baseline store has EC fans fitted on all cabinets and therefore no energy savings are assumed.

### Direct emission savings assumed

No direct emission savings are assumed.

## Defrost drain traps

### Energy savings assumed

The baseline store has drain traps fitted on all cabinets and therefore no energy savings are assumed.

### Direct emission savings assumed

No direct emission savings are assumed.

## Defrosts

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* (however, varies with different technologies) |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M (instrumentation required) |
| Technology independence | M |
| Maintainability | M/H (depends on technology) |
| Legislative concerns | L |
| Scope of application | All freezers |
| Energy savings (confidence) | Hot/cool gas (remote only): 33% reduction in defrost heat load (L)  Reverse cycle (integral only): 33% reduction in defrost heat load (L)  Warm liquid: 33% reduction in defrost heat load (L)  Heat bank: 48% in defrost energy compared with reverse cycle defrost (L)  Thermosiphon: 80% of defrost power plus 10% of compressor power (M)  Defrost on demand: 0 to 72.5% of defrost energy (M)  Ultrasonic defrost: no information (L) |
| Direct emissions (confidence) | Hot/cool gas: 5% increase in refrigerant leakage (M)  Reverse cycle: 5% increase in refrigerant leakage (M)  Warm liquid: 0% (M)  Heat bank: 0% (L)  Thermosiphon : 0% (L)  Defrost on demand: 0% (H)  Ultrasonic defrost: 0% (L) |
| Payback time (years) | Varied, at best 1 year |

### Energy savings assumed

Hot/cool gas: Based on an increase in defrost efficiency of 5% (from 15 to 20% efficient) cited in Cole (1989) and Lawrence and Evans (2008), the energy used in the defrost would reduce by 33%. Assumed to be applicable only to remote cabinets. An assumption is made that head pressures are not increased to enable the hot gas defrost to operate successfully.

Reverse cycle: No information is available on energy savings. It is assumed that the savings are similar to those achieved with a gas defrost. Assumed to be applicable only to integral cabinets.

Warm liquid: No information is available on energy savings. It is assumed that the savings are similar to those achieved with a gas defrost. Assumed to be applicable to all frozen cabinets.

Heat bank: 48% savings in defrost energy based on information from Dong et al. (2011). However, this is based on heat pumps and not display cabinets and so it should be noted that the results may not be transferable.

Thermosiphon: Based on work by Foster et al. (2015), 80% of defrost power can be saved and 10% of compressor power.

Defrost on demand: Information on defrost on demand systems indicate a large range in the potential savings; from zero up to 72.5% (Lawrence and Evans, 2008) of defrost energy.

Ultrasonic defrost: Insufficient information is available to suggest energy savings.

### Direct emission savings assumed

Hot/cool gas: Increase in refrigerant loss due to additional piping and valves (California Utilities Statewide Codes and Standards Team, 2013).

Reverse cycle: Increase in refrigerant loss due to additional piping and valves (California Utilities Statewide Codes and Standards Team (2013).

Warm liquid: No direct emission savings are assumed.

Heat bank: No direct emission savings are assumed. Potentially, the additional valves and the additional heat exchanger may increase leakage. This has been ignored in calculations. The technology is assumed to be applicable to both remote and integral cabinets.

Thermosiphon: No direct emission savings are assumed. Potentially, the additional valves and the additional heat exchanger may increase leakage. This has been ignored in calculations.

Defrost on demand: No direct emission savings are assumed. Technology has been applied to all frozen cabinets in the baseline store.

Ultrasonic defrost: No direct emission savings are assumed.

**References:**

California Utilities Statewide Codes and Standards Team. (2013). Codes and standards enhancement initiative (case) - Supermarket Refrigeration- California Building Energy Efficiency Standards. Sacramento: California Statewide Utility Codes and Standards Program.

Cole, R. A. (1989). Refrigeration Loads in a Freezer Due to Hot Gas Defrosts and their Associated Costs. ASHRAE Transactions, 95(2).

Dong, J.-K., Jiang, Y.-Q., Yao, Y., and Zhang, X.-D. (2011). Operating performance of novel reverse-cycle defrosting method based on thermal energy storage for air source heat pump. J. Cent. South Univ. Technol., 18, 2163.

Foster A., Campbell R., Davies T., Evans J. (2015). A novel passive defrost system for a frozen retail display cabinet with a low evaporator. The 24th IIR International Congress of Refrigeration, 2015, Yokohama, Japan.

Lawrence, J. M., and Evans, J. A. (2008). Refrigerant flow instability as a means to predict the need for defrosting the evaporator in a retail display freezer cabinet. International Journal of Refrigeration, 107-112.

## Diagonal compact fans

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | L |
| Maintainability | L (no greater than other fans) |
| Legislative concerns | L |
| Scope of application | All cabinets with convective flow |
| Energy savings (confidence) | Unknown (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

No information is available to justify savings that may be achieved.

### Direct emission savings assumed

No direct emission savings are assumed.

## Distributed refrigeration system

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | M |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | Remote refrigeration |
| Energy savings (confidence) | No information to justify that energy is saved (H) |
| Direct emissions (confidence) | 0% reduction (H) |
| Payback time (years) | None |

### Energy savings assumed

No information is available to justify savings that may be achieved. The baseline store already has six distributed packs.

### Direct emission savings assumed

No direct emission savings are assumed as baseline store already has six distributed packs. However, it should be noted that this technology may be demonstrated to be more beneficial in a different store configuration.

## Doors on cabinets

### Summary

|  |  |
| --- | --- |
| Quality of information | 5\* |
| Barriers to staff/customers | H |
| Availability barriers | L |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M (need for additional lighting) |
| Maintainability | M (supermarkets report frequent failures) |
| Legislative concerns | L |
| Scope of application | All open cabinets |
| Energy savings (confidence) | 18-51% of total - depends on usage (H) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1.4-1.6 years |

### Energy savings assumed

According to Fricke and Becker (2010a) the increase in lights and anti-sweat heat can counterbalance much of the reduction in compressor power. For the retrofit of doors it is quite likely that fan power would not be changed and for non-humid stores anti-sweat heaters would not be required.

Assuming a worst-case energy consumption which matches exactly that of Fricke and Becker (2010a) gives an overall energy saving of 18% for the chilled cabinets. As a best case, it is assumed that light power is not increased and anti-sweat heat is not added. This gives an overall energy saving of 51% for the chilled cabinets

This data applies to open fronted chilled cabinets only as all freezers in the baseline store already have doors.

### Direct emission savings assumed

No direct emission savings are assumed. However, potentially, if the refrigeration duty is reduced, there may be opportunities to reduce the size of heat exchangers. This has not been included in the calculations.

**Reference**

Fricke, B., and Becker, B. (2010a). Doored Display Cases: They Save Energy, Don't Lose Sales. ASHRAE Journal, 18-26.

## Dual port TEV

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | Unknown |
| Legislative concerns | L |
| Scope of application | All cabinets, primarily remote |
| Energy savings (confidence) | Unable to quantify (n/a) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | Unable to quantify (n/a) |

### Energy savings assumed

No information is available to justify savings that may be achieved. Potentially, there may be energy savings through increasing the evaporating temperature but there are no published studies to justify this.

### Direct emission savings assumed

No direct emission savings are assumed.

## Dynamic demand

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | No information |
| Legislative concerns | None (assuming food maintained at correct temperature) |
| Scope of application | All cabinets |
| Energy savings (confidence) | 0% (H) |
| Indirect emissions (confidence) | Unable to quantify (n/a) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | Unable to quantify (n/a) |

### Energy/carbon savings assumed

The technology does not save energy but can save carbon (and money) by shifting the energy used to lower carbon (and cost) sources. It is likely that only the defrost energy (freezers only in baseline store) could be moved to another time and therefore the carbon saving would be related only to defrosting of cabinets.

Using information from Earth Energy (Earth, 2016), the average/mean grid generation carbon intensity (ignoring transmission/distribution losses) is approximately 302 gCO2/kWh. The minimum hourly grid generation carbon intensity (ignoring transmission/distribution losses) is approximately 231g CO2/kWh. Therefore, potentially, carbon savings of 23.5% could be achieved by shifting from mean to minimum grid generation intensity.

The baseline store already shifts defrosts to outside of peak energy tariff rates.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Earth Notes (accessed Jan 2016). http://www.earth.org.uk/\_gridCarbonIntensityGB.html

## Economisers

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | M (requires compressor with economiser port) |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | Remote plant |
| Energy savings (confidence) | 4% increase in COP for MT packs and 7% for LT packs (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1 year |

### Energy savings assumed

At an evaporating temperature of -35ºC and condensing temperature of 25ºC for R404A, the pressure ratio is 7:1. Therefore the 16% improvement in COP at a pressure ratio of 12 stated by Jousson (1988) is too high. A 4% improvement in COP for the remote chillers and 7% for the remote freezers is therefore assumed, as shown by Bellstedt (2015).

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Bellstedt, M. (2015). Refrigeration Plant Energy Efficiency. How to minimize running costs! Minus40 Pty. Ltd Sydney. National Meat Industry Training Advisory Council Limited. http://www.mintrac.net.au/docs/pdf/20120314-N-E-MB.pdf (accessed 2015).

Jousson, S. (1988). Performance Simulations of Twin Screw Compressors with Economizer. International compressor Engineering Conference. Purdue University.

## Ejectors

### Summary

|  |  |
| --- | --- |
| Quality of information | 5\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | Better for CO2 |
| Maintainability | Unknown |
| Legislative concerns | None |
| Scope of application | Remote plant (specifically CO2) |
| Energy savings (confidence) | 10 to 20% improvement in COP (for R744) (H) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | <1 year |

### Energy savings assumed

A number of papers claim energy savings through use of an ejector for CO2 transcritical systems. Most claim savings of 10-20% (Ersoy and Sag, 2014, Frigo-Consulting Ltd, 2014, Hafner et al., 2014, Kornhauser, 1990, Li and Gross, 2004, Nekså et al., 2010, Pottker, 2012).

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Ersoy, H. K., and Sag, N. B. (2014). Preliminary experimental results on the R134a refrigeration system using a two-phase ejector as an expander. International Journal of Refrigeration, 43, 97-110.

Frigo-Consulting Ltd. (2014). First Ejector Is Operating Successfully. Retrieved from http://www.frigoconsulting.ch/en/news/ejektor.html

Hafner, A., Försterling, S., and Banasiak, K. (2014). Multi-ejector concept for R-744 supermarket refrigeration. International Journal of Refrigeration, 43, 1-13.

Kornhauser, A. (1990). The use of an ejector as refrigerant expander. Proceeding of the 1990 USNC/IIR-Purdue refrigeration Conference. Purdue.

Li, D., and Groll, E. A. (2004). Transcritical CO2 Refrigeration cycle with ejector expansion device. International Refrigeration and Air Conditioning Conference at Purdue. Purdue.

Nekså, P., Walnum, H. T., and Hafner, A. (2010). CO2: A refrigerant of the past with prospects of being one of the main refrigerants of the future. 9th IIR Gustav Lorentzen Conference. Sydney.

Pottker, G. (2012). Potentials for COP Increase in Vapour Compression Systems. Urbana-Champaign: University of Illinois.

## Electronic expansion valves

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | M (cost) |
| Limits to commercial maturity | L (mature product) |
| Ease of use and installation | M (easy at installation stage, lengthy retrofit in store) |
| Technology independence | M (interaction with floating head pressure controls) |
| Maintainability | L (some improvements reported versus TEV) |
| Legislative concerns | L (none) |
| Scope of application | All direct expansion systems (most cabinets) |
| Energy savings (confidence) | 0% (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

Published work (Lazzarin et al., 2009 and Carel, 2006) typically suggests savings in energy of 15-22% can be achieved. The baseline store already has EEVs fitted and therefore no savings are assumed.

It is not obvious why changing from a TEV to an EEV saves energy. However, controlling the superheat better may allow for higher evaporating temperature and EEVs allow the possibility of reducing condensing pressure during cold ambient conditions. For those reasons, no energy saving for this technology on its own is assumed. It should, however, be considered with other technologies, e.g. floating head pressure.

### Direct emission savings assumed

No direct emission savings are assumed.

## Expansion machines (e.g. turbines, not including vortex tubes)

Summary:

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | H |
| Limits to commercial maturity | H |
| Ease of use and installation | Not known |
| Technology independence | Larger benefit with higher pressure differential e.g. CO2 and gas cycles e.g. air/nitrogen. |
| Maintainability | Not known |
| Legislative concerns | None |
| Scope of application | Remote plant |
| Energy savings (confidence) | 6-15% reduction in compressor power (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | Limited information <1 year anticipated |

### Energy savings assumed

Greater savings are achieved at higher pressure ratios and so the technology would save more energy with a refrigerant such as R744. For the baseline store (which uses R404A) the savings are anticipated to be 6-15% of compressor energy, based on the work by Brasz (1995). It is assumed the technology only applies to remote cabinets.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Brasz, J. J. (1995). Patent No. EP0676600 B1. Europe.

## Fan motor outside of cabinet

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | L |
| Availability barriers | H |
| Limits to commercial maturity | H |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | L |
| Legislative concerns | None |
| Scope of application | All cabinets |
| Energy savings (confidence) | 1.5-5.5% of compressor load (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 2-3 years |

### Energy savings assumed

All the fan motor energy is assumed to enter the baselines cabinets as heat. It is assumed that all of this heat is removed from the cabinet with this technology. The savings are calculated by dividing the heat load of the fan by the COP of the refrigeration compressor. A COP for chilled cabinets of 4.5 and of 1.7 for freezer cabinets (as in the baseline store) was used. Depending on the cabinet type, this results in savings in compressor power of 1.5 to 5.5%.

### Direct emission savings assumed

No direct emission savings are assumed.

## Flooded evaporators (added to R744)

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | L |
| Legislative concerns | More refrigerant required |
| Scope of application | Remote |
| Energy savings (confidence) | 5.5 to 40% for compressor energy for R744 (L) |
| Direct emissions (confidence) | 0 (H) |
| Payback time (years) | No information available |

### Energy savings assumed

Evidence of energy savings is limited to the work of Johansen (2006) which indicated savings in compressor energy of 30-40% and Reulens (2009) which showed increase in COP of 5.5% at low ambient temperature. Minimum savings of 5.5% and maximum of 40% have therefore been applied to remote cabinets operating on R744.

