Technology Report
Industrial refrigeration and chilled glycol and water applications
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This report was prepared on behalf of the Office of Environment and Heritage by Minus40, and is based on calculations and energy modelling conducted in mid 2010. Modelling was based on conventional single and two stage industrial refrigeration systems. Arbitrary load profiles, representative of common industrial refrigeration applications, were used to represent refrigeration plant duty. Ambient weather conditions were factored into the modelling and were based on records obtained from the Australian Bureau of Meteorology. Energy saving technologies were modelled as variants of the conventional single or two stage refrigeration plants.
Executive summary

Industrial refrigeration plants are substantial energy users, yet often little consideration is given to a plant’s energy efficiency and operating costs or its environmental impact.

Other criteria like functionality and construction costs are frequently given higher priority. The energy costs of a refrigeration plant could be reduced by around 40 per cent, possibly more, through adoption of the most energy efficient equipment and techniques. A total of 15 energy-saving technologies have been identified in this report, and while many require capital outlays, a series of verified case studies in the report indicate the payback period is often less than three years. With rising energy costs and growing environmental awareness, there is a growing need to reduce power consumption in the industrial refrigeration sector in NSW.

This report outlines 15 energy saving technologies available to increase the energy efficiency of an industrial refrigeration plant. Where possible, for each technology the annual energy savings, capital costs and payback periods have been estimated by considering examples. These technologies are as follows:

Industrial refrigeration specific technologies
1. Variable plant pressure control
2. Automated compressor staging and capacity control
3. Remote control optimisation of refrigeration plant
4. Heat recovery from discharge gas and oil cooling
5. Defrost management
6. Variable cold store temperatures
7. Variable evaporator fan speeds
8. Condensate sub-cooling
9. Refrigeration plant design review
10. Plant condition maintenance – removal of air and water

Chilled glycol and water specific technologies
11. Chiller efficiency – full and part load
12. Chilled water/glycol circuit design and control
13. Heat recovery from chillers and chiller/heat pumps units
14. Variable chilled fluid temperatures
15. Variable cooling water temperatures

Other technologies
Apart from the above mentioned energy saving technologies, several other site-specific energy saving projects may be possible. Such projects include, but are not limited to the following:
• installing high stage heat pumps to generate large amounts of hot water at between 60°C and 65°C, which is not possible with the heat recovery techniques described in this report
• insulation upgrades.

These other technologies are not examined in this report.

Extent of possible savings
Considerable energy savings are possible even on a partly optimised plant by reviewing the control logic and conducting a thorough design review. These energy saving technologies are broadly applicable and most involve control modifications that can be implemented in the plant’s computer software. Some can also be continuously monitored and optimised remotely. The various energy saving technologies highlighted are applicable to most industrial refrigeration facilities. On conventional plants without these energy saving measures, the energy savings could be considerable and could reduce refrigeration related energy consumption by more than 40 per cent. Many of these technologies comprise control related upgrades and are best implemented together to maximise energy savings.
Energy modelling has been conducted for industrial refrigeration specific technologies (1-10) considering typical examples. The results of the modelling are detailed below:

### Industrial refrigeration specific technologies

<table>
<thead>
<tr>
<th>Energy saving technology</th>
<th>Partly optimised plant (Applicable to most plants)</th>
<th>Un-optimised plant (Energy inefficient plants)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Power consumption after implementation (%)</td>
<td>Power consumption after implementation (%)</td>
</tr>
<tr>
<td>1  Variable plant pressure control</td>
<td>3%</td>
<td>97%</td>
</tr>
<tr>
<td>2  Automated compressor staging and capacity control</td>
<td>5%</td>
<td>95%</td>
</tr>
<tr>
<td>3  Remote optimisation of refrigeration plant</td>
<td>5%</td>
<td>95%</td>
</tr>
<tr>
<td>4  Heat recovery¹</td>
<td>0%</td>
<td>100%</td>
</tr>
<tr>
<td>5  Defrost management</td>
<td>2%</td>
<td>98%</td>
</tr>
<tr>
<td>6  Variable room temperatures²,³</td>
<td>0%</td>
<td>100%</td>
</tr>
<tr>
<td>7  Variable evaporator fan speeds</td>
<td>0%</td>
<td>100%</td>
</tr>
<tr>
<td>8  Condensate sub-cooling</td>
<td>2%</td>
<td>98%</td>
</tr>
<tr>
<td>9  Plant design review</td>
<td>0%</td>
<td>100%</td>
</tr>
</tbody>
</table>

**Total power consumption of refrigeration plant (% of current)**  
82.4%  
54.3%

**Total potential % energy savings**  
17.6%  
45.7%

Indicative savings for chilled glycol and water specific technologies (11-15) are detailed below:

### Chilled glycol and water specific technologies

<table>
<thead>
<tr>
<th>Energy saving technology</th>
<th>Typical industrial plant</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Savings* (%)</td>
</tr>
<tr>
<td>11 Chiller efficiency – full and part load</td>
<td>10-50%</td>
</tr>
<tr>
<td>12 Chilled water circuit design and control</td>
<td>5-20%</td>
</tr>
<tr>
<td>13 Heat recovery from chillers and chiller – heat pump units</td>
<td>Heating cost offset</td>
</tr>
<tr>
<td>14 Variable chilled fluid temperatures</td>
<td>0-20%</td>
</tr>
<tr>
<td>15 Variable cooling water temperatures</td>
<td>5-15%</td>
</tr>
</tbody>
</table>

¹. Heat recovery does not necessarily reduce power consumption, but can reduce the consumption of other energy sources (gas, oil, coal, etc).
². Variable room temperature strategies do not necessarily reduce power consumption, but can reduce the power costs through load shifting and reduce demand costs.
³. In the case of an un-optimised plants, savings on power consumption of the refrigeration plant serving the cold stores alone, not on power consumption of the entire refrigeration plant.
⁴. Savings for technologies 11-15 are indicative.
The table below gives an indication of the applicability of the technologies discussed in this report to typical applications within different industry sectors. Specific sites may differ from industry norm, and therefore perhaps more or fewer technologies could be applicable in each case.

<table>
<thead>
<tr>
<th>Technology applicability by industry sector</th>
</tr>
</thead>
<tbody>
<tr>
<td>Abattoirs and poultry processors</td>
</tr>
<tr>
<td>Bakeries</td>
</tr>
<tr>
<td>Breweries</td>
</tr>
<tr>
<td>Cold storage</td>
</tr>
<tr>
<td>Dairy processors</td>
</tr>
<tr>
<td>Food and beverage</td>
</tr>
<tr>
<td>Meat packers and processors</td>
</tr>
<tr>
<td>Pet food manufacturers</td>
</tr>
<tr>
<td>Plastics and packaging</td>
</tr>
<tr>
<td>Wineries</td>
</tr>
</tbody>
</table>
Introduction

The Office of Environment and Heritage has established Energy Saver industrial refrigeration which aims to promote the widespread adoption of cost-effective, commercially proven and energy efficient technologies throughout the industrial refrigeration sector in NSW.

Typical facilities in this sector in NSW include abattoirs, bakeries, cold stores, dairy processors, food and beverage manufacturers, meat packers and processors, pet food manufacturers and wineries.

Energy Saver helps organisations identify the most cost-effective energy efficient industrial refrigeration technology upgrades and provides support for each phase of the implementation process:

1. Identify the energy savings projects applicable to the particular facility
2. Generate a business case i.e. conduct plant monitoring, calculate energy savings and estimate project costs, so as to demonstrate to the end user the viability of implementing the various energy saving technologies
3. Prepare and present a report on the findings
4. Prepare a technical specification including hardware description, functional description of controls and associated drawings
5. Invite tenders and select contractor
6. Project management of upgrade or modification
7. Commissioning
8. Measurement and verification of energy savings
9. Operations and maintenance training to ensure on-going optimum operation of plant
10. On-going measurement of energy and cost savings.