### Direct emission savings assumed

No direct emission savings are assumed. Logically a larger refrigerant charge will increase the amount of refrigerant leaked but for a R744 system this is has very low impact on CO2 emissions.

**References:**

Johansen, E. (2006). Energiriktige kuldeleveranser til dagligvarehandelen. FOKU.

Reulens, W. (2009). Natural refrigerant CO2. Diepenbeek: Katholieke Hogeschool Limburg. Diepenbeek C.

## Heat exchanger design

### Summary

|  |  |
| --- | --- |
| Quality of information | 5\* |
| Barriers to staff/customers | L |
| Availability barriers | M to H |
| Limits to commercial maturity | M to H |
| Ease of use and installation | M to H |
| Technology independence | M to H |
| Maintainability | M to H |
| Legislative concerns | L |
| Scope of application | All refrigerated cabinets |
| Energy savings (confidence) | Evaporator optimisation: 28% reduction of compressor power (L)  Micro-channel heat exchangers: 10% reduction of compressor power (L)  Heat exchange rifling: no information (n/a)  Enhanced internal heat transfer (micro-fins): no information (n/a)  Evaporative condensers: 0% reduction of compressor power (M) |
| Direct emissions (confidence) | Micro-channel heat exchangers: 30% (L)  All other technologies 0% (M) |
| Payback time (years) | 1-2 years, dependent on technology |

### Energy savings assumed

Evaporator optimisation: a reduction of compressor power has been assumed based on data from Chandrasekharan and Bullard (2004b).

Micro-channel heat exchangers: according to Danfoss (2015) micro-channel heat exchangers improve COP by around 10%. It is therefore assumed that compressor power will reduce by this amount.

Heat exchange rifling: there was insufficient data to calculate any savings.

Enhanced internal heat transfer (micro-fins): there was insufficient data to calculate any savings

Evaporative condensers: Clark and Gilles (2014) found that in the UK air-cooled condensers are more expensive both to install and operate than air cooled-alternatives below approximately 500 kW duty. As the refrigeration duty of the cold store is significantly lower than this value, there is assumed to be no benefit in evaporative condensers.

### Direct emission savings assumed

According to Danfoss (2015) micro-channel heat exchangers require 30% less refrigerant. It is therefore assumed that the direct emissions are reduced by this amount.

No direct emission savings are assumed for each of the other technologies. Any improvement in the effectiveness of heat exchangers should make them smaller, reducing the overall refrigerant charge.

**References:**

Chandrasekharan, R., and Bullard, C. (2004b). Design of energy-efficient display case evaporators. International Refrigeration and Air Conditioning Conference. Purdue.

Clark, J. and Gillies, A. Comparison of evaporative and air cooled condensers in industrial applications. Proc. Inst. R. 2014-15. 3-1.

Danfoss (2015). Retrieved 2015, from The Future Belongs to Micro Channel Heat Exchangers - Superior coil technology: http://www.danfoss.com/United\_Kingdom/NewsAndEvents/Archive/Refrigeration+News/Micro-Channel-Heat-Exchangers/6DB6E8EE-5D82-4B47-BCB1-6DA518910AB8.html.

## Heat from light outside cabinet

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | M |
| Availability barriers | H |
| Limits to commercial maturity | H |
| Ease of use and installation | Unknown |
| Technology independence | M |
| Maintainability | Unknown |
| Legislative concerns | None |
| Scope of application | Currently for cold rooms, potential for cabinets |
| Energy savings (confidence) | 1 to 3% of compressor load (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | Not yet quantified but anticipated to be around 2-3 years |

### Energy savings assumed

Energy savings have been based on the assumption that all the lighting energy is removed from the compressor duty. This was calculated by assuming a COP of 4.3 for chilled cabinets and of 1.6 for freezer cabinets (as in the baseline store). This results in savings in compressor power of 1 to 3% for chilled and frozen cabinets respectively.

### Direct emission savings assumed

No direct emission savings are assumed.

## Heat pipes

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | H |
| Limits to commercial maturity | H |
| Ease of use and installation | H |
| Technology independence | Depends on cabinet type |
| Maintainability | M (potentially gets in the way of cleaning and maintenance., could become damaged) |
| Legislative concerns | None |
| Scope of application | All cabinets. |
| Energy savings (confidence) | 1-3.5% of compressor energy (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 42 years |

### Energy savings assumed

Foster, Orlandi, and Evans (2014) showed savings of between 1 and 3.5% on compressor energy assuming that the cabinet set-point was increased to make use of the lower maximum temperature.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Foster, A. M., Orlandi, M., and Evans, J. A. (2014). Use of Heat Pipes to Improve Temperature Performance of a Chilled Refrigerated Display Cabinet. Proc. of 3d IIR International Conference on the Cold Chain and Sustainability. Twickenham, UK.

## Hydrophobic coating on evaporator

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | H |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | All frozen cabinets |
| Energy savings (confidence) | 10.8% less defrost energy (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 3-4 years |

### Energy savings assumed

Jhee, Lee and Kim (2002) found the defrosting efficiencies of hydrophobic heat exchangers were enhanced by about 10.8% whereas hydrophilic surface-treated heat exchangers were enhanced by about 3.5%. Wu and Webb (2001) showed that hydrophilic coatings should not be applied to the surfaces of evaporators that experience freezing conditions. Conversely, Shin et al. (2000) studied fin plates used in refrigerator evaporators. They found that the surface with better hydrophilicity provided a lower water holdup and, in turn, a more highly efficient defrost cycle.

Based on these results, a reduction in defrost energy of 10.8% is assumed using a hydrophobic treatment. However, these results are based on a small amount of conflicting information.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Jhee, S., Lee, K-S, and Kim, W-S. Effect of surface treatments on the frosting/defrosting behavior of a fin-tube heat exchanger, International Journal of Refrigeration, Volume 25, Issue 8, December 2002, Pages 1047-1053.

Shin, J., Kim, C., Ha, S., and Kim, J. (2000). A Study of Water Holdup on Two Surfaces with Different Hydrocharacteristics. Journal of Flow Visualization and Image Processing, 7(4), 343-351.

Wu, X M, and Webb, R. L. Investigation of the possibility of frost release from a cold surface. Experimental Thermal and Fluid Science 24, no. 3-4 (2001): 151-156.

## Improved axial fans

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | L |
| Maintainability | L (no greater than other fans) |
| Legislative concerns | L |
| Scope of application | All cabinets |
| Energy savings (confidence) | 20% of evaporator and condenser fan motor power (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1 year |

### Energy savings assumed

Assuming evaporator and condenser blades currently have an efficiency of 60% and that the efficiency was increased to 75%, this gives a fan motor power reduction of 20%, assuming that the fan motor power was reduced to maintain the same flow rate.

### Direct emission savings assumed

No direct emission savings are assumed.

## Improved cabinet loading

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | M (depends on solution adopted) |
| Availability barriers | H (cabinet concepts which are more tolerant of loading are not available as production cabinets) |
| Limits to commercial maturity | H (much work still to be done) |
| Ease of use and installation | L |
| Technology independence | L/M (depending on technology used) |
| Maintainability | L/M (depending on technology used) |
| Legislative concerns | L |
| Scope of application | Any open fronted cabinet |
| Energy savings (confidence) | No data available (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | Unable to quantify (n/a) |

### Energy savings assumed

No information is available on cabinet loading in store conditions. In test room conditions, savings in compressor power of 60% have been achieved (see earlier chapter). However, whether these translate into store conditions is unclear and would seem to be too high to apply to a whole supermarket. Insufficient evidence is available to quantify the potential savings in the baseline supermarket.

### Direct emission savings assumed

No direct emission savings are assumed.

## Improved cabinet location

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | H (moving cabinet may be detrimental to marketing of products) |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L (cabinet will work better away from draughts) |
| Legislative concerns | L |
| Scope of application | All cabinets, especially open fronted |
| Energy savings (confidence) | No evidence (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

No information is available on the effect of cabinet location in store conditions. In test room conditions, increasing air flow over the front of open fronted cabinets has been shown to increase heat transfer by up to 53% (Gaspar, Gonçalves, and Pitarma, 2011). This translates to 42% compressor savings (based on an 80% infiltration rate). This cannot practically be applied to all cabinets in the store. No evidence exists to make an accurate assessment of how many cabinets in the baseline store could be moved to a more appropriate location.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Gaspar, P. D., Gonçalves, L. C., and Pitarma, R. A. (2011). Experimental analysis of the thermal entrainment factor of air curtains in vertical open display cabinets for different ambient air conditions. Applied Thermal Engineering, 31(5), 961-969.

## Improved cabinet set-points

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | M (health and safety of food) |
| Scope of application | All retail cabinets |
| Energy savings (confidence) | 0% (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

Evidence suggests that temperatures are generally higher than recommended and so there appears little opportunity to increase set-point temperatures.

### Direct emission savings assumed

No direct emission savings are assumed.

## Improved Glazing

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | M |
| Availability barriers | L |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | Glass door cabinets |
| Energy savings (confidence) | 10% of chiller, 5% of freezer compressor power (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1-2 years |

### Energy savings assumed

It is assumed that the glazing improvements are additional to current glazing of cabinet doors. The data from Pilkington indicates that moving from single to double glazing on chillers would reduce conduction through the glass by 42%. For freezers, moving from double to triple glazing would result in savings in conduction of 39%.

It is calculated that the heat load through chilled cabinet glazing is 120 W/m2 for single and 70 W/m2 for double glazing. Moving from single to double glazing on a chiller would reduce the heat load on the cabinet by 50 W/m2 (assuming a 25°C temperature difference between the cabinet and ambient conditions). If a chilled cabinet has a heat load of 500 W, this then reduces the overall compressor load by 10%.

Moving from double to triple glazing on a freezer would reduce the heat load on the cabinet by 50 W (assuming a 45°C temperature difference between the cabinet and ambient air/environment which results in 126 W/m2 for double and 76 W/m2 for triple glazing). If a frozen cabinet has a heat load of 1000 W, this then reduces the overall compressor load by 5%.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference**

Pilkington (2001). Building Regulations Part L. http://www.pilkington.com/resources/brpartlbulletin4englandandwales.pdf

## Internet shopping

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | No information |
| Availability barriers | No information |
| Limits to commercial maturity | M |
| Ease of use and installation | H |
| Technology independence | No information |
| Maintainability | No information |
| Legislative concerns | L |
| Scope of application | n/a |
| Energy savings (confidence) | 0-100 % (H) |
| Direct emissions (confidence) | 0-100% (H) |
| Payback time (years) | Varied depending on assumptions made and technologies used |

**Energy/indirect and direct emissions savings assumed**

The information given in the sections above is mainly concerned with the emissions of transportation related to e-commerce compared to buying directly from a supermarket. There is no available data for the difference in emissions from refrigeration for internet shopping compared with supermarket shopping. If the food is removed directly from the stores’ retail cabinets for the internet shopping, there would be no benefit. However, if the food is taken from a cold store and does not enter the retail cabinets, significant emissions benefits would be expected. It has been assumed that minimum savings are associated with food removal from stores (i.e. no savings) and maximum savings are associated with the store being replaced by a cold store and the supermarket no longer being necessary.

## Inverter Drives

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | H (entire refrigeration plant must be suitable for varying flow rates. Fan motors and compressors have limits to speed that can be achieved) |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | Compressors and fan motors |
| Energy savings (confidence) | 0% for remote, 15% in savings for integral. |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 2 years (integrals only) |

### Energy savings assumed

The only data found to show the benefit of variable speed drives is that by Qureshi and Tassou, (1996). They showed that using variable speed control at 50% load uses about 55% of the full load power. However, the same result is shown for multiple-compressor control. As the refrigeration packs for the store will use multiple-compressor control, there is considered to be no benefit in this technology for the remote cabinets.

The integral cabinets will use on/off control. According to Qureshi and Tassou (1996), at 50% load, on/off control uses approximately 65% of full load power, whereas variable speed compressors use approximately 55% of full load power. There is therefore a reduction of 15% of compressor power if there is inverter control of the integral compressors when running at 50% load.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Qureshi, T. Q., and Tassou, S. (1996). Review paper. Variable-speed capacity control in refrigeration systems. Applied Thermal Engineering., Vol. 16, No. 2, Pp. 103-113.

## Motor Efficiency Controllers (MECs)

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | L |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | Compressors |
| Energy savings (confidence) | Savings of 12-15% on integral plant compressors and 7-10% on remote compressors |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1 year |

### Energy savings assumed

Based on data from EMS European Ltd, savings of 12-15% on integral plant compressors and 7-10% on remote compressors are assumed.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

EMS (European). (n.d.). EnviroStart Motor Control Products and Applications. Walsall, West Midlands, England.

## Lighting - cabinets

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | M (price still reducing, performance improving) |
| Ease of use and installation | M (consideration of drivers and light output) |
| Technology independence | L |
| Maintainability | L (longer life) |
| Legislative concerns | L |
| Scope of application | All cabinets |
| Energy savings (confidence) | 46-70% of lighting power if replacing T8 with LEDs (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | Cost to install of £59.02 per 4ft section based on 2015 figures.  ROI 3 years. |

### Energy savings assumed

The baseline store already has LED lighting in all cabinets. Therefore no savings are assumed.

### Direct emission savings assumed

No direct emission savings are assumed.

## Lighting (store), impact on cabinet performance

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | All stores |
| Energy savings (confidence) | No information (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

No data exists to suggest that there are savings on cabinet heat loads when changing store lighting.

### Direct emission savings assumed

No direct emission savings are assumed.

## Liquid pressure amplification (LPA)

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | Remote plant and floating head pressures |
| Energy savings (confidence) | 20-25% of compressor energy (H) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | ~1 year |

### Energy savings assumed

LPA technology is only relevant if head pressure is allowed to float to a level where the refrigerant liquid pressure needs to be increased to ensure correct operation of the cabinets. Most of the work that is presented in the section above suggests savings of 20-25% have been achieved. Savings only apply to remote plants.

### Direct emission savings assumed

No direct emission savings are assumed.

## Loading (food) temperature and duration

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | All retail cabinets |
| Energy savings (confidence) | Insufficient evidence (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

Insufficient evidence exists to quantify savings.

### Direct emission savings assumed

No direct emission savings are assumed.

## Low emissivity packaging

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | M |
| Availability barriers | H |
| Limits to commercial maturity | H (requires all food in low emissivity packaging) |
| Ease of use and installation | M |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | Chilled or frozen |
| Energy savings (confidence) | A maximum of 4 to 6% for open chillers and 20 to 29% for freezers. (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | <1 year |

### Energy savings assumed

The predicted savings presented by Davies et al. (2012) apply to a horizontal (open top) freezer cabinet. The cabinets in the baseline store are glass door and half glass door/well cabinets. Therefore the savings predicted by Davies et al. (2012) would only apply to part (the non-obstructed, open part) of the well cabinets and the savings that might be achieved cannot be predicted using the data available.

The radiation heat gain to the cabinet is estimated to be up to 5%, 7% and 25% and 37% of the total heat load for open vertical meat chillers, open vertical dairy/deli chillers, frozen open coffin and half glass door freezers respectively (Faramarzi, 2000 and Carbon Trust 2010).

Assuming a reduction in emissivity from 0.79 to 0.01, it is theoretically possible to reduce the heat load by 3.9, 5.5, 19.5 and 28.9% for the different cabinet types respectively. It is unlikely that these reductions would be seen for vertical cabinets which are facing other vertical cabinets.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Carbon Trust (2010). Refrigeration Roadmap. Queen’s Printer and Controller of HMSO, UK

Faramarzi, R.T., Sarhadian, R. and Sweetser, R.S. (2000).Assessment of indoor relative humidity variations on the energy use and power use of a refrigerated display case. Energy efficiency in buildings, ACEEE.

Davies G.F., Man C.M.D., Andrews S.D., Paurine A., Hutchins M.G. and Maidment G.G., Potential life cycle carbon savings with low emissivity packaging for refrigerated food on display. Journal of Food Engineering, Volume 109, Issue 2, March 2012, Pages 202-208.

## Magnetic refrigeration

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | H |
| Limits to commercial maturity | H |
| Ease of use and installation | H |
| Technology independence | M |
| Maintainability | Not known yet |
| Legislative concerns | L |
| Scope of application | Integral cabinets |
| Energy savings (confidence) | Currently worse COPs (L) |
| Direct emissions (confidence) | 100% (H) |
| Payback time (years) | >10 years |

### Energy savings assumed

Information on energy that could be saved by magnetic refrigeration systems in real life usage is currently unavailable. Aprea et al. (2015) suggest that for chilled cabinets the COPs could be similar to those achieved in current cabinets. It has therefore been assumed that magnetic refrigeration can achieve the same energy consumption as the chilled integral cabinets in the baseline store but that magnetic fridge systems will not be available for the freezers in the medium term (5 year timeframe). It has also been assumed that the majority of magnetic systems have maximum duty of around 600 W and so the systems, when available, will initially only be available for integral cabinets.

### Direct emission savings assumed

All direct emissions from refrigerants are assumed to have been removed. It is assumed that the technology would only be applied to chilled integral refrigeration systems.

**Reference:**

Aprea C., Greco A., Maiorino A. and Masselli C. (2015) Magnetic refrigeration: an eco-friendly technology for the refrigeration at room temperature. 33rd UIT (Italian Union of Thermo-fluid-dynamics) Heat Transfer Conference IOP Publishing. Journal of Physics: Conference Series 655 (2015) 012026 doi:10.1088/1742-6596/655/1/012026.

## Nanoparticles in refrigerant

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | H |
| Ease of use and installation | H |
| Technology independence | H |
| Maintainability | H |
| Legislative concerns | H (lack of legislation on H&S, potential unknown health issues) |
| Scope of application | All refrigerated cabinets |
| Energy savings (confidence) | 3-15% of total (M) |
| Direct emissions (confidence) | 0% |
| Payback time (years) | <6 months |

### Energy savings assumed

Energy savings have been shown to be between 3 and 15% for R404A refrigerant (Jaiswal, 2015). This falls within margins shown by Bi et al. (2008; 2011) for domestic refrigerators and therefore seems reasonable. This value has been assumed to be saved on all refrigeration systems.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Bi, S.; Guo, K.; Liu, Z. and Wu, J. Performance of a domestic refrigerator using TiO2-R600a nano-refrigerant as working fluid. Energy Conversion and Management 52 (2011) 733–737.

Bi, S.; Shi, L. and Zhang, L. Application of nanoparticles in domestic refrigerators. Applied Thermal Engineering 28 (2008) 1834–1843.

Jaiswal, R.K., Mishra, R.S (2015). First law efficiency improvement of vapour compression refrigeration system using nano particles mixed with R-404a eco-friendly refrigerant. International Research Journal of Sustainable Science and Engineering. 3:6 ISSN: 2347-6176.

## Night blinds and covers

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | M (necessary to close and open blinds for trading hours) |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | L |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | Open fronted/top cabinets |
| Energy savings (confidence) | 8% assuming 28 hours closed per week. (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1 year if 20% in savings |

### Energy savings assumed

Energy savings whilst the blinds are down have been shown to be between 50 and 75% (Axell and Fahlen (2000), Datta et al. (2005)). Similar savings have been suggested for cabinet covers. For the baseline store the opening times are 24 hours mid-week. However, the store is closed for 20 hours on weekends. If we assume the lower saving of 50% during hours when the store is closed (the lower figure is used, as it will take time to close and open the blinds and re-stocking will need to be done during this period). This provides an 8% reduction in compressor power to all of the open fronted cabinets.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Axell, M., and Fahlèn, P. (2000). VERTICAL DISPLAY CABINET. BORÅS, Sweden: SP (The Swedish National Testing and Research Institute), Box 857, S-501 15 BORÅS, Sweden.

Datta, D., Watkins, R., Tassou, S. A., Hadawey, A and Maki, A., (2005). Formal based methodologies for the design of stand alone display cabinets, Final report to DEFRA for project AFM144, November 2005, 177 pgs.

## Peltier cooling

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | Integral chillers |
| Energy savings (confidence) | -250% (i.e. increase) of compressor power (L) |
| Direct emissions (confidence) | 100% (H) |
| Payback time (years) | No payback |

### Energy savings assumed

Currently the COP of Peltier cooling is lower than that used in the baseline store. The only application for Peltier coolers would appear to be integral chillers where COPs of around 1 might be achievable (Min and Rowe, 2000). The COP of direct expansion integral systems was presented by Grace, Datta and Tassou (2000). They suggested that, depending on optimisation of the system, a chilled cabinet would have a COP of around 2.5. Therefore it is assumed that if Peltier cooling were used as a replacement for the current baseline store chilled integral cabinets, the energy usage would be 2.5 times higher (compressor energy only). It is assumed that that Peltier cooling would not be used on the integral freezers as COPs would be too low.

### Direct emission savings assumed

All direct emissions from refrigerants are assumed to have been removed. It is assumed that the technology would only be applied to integral chillers.

**References:**

Grace, I.; Datta, D.; and Tassou, S. A. (2002). Comparison Of Hermetic Scroll And Reciprocating Compressors Operating Under Varying Refrigerant Charge And Load " (2002). International Compressor Engineering Conference. Paper 1518.

Min, G, and Rowe, D. M. (2000). Improved model for calculating the coefficient of performance of a Peltier module. Energy Conservation and Management, 41(2), 163-171.

## Pipe insulation

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | More applicable to remote plant |
| Energy savings (confidence) | None (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

It is standard practise to insulate pipework. There is no evidence to suggest savings from improved pipe insulation.

### Direct emission savings assumed

No direct emission savings are assumed.

## Pipe pressure drop minimisation

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | All pipe runs |
| Energy savings (confidence) | Limited opportunities in current plant (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | None |

### Energy savings assumed

There is no evidence to suggest that energy could be saved in a typical supermarket store. The technology is unlikely to be applicable to the baseline store as the refrigeration system will already be designed for low pressure drops.

### Direct emission savings assumed

No direct emission savings are assumed.

## Radiant reflectors

Summary:

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | M |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M (may require alterations to cabinets) |
| Technology independence | L |
| Maintainability | M (some cleaning required) |
| Legislative concerns | L |
| Scope of application | Chilled or frozen |
| Energy savings (confidence) | Insufficient data (n/a) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | Insufficient data (n/a) |

### Energy savings assumed

There is insufficient data to suggest how much energy could be saved in the baseline store.

### Direct emission savings assumed

No direct emission savings are assumed.

## Recommissioning

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | All refrigeration systems |
| Energy savings (confidence) | 5 to 31% of total, (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | <6 months |

### Energy savings assumed

Information from Xcel Energy (2010) and Greening Retail (2014) suggest savings between 5 and 31%. The baseline store has undergone recent recommissioning and so the lower value (5%) has been selected.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Greening Retail. (2014). Best Practice Database. Retrieved September 2014, from Greening Retail: http://www.greeningretail.ca/best/energy-conservation/best\_energy\_conserv\_refrig.dot

Xcel Energy. (2010). Information sheet. Recommissioning. Refrigeration Recommissioning. Fine tune your refrigeration system. Colorado, Minnesota, USA.

## Reducing head pressure

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | H (requires EEV) |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | Remote refrigeration |
| Energy savings (confidence) | 15-23% of remote cabinet compressor energy (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 0.5-1 year |

### Energy savings assumed

Based on store trials, compressor energy savings of between 15-23% appear realistic (Ge and Tassou, 2000 and Toscano, Walker and Tetreault, 1983). It is assumed that savings only apply to remote cabinets.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Ge, Y. T. and Tassou, S. A. (2000). Mathematical modelling of supermarket refrigeration systems for design, energy prediction and control. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 101-114.

Toscano, W., Walker, D. and Tetreault, R. (1983). Research and Development of Highly Energy-Efficient Supermarket Refrigeration Systems: Vol 2 - Supplementary Laboratory Testing. Virginia: National Technology Information Service.

## Refrigerants - HFC retrofit with lower GWP HFC

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M/L (dependant on current system) |
| Maintainability | L |
| Legislative concerns | M (F-gas regulations) |
| Scope of application | All refrigeration systems |
| Energy savings (confidence) | 3 and 6% reduction in compressor power (M) |
| Direct emissions (confidence) | Reductions:  R407F 57% (H)  R407A 50% (H) |
| Payback time (years) | ROI 1.3 years, based on re-fit |

### Energy savings assumed

The theoretical improvement in COP given by Coolpack, Danfoss (2013) and Yana Motta, Spatz, and Vera Becerra (2012) varies between 3 and 6% for the replacement of R404A with lower GWP HFCs such as R407A and R407F (which appear the most relevant HFC replacement refrigerants in use today). There is no peer reviewed information to dispute these figures, therefore these figures are used.

### Direct emission savings assumed

The GWP of R407A (2100) is 50% and the GWP of R407F is 43% that of R404A. These have been selected as the maximum and minimum values.

**References:**

Danfoss. (2013, April). Towards more eco-friendly commercial refrigeration systems. Retrieved 2013, from Danfoss Commercial Compressors: http://www.danfoss.com/NR/rdonlyres/3CC6622D-EC8A-4823-A4C4-35C3FF2B2E69/0/FRCCEN085A302.pdf

Yana Motta, S. F., Spatz, M. W., and Vera Becerra, E. (2012). Low Global Warming Refrigerants for Commercial Refrigeration Systems. International Refrigeration and Air Conditioning Conference. Purdue.

## Refrigerants - HFC retrofit with hydrocarbons

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L (Flammability) |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M (Flammability) |
| Technology independence | M |
| Maintainability | L (as above) |
| Legislative concerns | L (will overcome future F-gas regulations) |
| Scope of application | Integral cabinets |
| Energy savings (confidence) | 8% of compressor power (M) |
| Direct emissions (confidence) | 95 to 98.2 % reduction in integral emissions (H) |
| Payback time (years) | <6 months if from new |

### Energy savings assumed

Energy savings of 8% on integral cabinets are assumed based on information from Pederson (2012).

### Direct emission savings assumed

Reductions in direct emissions of 95 to 98.2% on all integral cabinets are assumed based on replacing R404A with hydrocarbons due to their lower GWP.

**Reference:**

Pedersen, P-H. (2012). Low GWP Alternatives to HFCs in Refrigeration Environmental. Projekt no. 1425. Danish Technological Institute.

## Refrigerants - HFC retrofit with HFO

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M (mainly available for air conditioning) |
| Limits to commercial maturity | H |
| Ease of use and installation | M (address flammability issues and capacity loss) |
| Technology independence | M (compatibility, flammability) |
| Maintainability | L (as above) |
| Legislative concerns | L (will overcome future F-gas regulations) |
| Scope of application | Refrigeration compressors |
| Energy savings (confidence) | No data  All freezers 13% (M) |
| Direct emissions (confidence) | All chillers 86% (H)  All freezers 67% (H) |
| Payback time (years) | 1.5 years |

### Energy savings assumed

It is assumed that all chillers could use R450A. There is no data to show the difference in energy when R450A is used as a replacement for R404A, therefore energy savings have only been assumed for freezer cabinets.

It is assumed that all freezers could use R448A. It is assumed the COP of the compressors will increase by 13% compared to R404A (Honeywell).

### Direct emission savings assumed

It is assumed that all chillers could use R450A. This would result in a reduction in direct emissions of 86%, assuming GWP of 570 for R450A compared to 1360 for R134a.

It is assumed that all freezers could use R448A. This would result in a reduction in direct emissions of 67%, assuming GWP of 1400 for R448A compared to 4200 for R404A.

Yana Motta, S. F., Vera Bercerra, E. D., and Spatz, M. W. (2010). Analysis of LWGP Alternatives for Small Refrigeration (Plug-in) Applications. International Refrigeration and Air Conditioning Conference. Purdue: Purdue University.

## Refrigerant – R744

### Summary

|  |  |
| --- | --- |
| Quality of information | 5\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | Remote compressors |
| Energy savings (confidence) | 29% worse to 22% better (H) |
| Direct emissions (confidence) | 50 to 100% less (highest reduction of direct coincides with lowest reduction in indirect). (H) |
| Payback time (years) | 5-10 years |

### Energy savings assumed

It is assumed that all remote plants are replaced by CO2 equipment. There are many accounts of R744 systems using less energy than previous R404A systems, though much of this depends on how often the system runs transcritically and the type of CO2 system.