This report provides detailed technical analysis of the 15 industrial refrigeration and chilled glycol and water technologies, identified on page 5. It provides energy savings, capital costs and payback periods estimated where possible by considering examples.
Technology 1: Variable plant pressure control

Variable plant pressure control is a strategy which aims to improve a plant’s energy efficiency by optimising both the head pressure and the intermediate stage (inter-stage) pressure of the refrigeration plant, based on instantaneous plant load and ambient conditions.

Principles

Variable head pressure control

Many industrial refrigeration plants are run on a “set and forget” basis, particularly where the plant’s head pressure is concerned. The lack of flexibility in setting the head pressure leads to unnecessary energy consumption.

The head pressure of a refrigeration plant is the discharge pressure of the high stage compressors, and is slightly higher or equal to the pressure at which the refrigerant condenses. In a conventional plant, head pressure is fixed and the plant control system attempts to maintain that fixed value.

If the head pressure set point is too low for a given ambient temperature and plant load condition, the condensers will reach capacity and the head pressure will fluctuate with load, potentially causing temperature fluctuations in the plant. If the head pressure set point is too high, there is an increase in compressor power consumption.

Variable head pressure control (VHPC) aims to optimise the head pressure of a refrigeration plant at any given time while taking into account operational factors such as minimum compression ratios and oil separation as well as variables such as ambient conditions and plant load. When head pressure is optimised, the combined power consumption of the high-stage compressor and the condenser fan is minimised.

The condensers are designed to cope with the worst possible conditions in terms of plant load and wet bulb temperatures therefore, for the greater part of the year, the plant condenser(s) is subject to lower plant loads and wet bulb temperatures. This means that, for most of the year, the condensers are oversized for the immediate task and condensing pressures can, in turn, be reduced to lower power consumption.

Furthermore, ambient wet bulb temperature is generally stable for long periods of the day and tends to fluctuate only with a change of weather. As VHPC depends on wet bulb temperature and plant load, a well-defined VHPC logic would dampen head pressure fluctuations. Head pressure on an otherwise conventional setup tends to fluctuate with plant load. Therefore, VHPC, in addition to reducing head pressure when possible, also stabilises the head pressure of the plant, resulting in more efficient and steadier plant operation over the year.

Plant head pressure is often deliberately raised to facilitate hot gas evaporator defrosts. This approach is inherently inefficient as overall performance of the plant is penalised to facilitate a relatively infrequent and minor function. The simple alternative would be to dedicate a single compressor in the facility as a “defrost compressor”. It would run at a higher discharge pressure to the remaining compressors and would equalise (in pressure) with these compressors via a pressure regulator mounted in the discharge line. This allows the plant to operate VHPC logic, whilst allowing a hot gas defrost line to be connected upstream of the pressure-regulating valve and fed to the plant as required.

Inter-stage pressure control

As with head pressure, the inter-stage pressure of a plant can also affect the energy consumption of a plant.

The inter-stage pressure of a refrigerant plant is the intermediate pressure between the low stage and high stage compressors. The optimal inter-stage pressure will vary according to plant load and head pressure.

Inter-stage pressure control (ISPC) aims to optimise the inter-stage pressure of a refrigeration plant in step with variations in head pressure. On a plant where the head pressure is variable, the inter-stage pressure should also be varied to optimise the balance between low stage and high stage power consumption.

ISPC is only possible where plant inter-stage temperature can be varied. If the inter-stage vessel is used to provide refrigeration to other plant applications, such as cool rooms or glycol/water chilling, then the inter-stage pressure may have to be maintained at a fixed value or varied only within a narrow range.