Carrier (2014), Danfoss (2011), and Alessandro, Filho, and Antunes (2012) all showed savings between 10% and 22%. The greatest saving was with a cascade system (Alessandro, Filho, and Antunes, 2012). Yamasaki, Yamanaka, and Matsumo (2012) showed a 20% improvement using a transcritical system compared to R134a. Alessandro, Filho, and Antunes (2012) showed a 22% improvement for a cascade cycle (CO2/R404A) compared to R404A. On the negative side, Shilliday et al. (2009) showed a 10 to 29% reduction in COP comparing a single-stage transcritical system to R404A.

Based on these results, minimum energy savings of -29% (single-stage transcritical) and maximum of +22% (cascade) on the remote plant are assumed.

### Direct emission savings assumed

It is assumed that all remote plants could be replaced by CO2 with savings of direct emissions of 100% when using a single-stage transcritical CO2 system. However, the most efficient system reported was a cascade system using R404A in the high side. Therefore the direct emissions reduction would be similar to centralised air distribution, which is a reduction of 50%.

**References**

Alessandro, d., Filho, E. P., and Antunes, A. H. (2012). Comparison of a R744 cascade refrigeration system with R404A and R22 conventional systems for supermarkets. Applied Thermal Engineering, 41, 30-35.

Carrier. (2014, April 15). Southern-most Carrier CO2OLtec® Refrigeration System Installed in Valencia. Retrieved from http://www.carrier.com/commercial-refrigeration/en/ib/news/news-article/southern\_most\_carrier\_co2oltec\_\_refrigeration\_system\_installed\_in\_valencia.aspx.

Danfoss. (2011). Save Energy in your Supermarket with a CO2 Refrigeration system - Benchmarking energy optimised HFC stores with transcritical CO2 booster systems. Retrieved 2014, from Danfoss: http://www.danfoss.com/NR/rdonlyres/304EF5E3-63AD-4FA7-BEAA-1729D73E30AA/0/DanfossCaseStudyFakta.pdf

Shilliday, A.; Tassou S.A. and Shilliday N. (2009). Comparative energy and exergy analysis of R744, R404A and R290 refrigeration cycles. Intl Jnl of Low-Carbon Technologies. Volume 4, Issue 2. Pp. 104-111

Yamasaki, H., Yamanaka, M., and Matsumo, K. (2012). Introduction of transcritical refrigeration cycle utilizing CO2 as working fluid. International Compressor Engineering Conference. Purdue.

## Shelf risers and weir plates

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | M |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | M – potential to be removed or broken |
| Legislative concerns | L |
| Scope of application | Open fronted refrigerated display cabinet |
| Energy savings (confidence) | 4 to 8 % of compressor power (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 2-3 years |

### Energy savings assumed

Energy savings between 8 and 16% resulting from the use of risers have been measured by the authors (internal information from authors). If it is assumed that approximately half the open fronted cabinets could benefit from risers (the other half already have them fitted), this generates a benefit of between 4 and 8%.

### Direct emission savings assumed

No direct emission savings are assumed.

## Secondary systems

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | H |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | Remote cabinets |
| Energy savings (confidence) | -15 to +4.9% of compressor power (H) |
| Direct emissions (confidence) | 50 to 100% (H) |
| Payback time (years) | 4-to10 years |

### Energy savings assumed

Energy savings ranging from an increase in energy of 15% (Devotta and Sicars, 2005) to a saving of 4.9% (DelVentura, Evans and Richter, 2008) have been used in the analysis. The savings seem to be due to improved performance of the refrigerated cabinets enabling an elevated evaporation pressure on the refrigeration system. Higher savings reported by Pajani et al. (2004) are likely due to other improvements in the system and therefore have been ignored in this analysis.

### Direct emission savings assumed

The reduction in direct emissions is dependent on whether the refrigerant of the primary circuit remains R404A or is changed to a refrigerant with a lower GWP. The best case is to change to a refrigerant with a GWP close to 0, in which case the direct emissions can be reduced by almost 100%. If the refrigerant remains R404A, the lower charge and reduced circuit could be expected to reduce the GWP by 50%. This is based on the 3% leakage rate reported by Pajani et al. (2004) compared to the baseline store’s leakage rate of 6%.

**References:**

DelVentura, R., Evans, C. and Richter, I. (2008). *Secondary loop systems for the supermarket industry.* Bohn.

Devotta, S. and Sicars, S. (2005). Refrigeration – Chapter 4. In IPCC/TEAP Special Report: Safeguarding the Ozone Layer and the Global Climate System.

Pajani, G., Giguère, D. and Hosatte, S. (2004). Energy efficiency in supermarkets-secondary loop refrigeration pilot project in the Repentigny Loblaws, CANMET Energy Technology Centre – Varennes, Natural Resources Canada, Report Ref. CETC-Varennes 2004- (PROMO) 170-LOBLA2, 5 pgs.

## Short air curtains

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | H |
| Limits to commercial maturity | H (demonstration cabinets in store. No production cabinet available ) |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | Currently unknown |
| Legislative concerns | L |
| Scope of application | Open fronted cabinets |
| Energy savings (confidence) | 30% of open fronted chilled cabinet energy (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 1.5 years |

### Energy savings assumed

Energy savings of 30% are claimed by Adande (2015). The technology can only be applied to open fronted cabinets (integral and remote).

### Direct emission savings assumed

No direct emission savings are assumed.

## Store dehumidification

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | M ( integration with store HVAC) |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | All refrigerated cabinets. Benefits will be less if chilled cabinets are closed door. |
| Energy savings (confidence) | 2 to 4% refrigeration energy (M), but consequent increase in HVAC energy. |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | n/a |

### Energy savings assumed

There are clearly benefits to the refrigeration system in dehumidification throughout the year. However, these benefits may only be practically possible in summer. According to Howell, 1993a, Howell, 1993b and Howell et al., 1999 there should be overall benefits in the UK in the summer months (UK dew point is above 7ºC for approximately 5 months of the year). Orphelin, Marchio and D’Alanzo (1999) estimated refrigeration savings of between 2 and 4% in a French supermarket, and these values have been used in the analysis.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Howell, R. H. (1993a). Effects Of Store Relative Humidity On Refrigerated Display Case Performance. ASHRAE Transactions: Research, 667-678.

Howell, R. H. (1993b). Calculation Of Humidity Effects On Energy Requirements Of Refrigerated Display Cases. ASHRAE Transactions: Research, 679-693.

Howell, R. H., Rosario, L., Riiska, D., and Bondoc, M. (1999). Potential Savings In Display Case Energy With Reduced Supermarket Relative Humidity. 20th International Congress of Refrigeration. Sydney: International Institute of Refrigeration.

Orphelin, M., Marchio, D., and D'Alanzo, S. L. (1999). Are There Optimum Temperature and Humidity Set points for Supermarkets? ASHRAE Journal, 497-507.

## Store temperature control

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | M (balance energy and customer comfort) |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | H (interactions between refrigeration and HVAC) |
| Maintainability | M ( additional plant possibly required. Optimisation of controls may be complex) |
| Legislative concerns | L |
| Scope of application | All stores |
| Energy savings (confidence) | 4% of compressor energy for chillers (L)  2% of compressor energy for freezers (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | <1 month |

### Energy savings assumed

Assuming store temperature is reduced by 1°C, the impact on compressor energy based on temperature difference between the cabinet and the ambient conditions (25°C difference for chillers, 40°C for freezers) would be 4% for chillers and 2% for freezers. This assumes that the reduction in set-point does not come from air conditioning.

### Direct emission savings assumed

No direct emission savings are assumed.

## Strip curtains

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | H |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | L |
| Maintainability | M ( renewal of damaged or aged strips required) |
| Legislative concerns | L |
| Scope of application | Open cabinets |
| Energy savings (confidence) | 18-60% of cabinet energy (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 4 months-1 year |

### Energy savings assumed

Energy savings of 18 to 60% resulting from the use of strip curtains have been reported by Redwood Strip Curtains Ltd. However, there is no other independent data to back this up. There are previous studies on cold stores which show 80 to 92% reduction of infiltration through an open door when using strip curtains (Cleland and Chen, 2004). It is considered by the authors that the savings suggested by Redwood are feasible and will depend a great deal on the condition of the strip curtain and how often they are opened by customers.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Cleland and Chen (2004). A modified model to predict air infiltration into refrigerated facilities through doorways. ASHRAE Transactions. Part 1. No. 4672. 58-66.

Redwood Strip Curtains Ltd. (n.d.). Coolstrip Chiller Curtains and Blinds. Retrieved 12 27, 2013, from Redwood Strip Curtains Ltd: http://redwoodstripcurtains.co.uk/chiller-curtains-chiller-blind.

## Suction-liquid heat exchangers (SLHE) or liquid-suction heat exchangers (LSHE)

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | Depends on refrigerant |
| Maintainability | L |
| Legislative concerns | None |
| Scope of application | Remote refrigeration |
| Energy savings (confidence) | 20% reduction in compressor energy for freezers and no benefit for chillers (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | < 1 year |

### Energy savings assumed

Klein et al. (2000) show a 20% increase in capacity and therefore COP for R404A with a SLHE effectiveness of 1. Nuriyadia et al. (2015) show a maximum benefit in capacity/COP of 20% for a freezer at -20ºC with a HE effectiveness of about 0.5. However, for a refrigerator at 0ºC there was a 5% reduction in capacity/COP.

A 20% benefit for freezers has been assumed. The technology has not applied to chillers. This assumes that freezers do not already have SLHEs.

### Direct emission savings assumed

No direct emission savings are assumed.

**References:**

Klein, S. A., Reindl, D. T., and Brownell, K. (2000). Refrigeration System Performance using Liquid-Suction Heat Exchangers. International Journal of Refrigeration, 23(8), 588-596.

Nuriyadia, M., Sumerua, Nasutionb, H. (2015). The effect of liquid suction heat exchanger sub-cooler on performance of a freezer using R404A as working fluid. Jurnal Teknologi, 76:11 (2015) 57–61.

## Suction pressure control

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | M (need to make sure control parameters are correct) |
| Legislative concerns | L |
| Scope of application | All remote systems |
| Energy savings (confidence) | 11.4% of compressor energy for all remote frozen cabinets and 1.5% for remote chillers (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | <1 year |

### Energy savings assumed

Limited information is available. Parker Hannifin Corporation (2010) showed an 11.4% reduction in energy for the low temperature pack and 1.5% reduction for medium temperature pack when suction pressure was controlled. It is assumed these savings apply only to remote cabinets.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Parker Hannifin Corporation. (2010). Proven Value of Electronic Suction Control. Retrieved 2013, from Sporlan Online: http://sporlanonline.com/literature/100/ev/100-302.pdf

## Tangential fans

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | M |
| Maintainability | M |
| Legislative concerns | None |
| Scope of application | Refrigerated cabinets |
| Energy savings (confidence) | 2% of compressor energy on remotes and integrals (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | >10 years |

### Energy savings assumed

Based on the work by Faramarzi et al. (2000), savings of 2% on compressor energy for all cabinets in the baseline store are assumed.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Faramarzi, R.; Sarhadian, R.; Coburn, B.; Mitchell, S. and Lutton, J. (2000) Analysis of energy enhancing measured in supermarket display cases. ASHRAE Annual Meeting, Anaheim.

## Thermostatic flow control (TFC)

### Summary

|  |  |
| --- | --- |
| Quality of information | 1\* |
| Barriers to staff/customers | L |
| Availability barriers | H |
| Limits to commercial maturity | H |
| Ease of use and installation | M |
| Technology independence | M (issue with glide) |
| Maintainability | L |
| Legislative concerns | M (more refrigerant) |
| Scope of application | Integral cabinets |
| Energy savings (confidence) | 17% of compressor energy in integrals (L) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | <6 months if energy consumption claims realistic |

### Energy savings assumed

Based on the work of Zimmermann (2004), savings on compressor energy of 17% are assumed. This is applied to integral cabinets only.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Zimmermann, L. C. W. (2004). Patent No. PCT/DK2004/000611. International.

## Training and maintenance

### Summary

|  |  |
| --- | --- |
| Quality of information | 2\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | L |
| Ease of use and installation | L |
| Technology independence | L |
| Maintainability | L |
| Legislative concerns | L |
| Scope of application | All refrigeration systems |
| Energy savings (confidence) | No quantified evidence for savings (L) |
| Direct emissions (confidence) | 0% (M) |
| Payback time (years) | n/a |

### Energy savings assumed

Options to reduce energy consumption through training are vast and include better training of staff when using the cabinets, better training of refrigeration engineers, better design of supermarkets and training of cabinet manufacturers to develop more energy efficient cabinets. Therefore it is difficult to exactly quantify savings. Many options are included in other sections (e.g. defrosting, settings, store operation). Based on the information available, it has been assumed that savings cannot be quantified.

### Direct emission savings assumed

Limited information is available on refrigerant loss reductions through better maintenance. Historical evidence from supermarkets has shown that refrigerant losses can be reduced by around 15% by better maintenance but there is little published evidence to corroborate this. The baseline store already has reduced refrigerant losses to approximately 6% and so it has been assumed that most training and maintenance initiatives have already been applied and no further savings are viable.

## Trigeneration

### Summary

|  |  |
| --- | --- |
| Quality of information | 4\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | Unknown |
| Legislative concerns | L |
| Scope of application | Compressor |
| Energy savings (confidence) | 100% on remote chiller (M) |
| Direct emissions (confidence) | 100% of remote chiller (M) |
| Payback time (years) | At best 3 years |

### Energy savings assumed

Maidment et al. (1999) and Maidment and Tozer (2002) have shown it theoretically possible to use the waste heat from a CHP system to cool chiller packs using an absorption chiller. It is assumed that there is enough waste heat to do this and this heat would otherwise be wasted, and that grid waste heat is also wasted. With these assumptions, all of the remote chiller compressor power would be saved.

### Direct emission savings assumed

If the remote chiller packs are replaced with a secondary absorption system, it is assumed that there are no direct emissions with this system, as ammonia has a GWP of 0.

**Reference:**

Maidment, G. G. and Tozer, R. M. (2002). Combined cooling heat and power in supermarkets. Applied Thermal Engineering, 22, 653–665.

## Two stage compression

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | Mainly R744 |
| Maintainability | M |
| Legislative concerns | None |
| Scope of application | Remote freezers |
| Energy savings (confidence) | 12% of compressor energy (M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | Unknown |

### Energy savings assumed

Coolpack v1.50 (IPU) was used to calculate the COP of R404A DX systems with an evaporating temperature of -40ºC and condensing temperature of 35ºC, one single-stage and the other two-stage with liquid subcooling. The COP was 1.297 for the single-stage and 1.454 for the two-stage system. Therefore we will assume a 12% benefit in energy for the two-stage system for the freezer packs. The benefits of subcooling and reduced discharge temperature are reasonably insignificant for the chiller packs and therefore, the technology is only considered appropriate to the freezer packs.

### Direct emission savings assumed

No direct emission savings are assumed.

## Vacuum insulated panels (VIP)

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | L |
| Limits to commercial maturity | M (available for smaller cabinets) |
| Ease of use and installation | H (install during manufacture of cabinet) |
| Technology independence | M (structural considerations necessary in some cases) |
| Maintainability | L (maintenance free) |
| Legislative concerns | L |
| Scope of application | All compressors |
| Energy savings (confidence) | 1.2 to 1.5% of compressor power for chillers and 9 to 11.5% for freezers.(M) |
| Direct emissions (confidence) | 0% (H) |
| Payback time (years) | 3.1 years for freezers |

### Energy savings assumed

For aisles of remote cabinets, there is much less benefit than that shown above due to the end walls only being at the ends of the aisles. The main benefits in VIP panels would be to the top of the canopy and the rear. The greatest benefit would concern solid door freezer integral cabinets where there are many surfaces which would benefit from VIPs and a large temperature difference.

The conduction heat load is approximately 2% for an open chilled cabinet and 15% for a frozen cabinet. Assuming a worst-case benefit of 2.5 times better than PU and a best case of 4.3 times better than PU (Hammond and Micic, 2013), the heat load for a chilled cabinet would reduce by 1.2 to 1.5% and frozen cabinets by 9 to 11.5%.

### Direct emission savings assumed

No direct emission savings are assumed.

**Reference:**

Hammond, E., and Micic, G. (2013). Simulation vs. Experimental Results of VIPs Embedded Within PU Insulation. International Cold Chain Conference. Paris: International Institute of Refrigeration.

## Water loop systems (R1270)

### Summary

|  |  |
| --- | --- |
| Quality of information | 3\* |
| Barriers to staff/customers | L |
| Availability barriers | M |
| Limits to commercial maturity | M |
| Ease of use and installation | M |
| Technology independence | M |
| Maintainability | M |
| Legislative concerns | L |
| Scope of application | Remote packs |
| Energy savings (confidence) | 7.9 to 16% LT and HT compressors (M) |
| Direct emissions (confidence) | 100% (H) |
| Payback time (years) | 2-3 years |

### Energy savings assumed

The benefit to the refrigeration energy consumption in integral systems derives from replacing the refrigerant with a more efficient refrigerant, e.g. R1270, in which case a 16% reduction in COP was reported by King et al. (2010). The benefit to the store energy consumption is the reduction in heating required to overcome the cooling of the remote cabinets.

It is assumed that the integral systems all operate on R1270 on both the LT and MT packs and that the COP increases between 7.9 and 16% (Pederson, 2012 and King et al., 2010).

### Direct emission savings assumed

It is assumed that leakage rate drops from 6.1 (baseline leakage rate) to 1.5% (leakage rate for integrals) and the GWP drops from 4200 to 1.8, giving a reduction of effectively 100%.

**References:**

King, L.; Garvey, I., Gartshore, J. and Benton, S. Hydrocarbons in Commercial Refrigeration. Proc. Inst. R. 2010-11. 5-1.

Per Henrik Pedersen (2012). Low GWP Alternatives to HFCs in Refrigeration Environmental. Projekt no. 1425. Danish Technological Institute.

# Results from refrigeration system modelling

### Validation of model

The predicted total power of the refrigeration system for the store was compared with the total power of the 9 refrigeration electricity meters in the store. The total estimated power was 8.7% lower than the average electricity meter power over a year. It should be noted that it was not possible to be entirely sure what equipment was connected to each of the electricity meters, and therefore the refrigeration energy from the meters can only be considered an estimate.

### Technologies excluded from analysis

A number of technologies were considered but were not included in the graphs due to either the baseline supermarket already having applied the technologies or there being insufficient evidence to be able to quantify the savings that the technologies could achieve (Table 15).

Table . Technologies excluded from analysis.

|  |  |
| --- | --- |
| **Already applied in baseline supermarket** | **Insufficient evidence** |
| Anti-sweat heaters  DC (EC) evaporator fans  Distributed system  Lighting - cabinets (LED)  Pipe insulation  Minimising pipe pressure drops | Absorption  Adsorption  Cabinet loading  Dynamic demand  Improved cabinet loading  Improved cabinet location  Improved cabinet temperature control  Diagonal compact fans  Dual port TEV  Dynamic demand  Electronic expansion valves  Enhanced internal heat transfer (micro-fins)  High-efficiency compressors  Polygeneration  Radiant reflectors  Training and maintenance  Ultrasonic defrosting of evaporators |

### Cabinet technologies

Cabinet technologies were divided into those that could be applied to the current cabinets and those that could only be applied to new cabinets. Figure 15 and Figure 16 show results for current cabinets and Figure 17 and Figure 18 show the results for new cabinets.

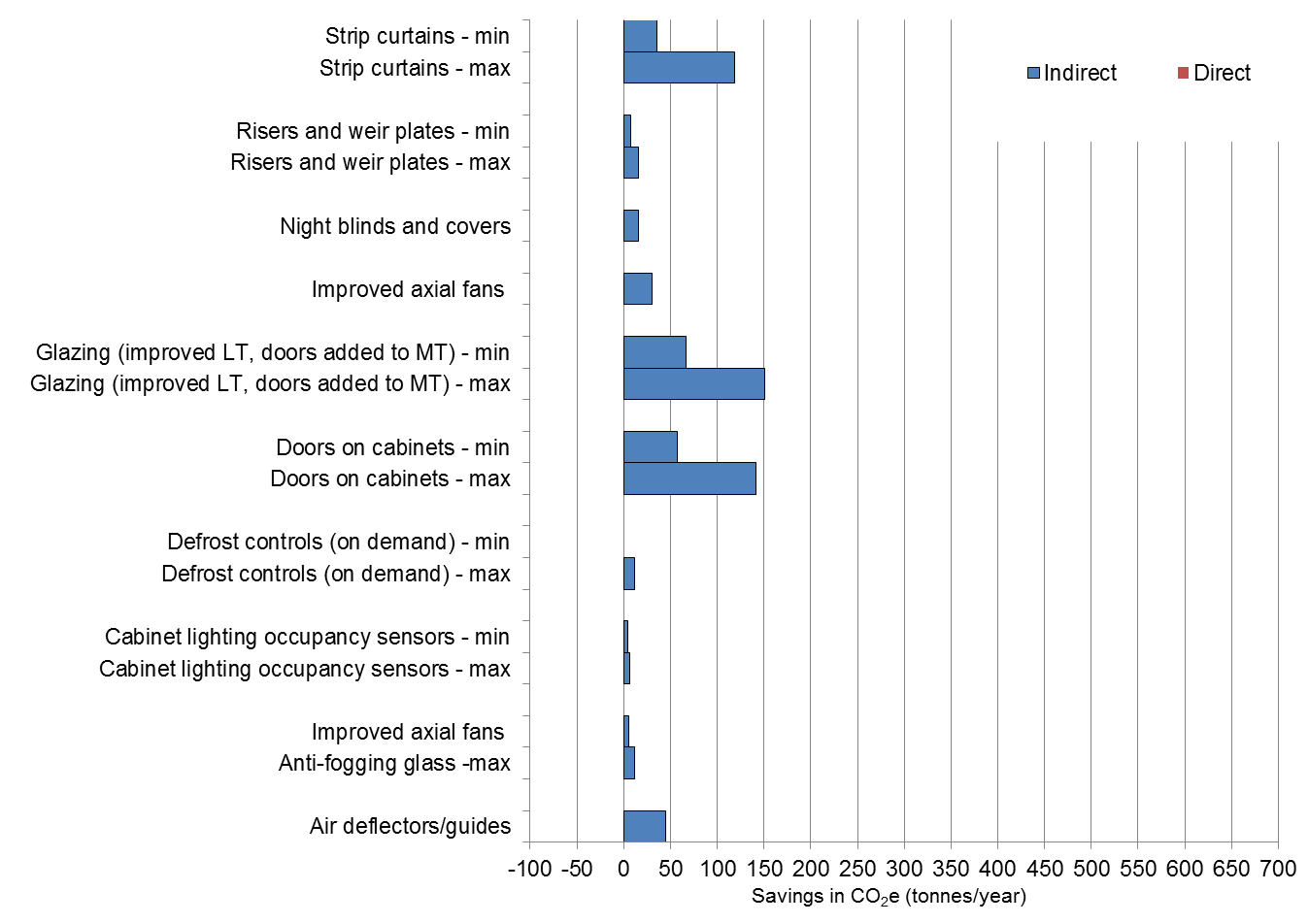


Figure . Graph showing technologies that could be applied to current cabinets.

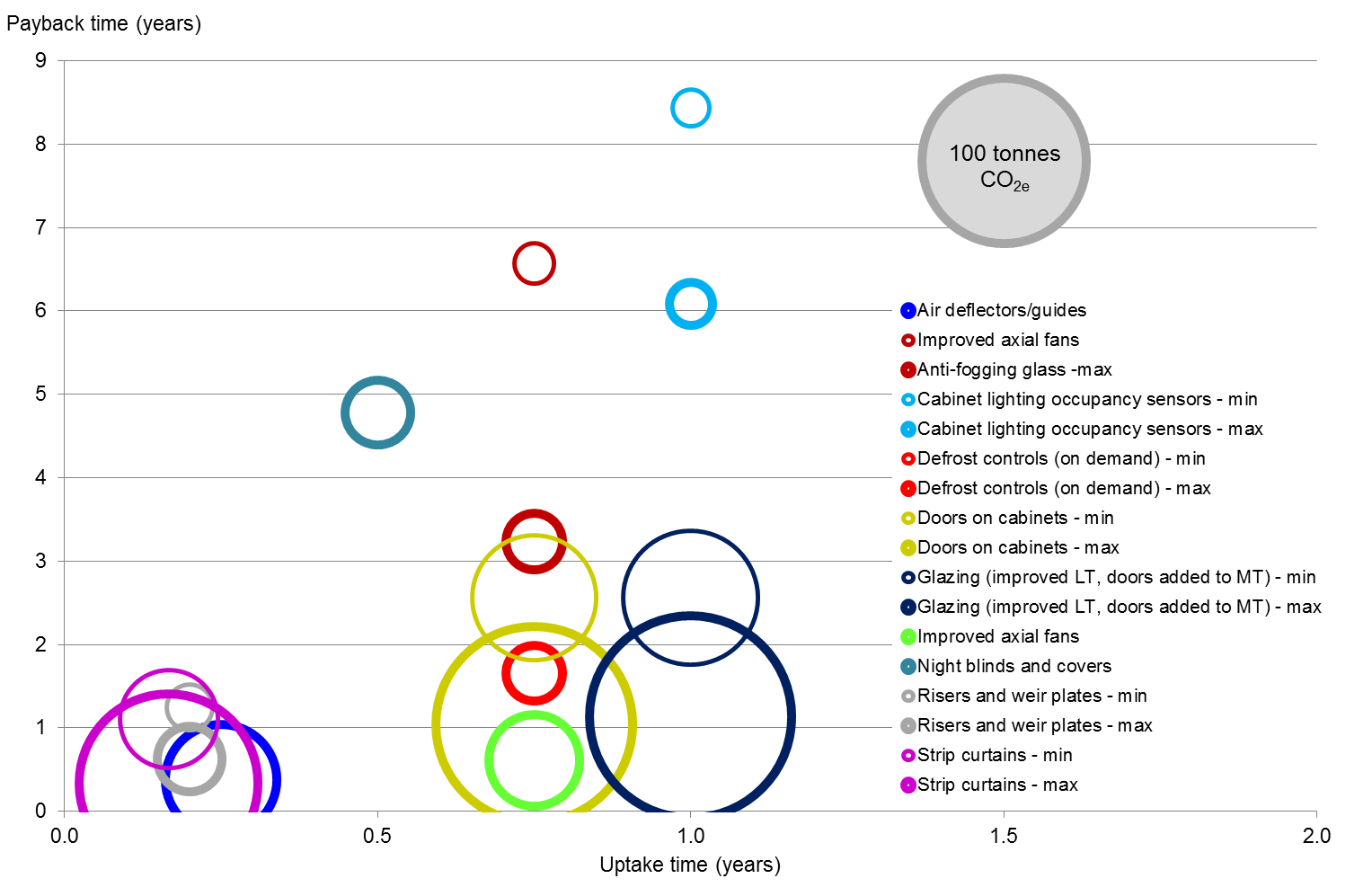


Figure . Bubble map showing technologies that could be applied to current cabinets.