Plant energy savings of up to 12%
Plant and equipment requirements

To implement variable plant pressure control, the following information and plant equipment are required:

### Equipment requirements

- Ambient dry bulb temperature and relative humidity sensor
- Discharge and inter-stage pressure transmitters
- For screw compressors: slide valve potentiometers connected to the main plant controller
- For reciprocating compressors: capacity control solenoids connected to the main plant controller
- Variable speed drives (VSDs) for each condenser fan
- Defrost compressor: one of the existing compressors in the plant can be dedicated to serve as a defrost compressor, if required
- Sufficient control system hardware and software capability to define the logic.

### Information requirements

#### Head pressure control:

- number of compressors
- type of compressor – screw or reciprocating
- make and model of each compressor
- number of condensers
- type of condenser – air cooled, water cooled or evaporative and
- age of the condenser – to allow for sufficient condenser de-rating on a plant with old condensers.

#### Inter-stage pressure control:

- current plant operating suction, inter-stage and discharge pressures and
- plant load status.

### Achievable savings

#### Variable head pressure control

Modelling suggests that, compared to a system with fixed head pressure settings, VHPC can generate average annual energy savings as set out below:

- **Fixed head pressure = 25°C:** 9% to 12% of high stage compressor power consumption
- **Fixed head pressure = 30°C:** 20% to 25% of high stage compressor power consumption
- **Fixed head pressure = 35°C:** 30% to 35% of high stage compressor power consumption

**NB:** These figures are based on modelling of three typical applications as explained below.

**NB:** These indicated savings are only achievable if the control logic is suitably optimised, otherwise savings may have to be discounted off the above-mentioned figures.

Figure 1 illustrates the annual energy saving profiles for a cold store running at 25°C, 30°C and 35°C fixed head pressures for 24 hours a day, seven days a week. Profiles for production facilities operating 16 hours a day, five days a week and 16 hours a day, seven days a week are similar in trend and magnitude to Figure 1.

Figure 2 illustrates the energy savings possible with variable head pressure control for various applications when compared to a system with fixed condensing temperatures of 25°C, 30°C or 35°C.

#### Annual energy savings for following three typical operating regimes have been modelled.

- **A cold store with a variable load profile.** Plant load varies in a sinusoid between 60% and 100% of maximum plant capacity based on the time of the day.
- **A 16/7 production schedule which operates at 100% of maximum plant capacity for two shifts everyday and at 30% at other times.** This is representative of both production and storage on such sites.
- **A 16/5 production schedule that operates at 100% plant capacity for two shifts every week day and at 50% at other times.** This is representative of both production and storage on such sites.

The following assumptions have been made for each operating regime:

- Refrigerant used – ammonia
- Ambient conditions – annual weather conditions for Sydney
- Condenser/ambient wet bulb approach (temperature difference)
  - Linear with 10°C approach @ 90% plant load and 3°C approach @ 10% plant load
  - Minimum condensing temperature = 20°C, factoring in practical considerations.

The following methodology was used to calculate annual savings for each regime over one year.

1. The condensing temperature was calculated for every hour of the year using a derived load profile and ambient wet bulb temperature data.
2. The efficiency of the system with variable head pressure control was then calculated based on the condensing temperature and a fixed suction temperature of the system.
3. The coefficient of performance (COP or efficiency) of the conventional system with a fixed head pressure setting was then calculated. Each of the regimes was run three times, using condensing temperatures fixed at 25°C, 30°C and 35°C.
4. For the 25°C scenario, the system attempts to maintain 25°C condensing temperature as far as the load and ambient conditions permit. As the load and ambient wet bulb temperature increase, the condensing temperature increases correspondingly. Below 25°C, savings are minimal as both systems run at similar head pressures. The savings due to variable head pressure control are possible between condensing temperatures of 20°C and 25°C, which is mainly during the cold winter months.
5. The same argument is true for scenarios of 30°C and 35°C. Savings increase as the fixed head pressure setting increases, because the variable head pressure control can run below the fixed head pressure control set-point for a greater portion of the year.
The achievable savings depend on the following:

i. **The application**: variable production cycles, such as batch cycles and cold storage applications have variable loads and, hence, greater savings potential compared to plants with fixed loads i.e. facilities having a constant throughput profile.