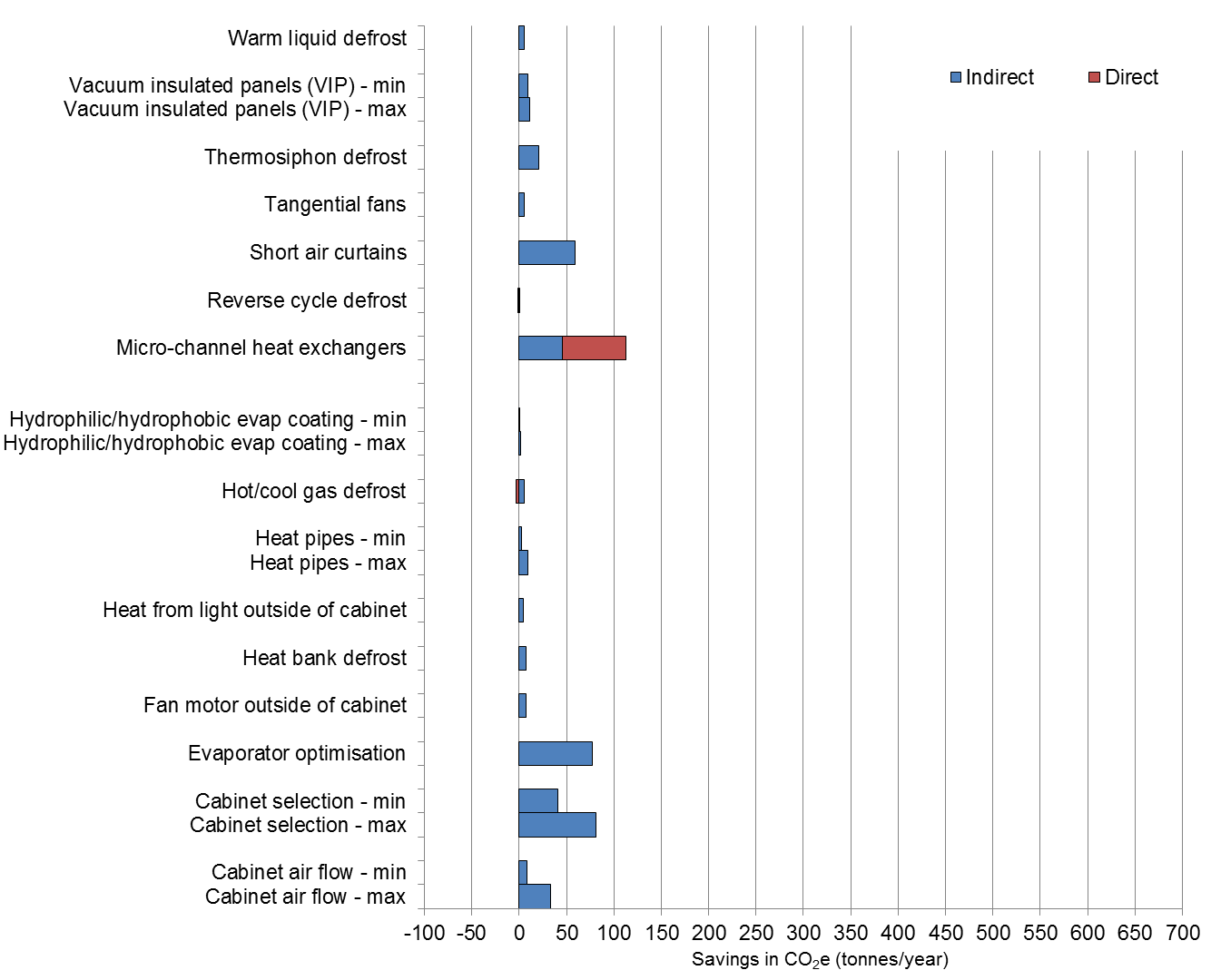


Figure . Graph showing technologies that could be applied to new cabinets.

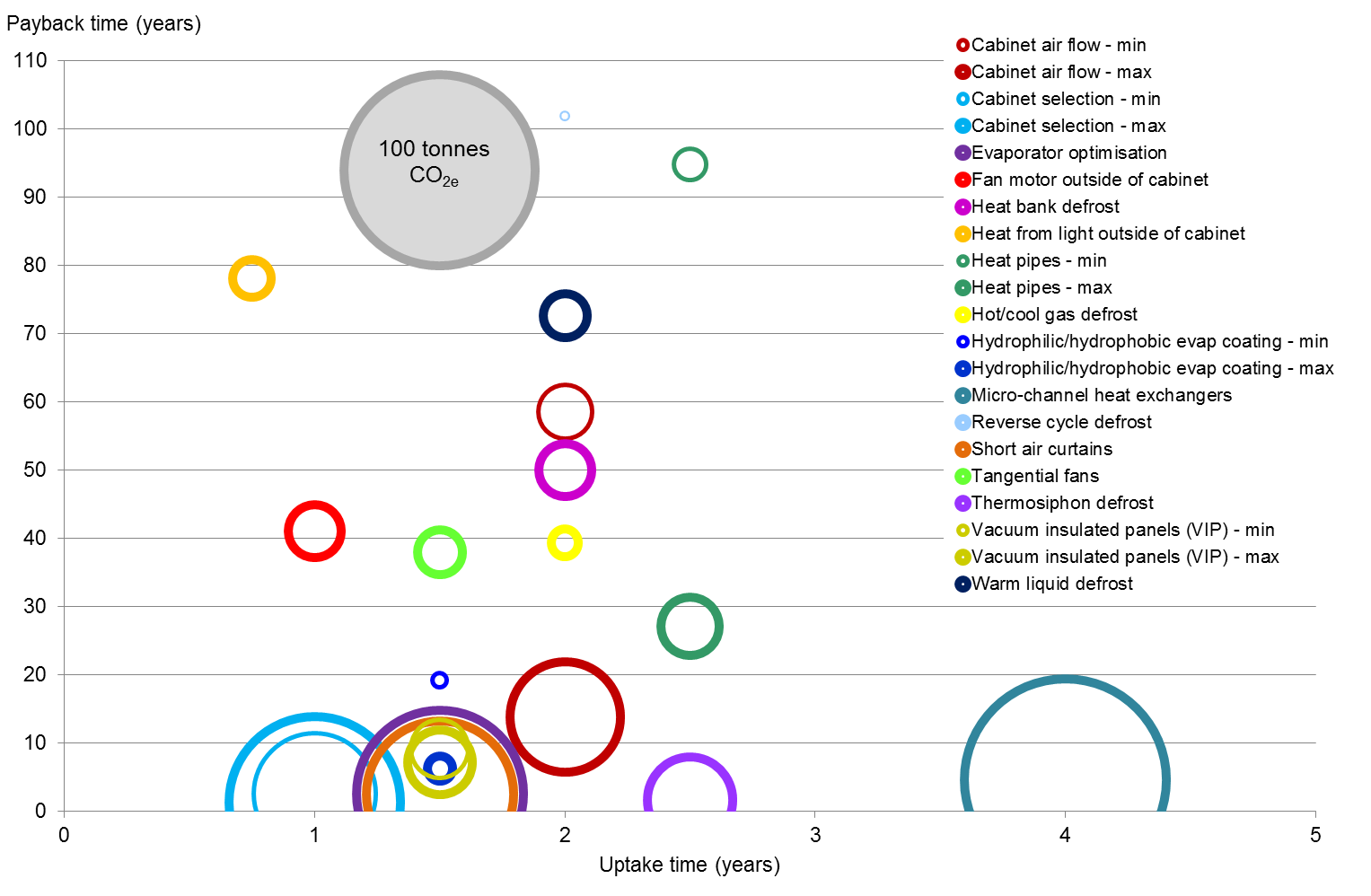


Figure . Bubble map showing technologies that could be applied to new cabinets.

### Refrigeration system technologies

Refrigeration system technologies were divided into those that could be applied to the current system and those that could only be applied to a new system. Figure 19 and Figure 20 show the results for the current refrigeration system and Figure 21 and Figure 22 show the results for a new refrigeration system.

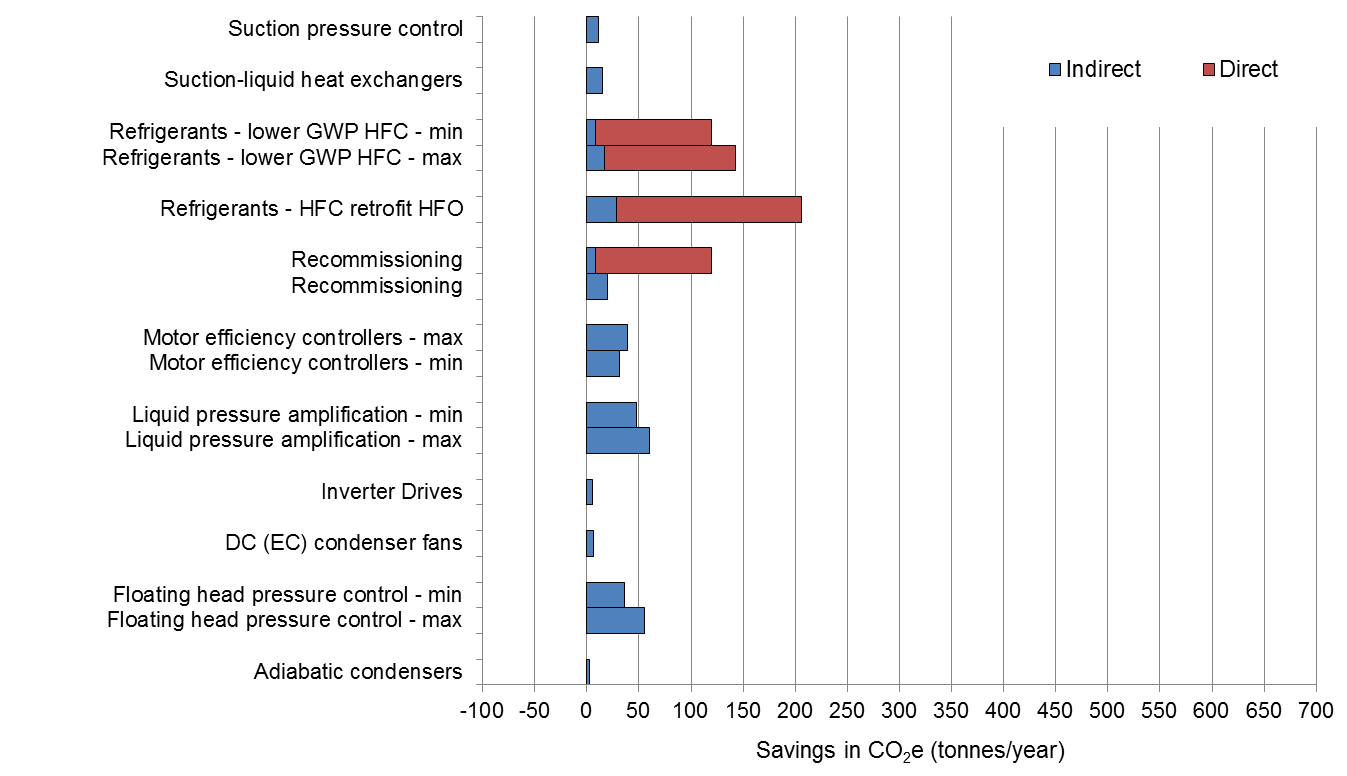


Figure . Graph showing technologies that could be applied to current refrigeration systems.

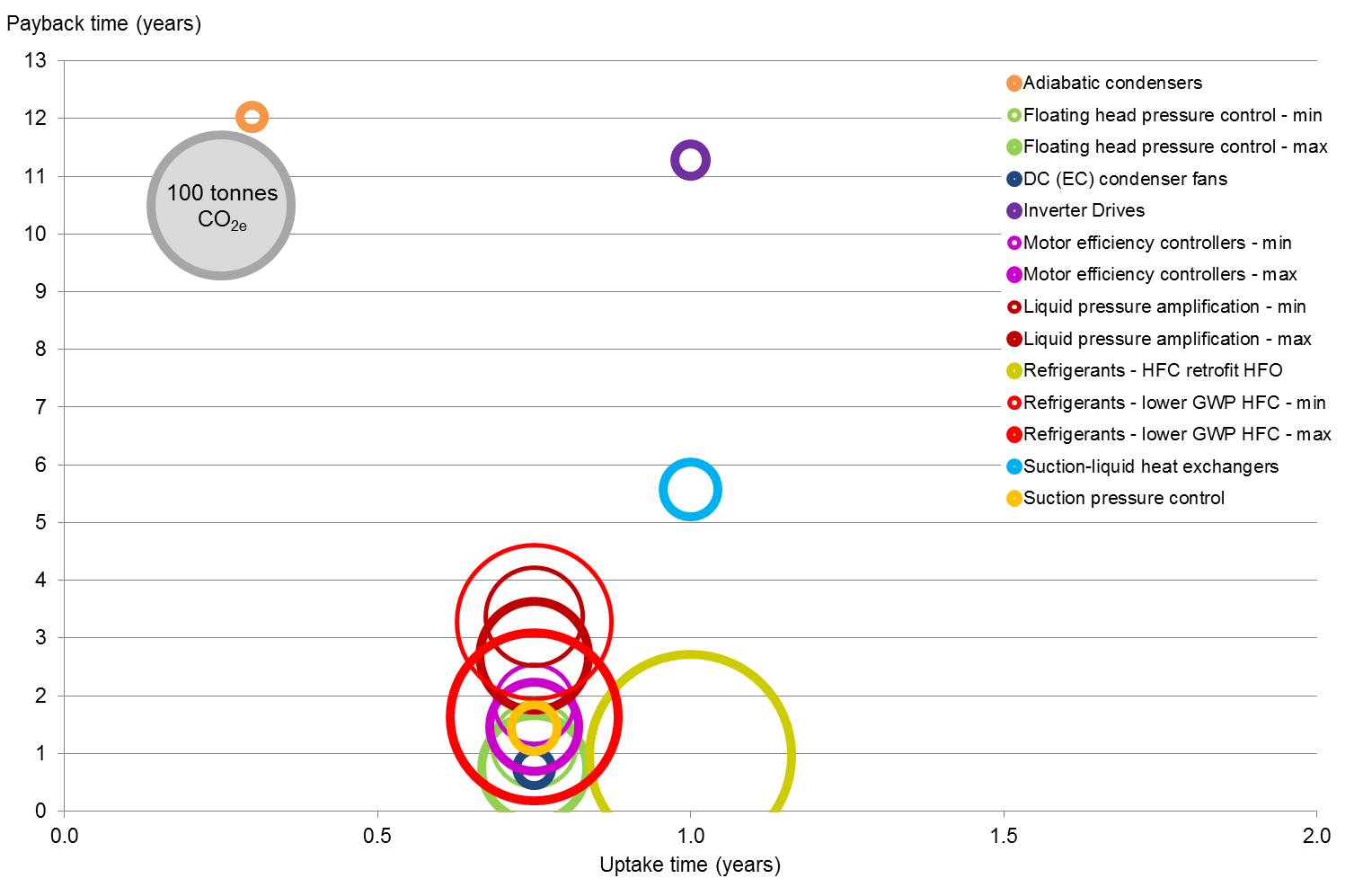


Figure . Bubble map showing technologies that could be applied to current refrigeration systems.