ii. **Heat rejection equipment in the plant**: if condensers are over-sized, energy savings are achieved by running lower head pressure and lower fan speed where possible.

iii. **Oil separator design**: when a plant runs at lower head pressures than design, the discharge gas velocities increase and hence the velocity through the oil separator increases. At a certain point, the gas velocity could be so high that the oil is carried over with the discharge gas and accumulates in the low-pressure vessel of the plant. This will result in excessive oil consumption and increased maintenance requirements. Hence, a plant with over-sized oil separators can handle greater gas velocities, thus permitting lower head pressures and greater energy savings as compared to plants with smaller oil separators.

### Savings due to variable head pressure control for a typical cold store application

![Figure 1: Annual energy saving profile for a cold store application](image)

### Savings due to variable head pressure control for various typical applications

![Figure 2: Average annual energy savings for various applications](image)
**Inter stage pressure control**

For a plant with a suction temperature of -30°C and a condensing temperature of 35°C, the graph in figure 3 illustrates the relationship between achievable savings and the inter-stage temperature/pressure.

**Energy savings due to varying inter stage pressure**

![Graph showing energy savings due to varying inter stage pressure](image)

**Figure 3: Savings associated with varying inter stage pressure on a 2 stage plant**

Figure 3 illustrates the effect of varying inter-stage pressure on the efficiency of a refrigeration plant. Results are compared against a plant with an inter-stage temperature of -10°C. The optimum inter-stage pressure of this system corresponds to a saturation temperature of -4°C. At an inter-stage temperature above -4°C, the increase in absorbed power of the low stage system is greater than the decrease in absorbed power of the high stage system. Conversely, at an inter-stage temperature below -4°C, the increase in absorbed power of the high stage system is greater than the decrease in absorbed power of the low stage system.
Estimated financial returns

Variable head pressure control

The capital costs of implementing variable head pressure control depend on the following:

i. number and size of variable speed drives required; this depends on the number of condensers or cooling towers in the plant and their respective fan motor sizes

ii. location of variable speed drives relative to the fan motors; if the distance is great for practical reasons, capital costs would increase due to the need for greater quantities of shielded cabling.

The following example has been considered for estimating energy savings:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type of application</td>
<td>Cold store only</td>
</tr>
<tr>
<td>High stage refrigeration load</td>
<td>1000 kW</td>
</tr>
<tr>
<td>High stage absorbed power (-10°C SST; 35°C SCT)</td>
<td>250 kW</td>
</tr>
<tr>
<td>Design condensing temperature</td>
<td>35°C</td>
</tr>
<tr>
<td>Design ambient wet bulb temperature</td>
<td>24°C</td>
</tr>
<tr>
<td>Number of evaporative condensers</td>
<td>2</td>
</tr>
<tr>
<td>Fan motor capacity per condenser</td>
<td>15 kW</td>
</tr>
<tr>
<td>Average power cost paid by facility</td>
<td>$0.10 per kWh</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Item</th>
<th>Estimated costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment</td>
<td>$30,000</td>
</tr>
<tr>
<td>Labour</td>
<td>$8,000</td>
</tr>
<tr>
<td>Engineering</td>
<td>$6,000</td>
</tr>
<tr>
<td>Programming</td>
<td>$6,000</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>$50,000</strong></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Condensing temperature set point for fixed head pressure system [°C]</th>
<th>Energy consumption for fixed head pressure [kWh/year]</th>
<th>Energy consumption for variable head pressure [kWh/year]</th>
<th>Energy savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
<th>Payback [years]</th>
</tr>
</thead>
<tbody>
<tr>
<td>25°C</td>
<td>1,382,000</td>
<td>1,250,000</td>
<td>132,000</td>
<td>$13,200</td>
<td>$50,000</td>
<td>3.78</td>
</tr>
<tr>
<td>30°C</td>
<td>1,584,000</td>
<td>1,250,000</td>
<td>334,000</td>
<td>$33,400</td>
<td>$50,000</td>
<td>1.50</td>
</tr>
<tr>
<td>35°C</td>
<td>1,820,000</td>
<td>1,250,000</td>
<td>570,000</td>
<td>$57,000</td>
<td>$50,000</td>
<td>0.88</td>
</tr>
</tbody>
</table>

1. Equipment allowed in the above costing includes two variable speed drives (15 kW each), ambient wet bulb temperature sensor, shielded cabling, electrical wiring and enclosures for VSDs (if required).

Inter-stage pressure control

Generally the costs involved in relation to the implementation of variable inter-stage pressure logic involve engineering and programming costs only, and savings achievable depend on the degree to which inter-stage pressures can be varied on the specific plant.

Further optimisation

For best results, the variable plant pressure control logic should be optimised over the full range of the plant operating conditions. Generally this requires observation of the plant under a range of load and climatic conditions, and the fine tuning of various parameters. Refer to TECHNOLOGY 3: Remote Optimisation of Refrigeration Plant.
Most large scale industrial refrigeration plants use several compressors. In most cases, the system controlling the compressors maintains and controls capacity without necessarily optimising efficiency.

It is also common for screw compressors to unload using slide valve control and, as a result, they are inefficient when operating at part load. It is common in large scale industrial refrigeration plants to see multiple screw compressors running at part load and, therefore, inefficiently.

**Principles**

**Slide valve vs. variable speed control**

The slide valves used on industrial screw compressors to reduce cooling capacity are notoriously inefficient. This is true whether the slide valve opens in a continuous manner or in stages, such as 100%, 75%, 50% or 25%. Slide position is approximately representative of the refrigeration capacity of the compressor.

As indicated in Figure 4, with slide control, the reduction in refrigeration capacity of a screw compressor relative to power consumption is disproportionate. For example, at 30% slide position, the refrigeration capacity of a screw compressor is approximately 40% whereas the power consumption is excessive at approximately 60%. The alternative to slide valve control is to use a variable speed drive (VSD) to modulate the capacity of a compressor. Also, as indicated in Figure 4, the reduction in refrigeration capacity relative to power consumption is essentially equal, particularly between 40% and 100% refrigeration capacity.

Figure 4 represents the potential energy savings achievable between a slide valve and VSD control.

**Relation between % capacity and % power consumption**

![Figure 4: Slide valve and VSD control disparity](image-url)
Compressor staging and capacity control logic

On an industrial refrigeration plant using multiple screw compressors, implementing careful compressor staging and capacity control can make considerable energy savings. Capacity control can be achieved either by variable speed control or by efficient slide valve control. Efficient slide valve control refers to the active control of the slide valve between 75% and 100%, the efficient operation range for a screw compressor. On reciprocating compressors, capacity control is achieved by active control of the cylinders.

An automated compressor staging system takes into account the various compressors in the refrigeration plant and progressively turns them ON (during increasing load) or OFF (during decreasing load) based on suction pressure.

Critical consideration should be given to running the compressors as efficiently as possible, that is:

i. running variable speed-controlled screw compressors at 75% to 100% slide valve position and speed control down to a minimum of 50% speed
ii. running the slide-controlled screw compressors between 75% and 100% slide.

Figures 5 and 6 show an example of a refrigeration plant with two screw compressors, one of which is variable speed controlled.