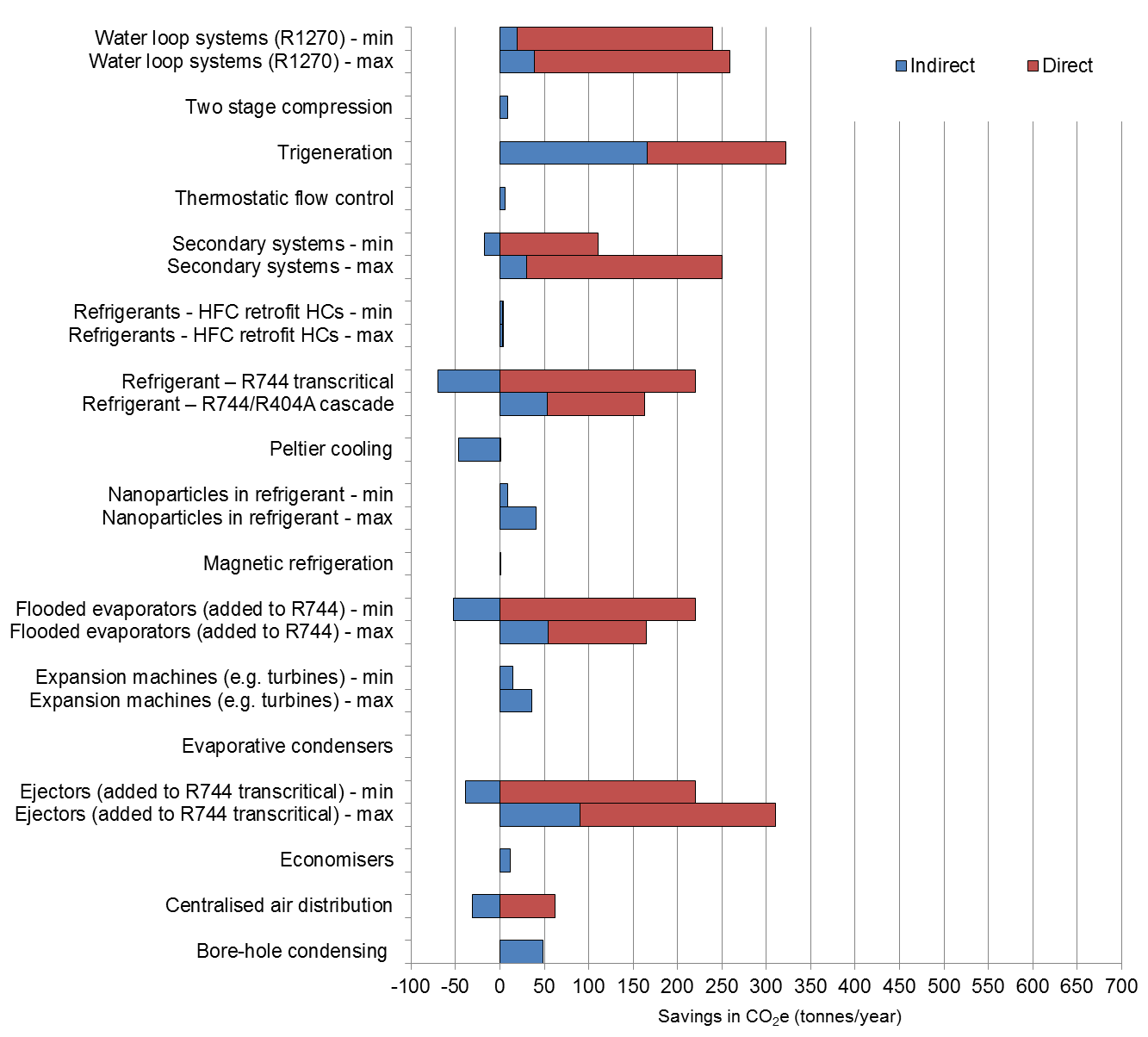


Figure . Graph showing technologies that could be applied to new refrigeration systems.

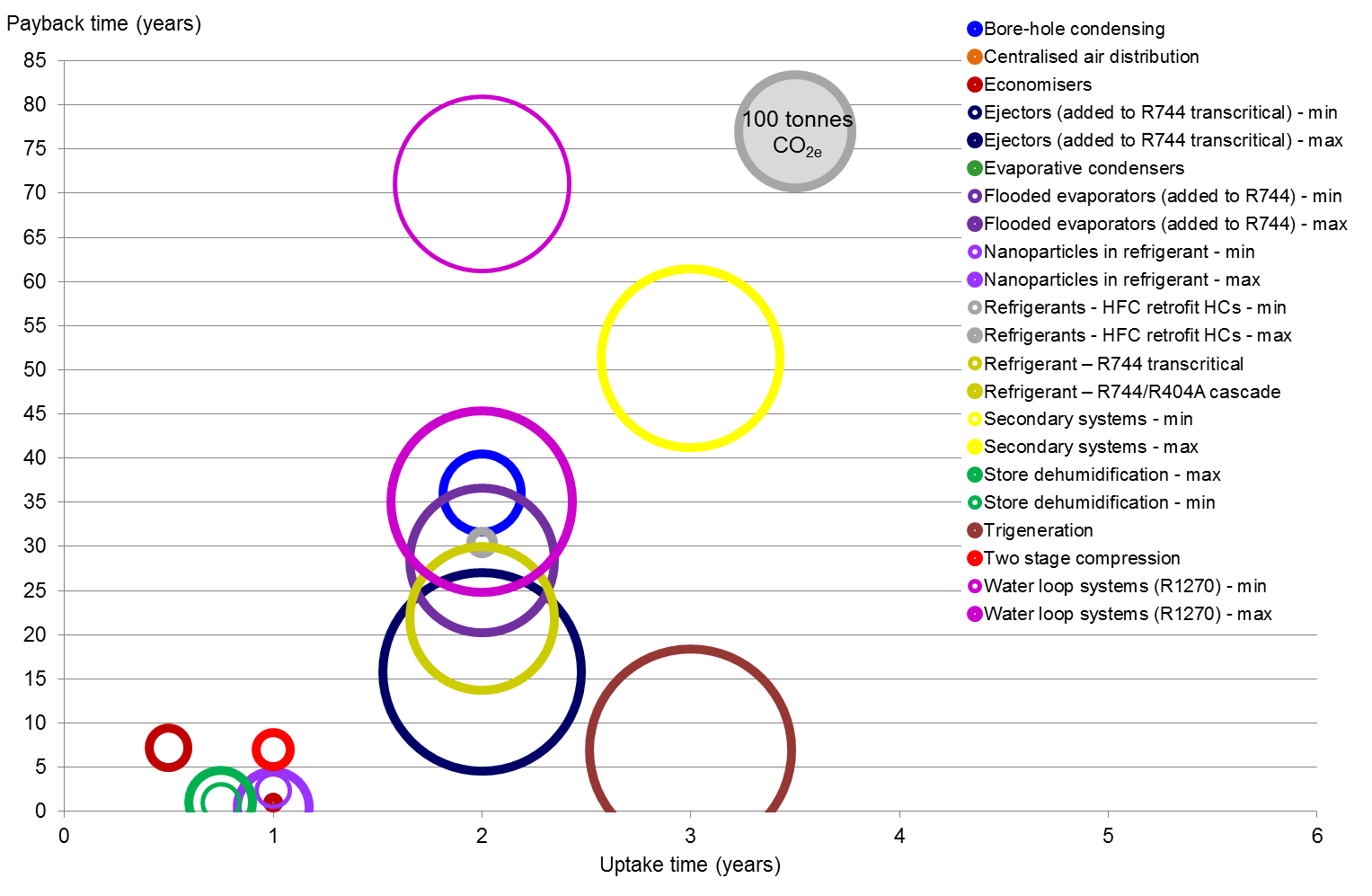


Figure . Bubble map showing technologies that could be applied to new refrigeration systems.

### Other technologies

Other refrigeration related technologies which did not fall directly under the cabinet or refrigeration system headings are shown in Figure 23 and Figure 24.

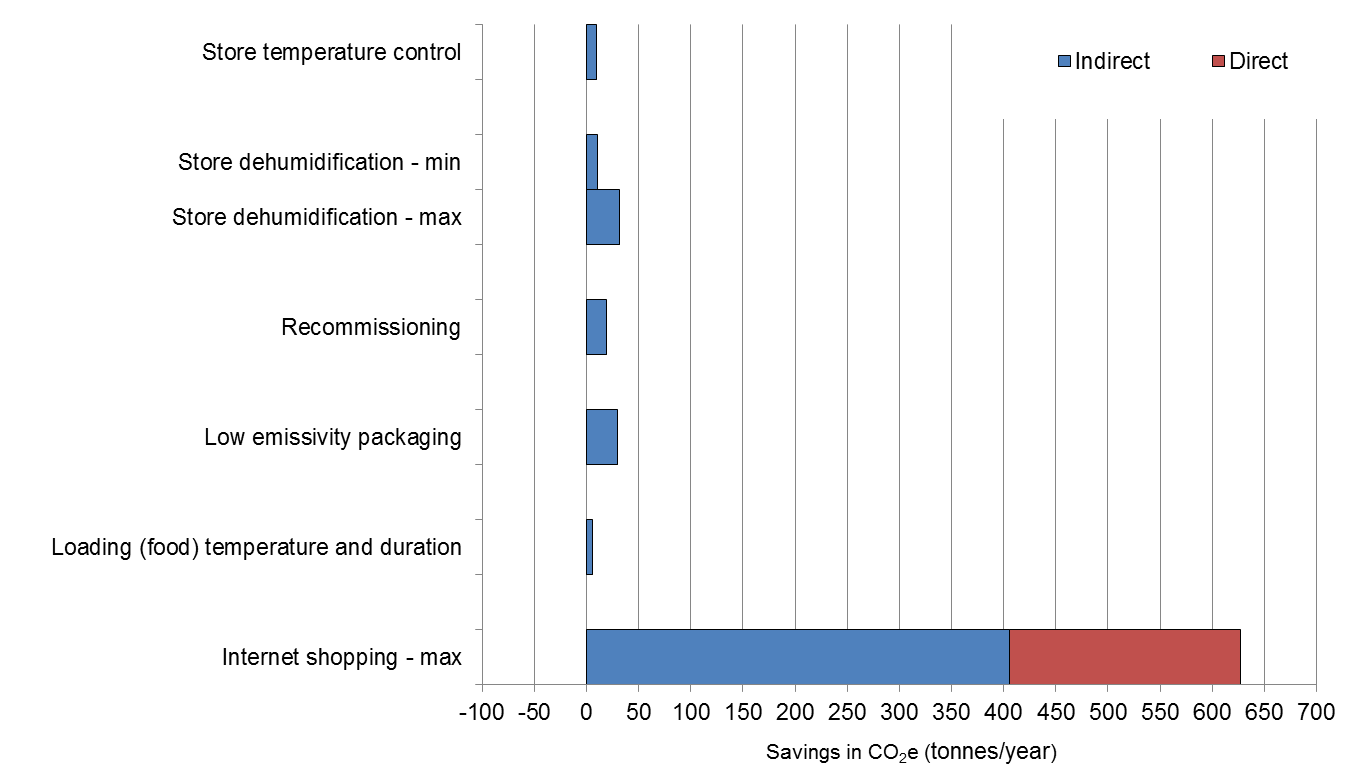


Figure . Graph showing technologies that could be applied to supermarket refrigeration systems.

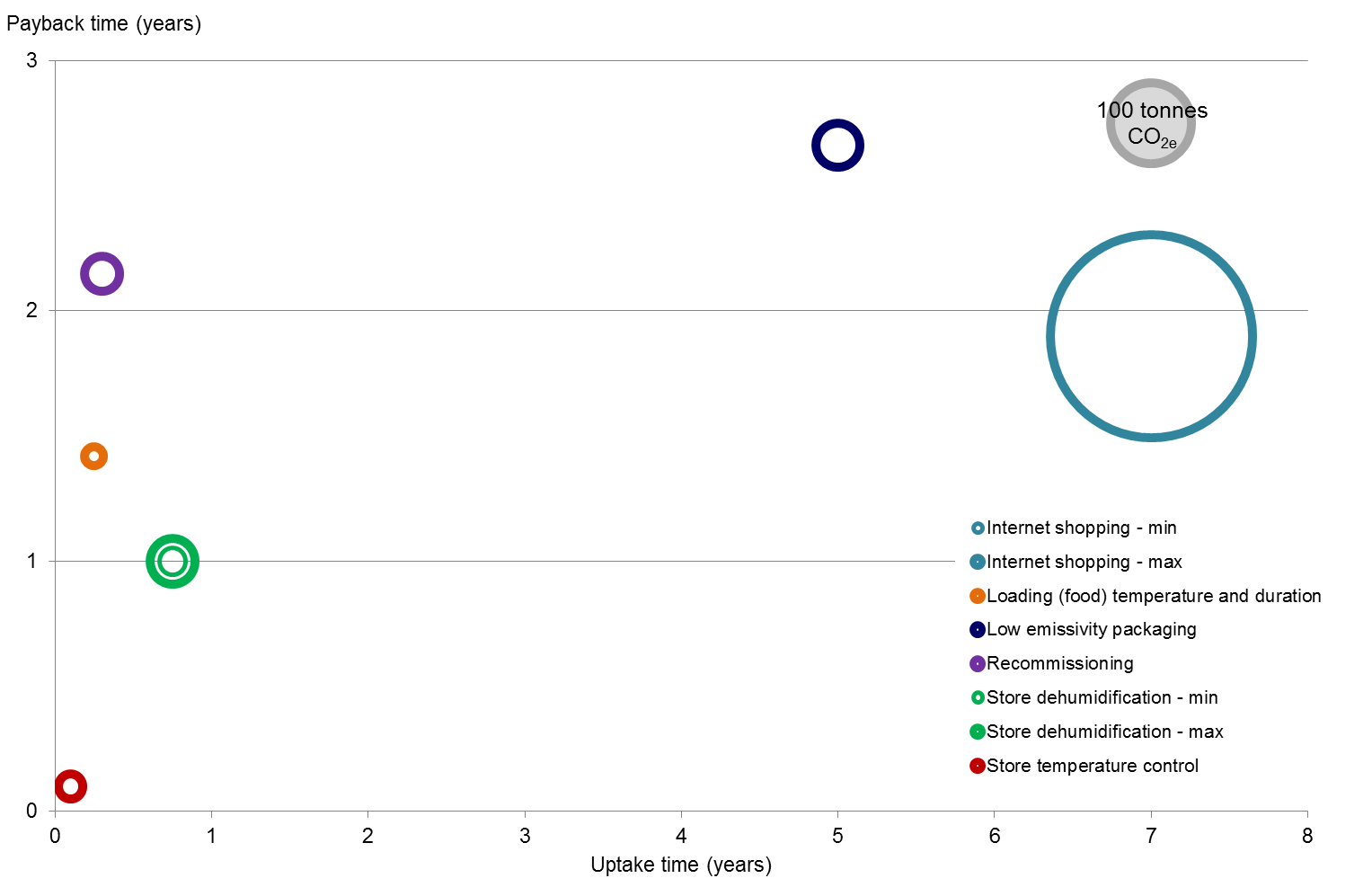


Figure . Bubble map showing technologies that could be applied to supermarket refrigeration systems.