Loading sequence on a two-compressor plant

Four modes of operation are defined for this plant. During increasing load, C1 is first slide controlled (mode A) until it reaches 100% slide position at 25 Hz (min speed), after which its speed is increased between 25 and 50 Hz (mode B). When C1 is at 50 Hz and the load is still increasing, C2 is turned ON and immediately loaded to 75% slide position. At this time C1 has to reduce its speed due to the sudden increase in capacity. C2 is then slide-controlled between 75 and 100% (mode C) after which C1’s speed can be increased again between 25 and 50 Hz (mode D).
During decreasing load, the reverse logic is followed. The control program must include sufficient “dead-bands” (temperature ranges where no change is made) to be programmed to avoid continuous changing of modes and short cycling of compressors.

On a large industrial refrigeration plant with multiple compressors, following the above logic could be complicated and an “unused swept volume” analysis could be better suited.

Additionally, an automated compressor staging program for a large industrial plant with multiple operating compressors should include a weekly or monthly rotation sequence so that all compressors would run for an equal number of hours. If the plant has one compressor on a VSD, it would be the base load machine regardless of its operating hours. If the plant has some compressors that are worn or aged or intended to be retained only as back-up, they would always be last in the rotation sequence and hence to operate only during times of high plant load.

Equipment requirements
To implement automated compressor staging and capacity control logic, the following equipment is required:

i. a suction pressure transmitter
ii. a slide valve potentiometer for each screw compressor, connected to the main plant controller
iii. capacity control solenoids for reciprocating compressors, connected to the main plant controller
iv. variable speed drive(s) on selected compressors
v. sufficient control hardware and software capability to define the logic.

Plant benefits
An optimised and well defined compressor staging and capacity control system:

i. promotes efficient operation by active slide control (75% to 100%) on screw compressors, active cylinder control on reciprocating compressors and active speed control on VSD-enabled compressors, thus saving considerable amounts of energy. Installing a VSD also provides a “soft start” feature for the compressor, thus preventing power spikes during start-up and giving the motor a longer operating life
ii. prevents several compressors all operating at part load, the major cause of inefficiencies on conventional industrial refrigeration plants
iii. prevents the compressors working on a short cycle by defining appropriate dead-bands during various modes of operation
iv. stabilises plant suction pressure and
v. rotates compressors on a regular basis (weekly/monthly) to share base load and hence generate equal run hours on all compressors.

Achievable savings
To calculate the savings achievable, a single stage refrigeration plant with two compressors using this technology was modelled against the same plant using conventional compressor staging control. Presented in the table below are:

- the energy savings achievable with instantaneous plant load
- the total annual energy savings achievable, by aggregating instantaneous energy savings over the annual plant load profile presented in APPENDIX I: Usage Profile Considered for Energy Savings Calculations.
<table>
<thead>
<tr>
<th>Plant load</th>
<th>Energy saving</th>
</tr>
</thead>
<tbody>
<tr>
<td>10%</td>
<td>40%</td>
</tr>
<tr>
<td>20%</td>
<td>33%</td>
</tr>
<tr>
<td>30%</td>
<td>24%</td>
</tr>
<tr>
<td>40%</td>
<td>41%</td>
</tr>
<tr>
<td>50%</td>
<td>32%</td>
</tr>
<tr>
<td>60%</td>
<td>14%</td>
</tr>
<tr>
<td>70%</td>
<td>6%</td>
</tr>
<tr>
<td>80%</td>
<td>5%</td>
</tr>
<tr>
<td>90%</td>
<td>3%</td>
</tr>
<tr>
<td>100%</td>
<td>0%</td>
</tr>
</tbody>
</table>

**Total annual energy saving** 24%

Over a typical year, the following methodology was used to calculate annual savings:

i. the lead compressor is variable speed controlled
ii. the lag compressor is slide valve-controlled between 75% and 100%
iii. the comparison is done against a plant where the compressors have no variable speed drives or proper slide valve control. This is based on observations in real time on a certain plant
iv. the observations indicated that the lag compressor would turn on when the lead compressor was at 75% load or higher. Hence the calculations have been conducted on this basis
v. the usage profile as presented in APPENDIX I was used for the purpose of this analysis.