### Impact of results

The assessment of carbon savings in the baseline store demonstrated that considerable savings (over 300 tonnes CO2e p.a.) could be achieved with a single technology. Most of the carbon savings associated with cabinets were related to indirect emissions. All retrofit refrigeration system technologies had some indirect savings with the refrigerant technologies adding direct savings. With new refrigeration systems, the direct carbon savings tended to be greater, although almost all technologies demonstrated indirect savings.

For retrofitting to current cabinets, the greatest savings could be achieved by fitting doors (between 103.3 and 140.7 tonnes CO2e/year which could be increased to between 112.5 and 149.9 CO2e/year if improved glazing was also fitted to these doors and the freezer cabinets). Strip curtains were also an option to reduce emissions but were unlikely to be acceptable to the supermarket. Air deflectors were estimated to save 44.9 CO2e/year and so would be a good option if doors were not acceptable to the supermarket. For new stores the best option was to select the best cabinets available currently. If looking for further improvements, evaporator optimisation and new evaporator technologies or the use of short air curtains were attractive options.

The greatest savings in emissions for the baseline store refrigeration plant were related to alternative refrigerants. Using lower GWP HFCs could save up to 142.8 CO2e/year and the use of a HFO, 208.4 CO2e/year. A large proportion of these savings were from reductions in direct emissions. For new refrigeration systems, the use of trigeneration and water loop systems seemed the most attractive options to reduce emissions. Although not providing such large savings, the use of R744 with or without ejectors and the use of secondary systems also had high emissions savings.

### Assessment of application of several technologies at the baseline store

The savings for each of the technologies cannot be added together. For example, if doors were put on cabinets there would be a reduction in compressor energy, therefore a technology which reduced compressor energy would have a lower individual emissions saving. The financial paybacks for each system will vary and may make some technologies that have the potential to reduce emissions look less attractive when balanced against their high cost and the potentially long time required to apply the technology.

Marginal abatement cost (MAC) curves were created for the technologies that could simply and practically be applied to the baseline store. A MAC curve demonstrates which carbon reduction measures save the most money. An assumption was made in all calculations that energy and carbon savings were applied over a 3 year period. If minimum and maximum carbon reduction values were calculated, a mean of the 2 values was used in the MAC curves. All other assumptions were as described above for the store modelling.

#### Technologies suggested for cabinets in the baseline store

The most attractive technology (in terms of carbon savings) for cabinets was the addition of doors on all cabinets and the use of highly efficient glazing on frozen cabinets. This then eliminated some other technologies as they were not compatible with cabinet doors (e.g. air deflectors, night blinds etc.). A decision flow chart depicting the technology selection process is shown in Figure 25. The final MAC graph is shown in Figure 26. Cabinet doors, short air curtains, improved cabinet air flow, evaporator optimisation and improved axial fans all showed a negative cost of abatement for carbon reduction. However, there was a relatively high cost for the application of occupancy sensors of £1,285/tonne of carbon abated.

**Option 1:   
Doors**

**Option 2:   
Short air curtains**

**Cabinet air flow**

**Evaporator optimisation   
Improved axial fans**

**Option 3:   
Cabinet lighting occupancy sensors**

Figure . Options for selection of cabinet technologies in baseline store.

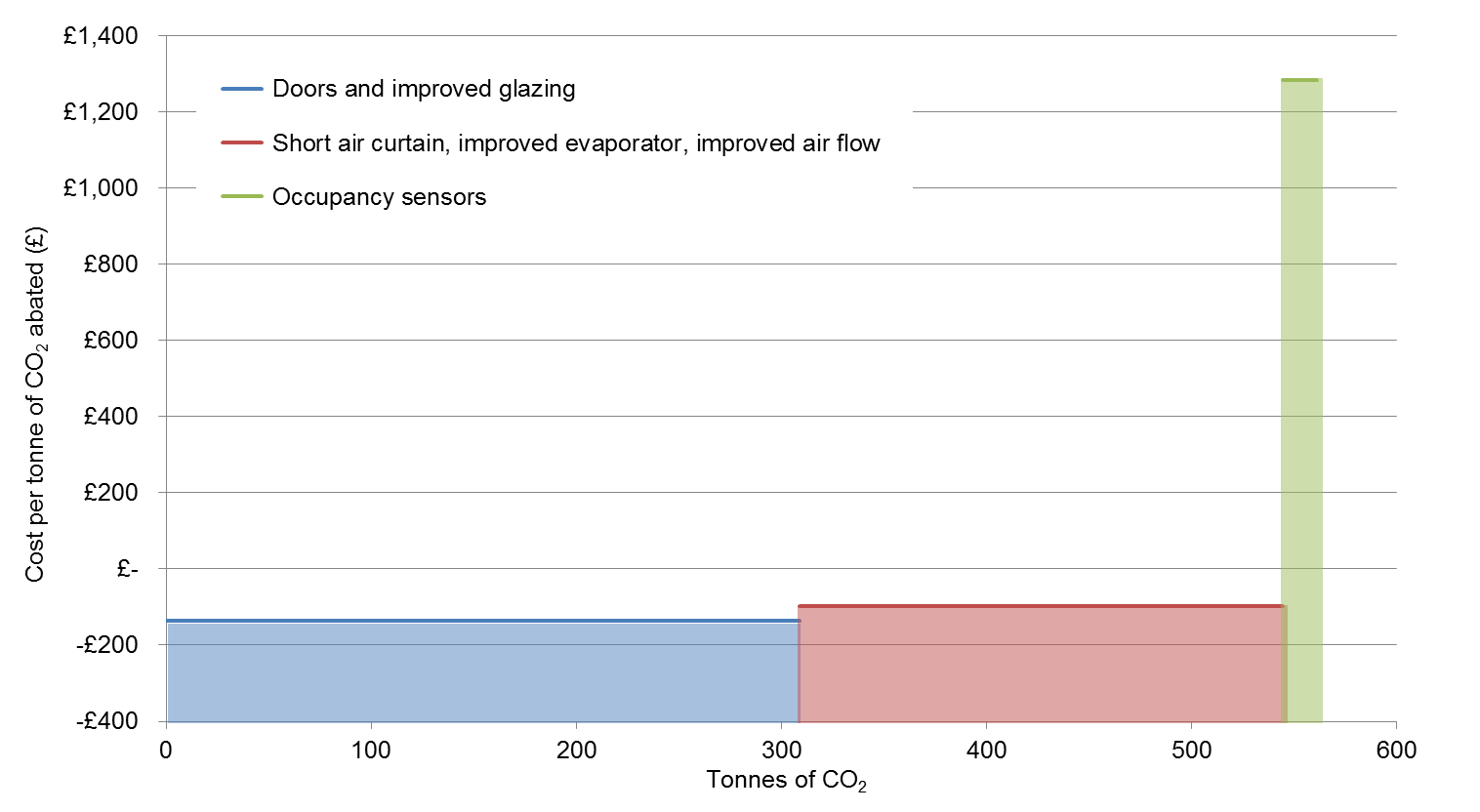


Figure . MAC curve for selected cabinet technologies in baseline store.

#### Technologies suggested for refrigeration system in the baseline store

The most attractive technology (in terms of carbon savings) for the refrigeration system was the replacement of the current refrigerant (R404A) with a low GWP HFO refrigerant. At the same time the refrigeration plant could be recommissioned. This then eliminated some other technologies as they were not compatible with the technology (e.g. other replacement refrigerants or refrigeration systems). Further technologies could also be added such as floating head pressure controls, suction pressure controls, liquid pressure amplification and motor efficiency controls. A decision flow chart showing the technology selection process is shown in Figure 27. The final MAC graph is shown in Figure 28. The technologies showed a negative abatement cost or were cost neutral per tonne of carbon abated.

Figure . Options for selection of refrigeration system technologies in baseline store.

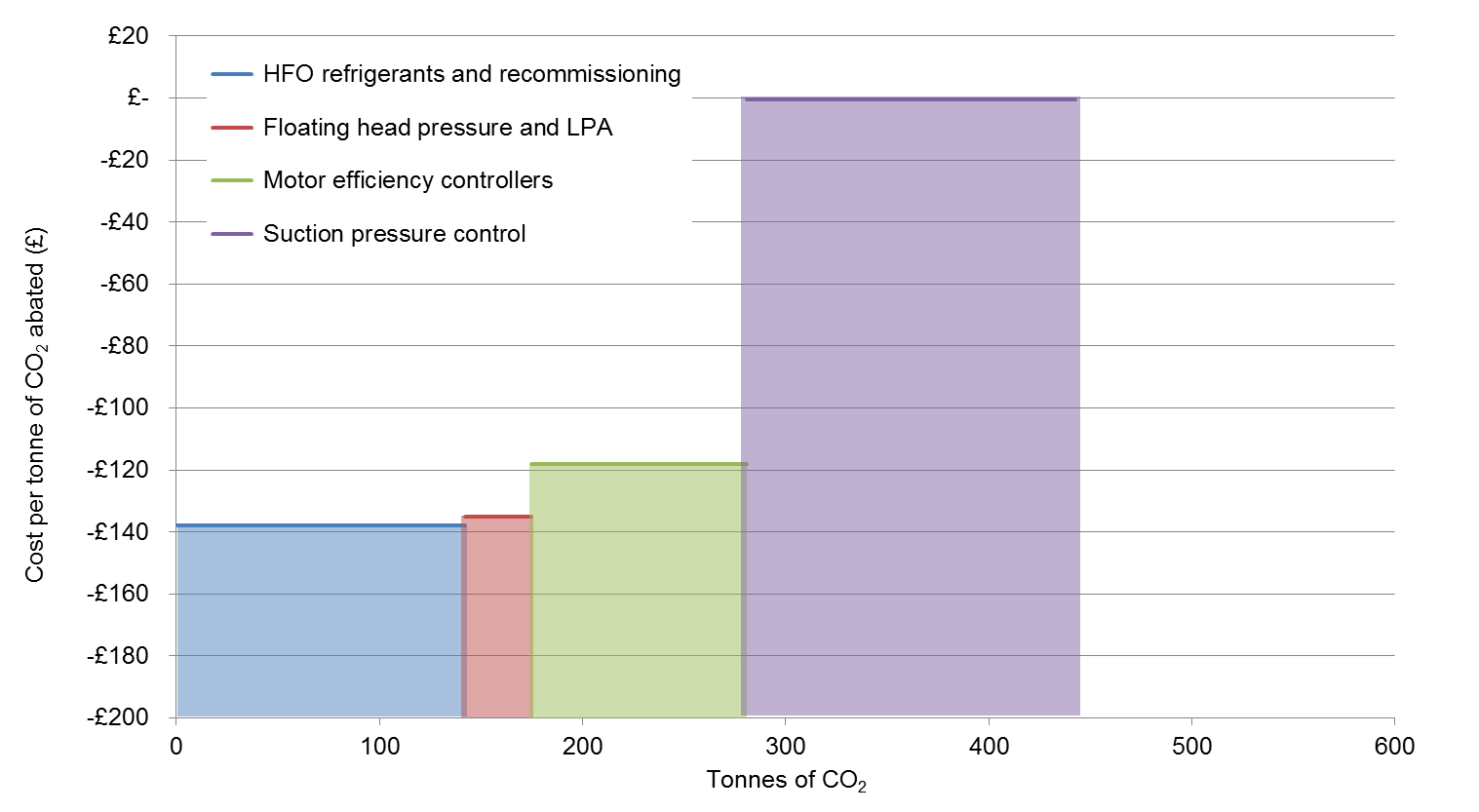


Figure . MAC curve for selected refrigeration system technologies in baseline store.

#### Options for baseline supermarket

In terms of abatement cost per tonne of carbon, the options to move to an HFO refrigerant with associated recommissioning and to add doors to all cabinets were predicted to be the most effective options for the baseline store. Adding doors abated 309 tonnes of carbon over 3 years with the supermarket, saving £136/tonne of carbon abated. Changing to a HFO refrigerant and recommissioning had almost identical carbon abatement costs (-£138/tonne of carbon) but the total quantity of carbon abated was less than that for adding doors at 142 tonnes over the 3-year period. The application of the cabinet based technologies greatly increased the carbon abated to 564 tonnes over the 3-year period with the short air curtains, enhanced evaporator and improved air flow options providing a negative abatement cost. All the refrigeration system options produced negative abatement costs and saved a total of 443 tonnes of carbon over the 3-year period.

Over the 3-year period, the cabinet and refrigeration system technologies were predicted to save 53% of the supermarket carbon emissions. Options to further reduce carbon would be available in a new or redesigned store where additional/alternative technological options could be applied.