Potential savings achievable by automated compressor staging and control depend on the following:

i. the load profile of the plant
ii. the number, size and condition of compressors in the plant.

The indicated savings can be realised if the control logic is optimised for the full range of expected operating conditions. Where static implementation of the logic is applied, these savings estimates may need to be discounted, typically by 30% to 40% of the abovementioned figures. Remote optimisation can achieve savings close to the predicted levels. Refer to TECHNOLOGY 3: Remote Optimisation of Refrigeration Plant.
Estimated financial returns

Capital costs to implement compressor staging and capacity control logic depend on the following:

i. the number of compressors in the plant
ii. the number of compressors that need to be equipped with VSDs.

For the example considered and modelled above, typical capital costs could be as follows:

<table>
<thead>
<tr>
<th>Item</th>
<th>Estimated cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment</td>
<td>$78,000</td>
</tr>
<tr>
<td>Labour</td>
<td>$10,000</td>
</tr>
<tr>
<td>Engineering</td>
<td>$6,000</td>
</tr>
<tr>
<td>Programming</td>
<td>$6,000</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>$100,000</strong></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Energy consumption of conventional system [kWh/year]</th>
<th>Energy consumption of improved system [kWh/year]</th>
<th>Energy savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
<th>Payback [years]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2,135,000</td>
<td>1,628,000</td>
<td>507,000</td>
<td><strong>$50,700</strong></td>
<td>$100,000</td>
<td>1.97</td>
</tr>
</tbody>
</table>

1. Equipment allowed in the above costing includes a 315 kW VSD, two slide valve potentiometers, shielded cabling, electrical wiring, enclosure for VSD and additional PLC hardware.
2. Average cost of power paid by facility is considered as $0.10 per kWh.

Further optimisation

For best results, the variable plant pressure control logic should be optimised over the full range of the plant operating conditions. Generally this requires observation of the plant under a range of load and climatic conditions, and the fine tuning of various parameters. Refer to TECHNOLOGY 3: Remote Optimisation of Refrigeration Plant.
As a rule, such optimisation requires that the performance of the control logic be continuously monitored and the various control parameters fine tuned over an extended period so as to capture variations in production and seasonal conditions. Manual monitoring and fine tuning can be a time consuming and, hence, cost-intensive process if conducted at local level, especially for regional sites. Where the logic is implemented on the plant’s computer, “static optimisation” – optimisation under one set of conditions at the time of implementation – is commonly applied. In most cases, static implementation does not achieve the best savings over the full range of production and seasonal variations so, to a certain degree, discounting of the predicted savings is required. Remote optimisation of the control logic, incorporating appropriate communication with site personnel, is generally a more practical and cost effective way to maximise the benefits of any advanced control logic.

Principles
Conventionally, all control logic applied to a refrigeration plant is implemented within the main plant computer. Changes to the program generally require that the software on the computer is downloaded, modified and uploaded back onto the computer. Whilst this can be done remotely through internet or other means of access, the effort involved is significant and does not generally lend itself to the ongoing refinement of the system. Another approach would be to limit local control to the essential refrigeration plant functions. Another system, based on a personal computer, would then be installed to enable remote management of the local control system. This control management system would:

- access information on the local system
- contain the intelligent control logic
- override the local system as needed to enhance energy savings.

The control software can easily be accessed remotely, for example via a 3G connection, and the operation of the plant observed and analysed. Adjustments to the control logic are applied without affecting the plant or the local system, and the effects of any changes observed. The simplicity and convenience of this approach makes it feasible and practical to fine tune the control logic over an extended period and under different seasonal and production conditions. This is often referred to as remote optimisation.

Remote optimisation of the control logic, incorporating appropriate communication with site personnel, is generally a more practical and cost effective way to maximise the benefits of any advanced control logic.

Equipment requirements
The following minimum equipment is required for remote optimisation of a refrigeration plant:

i. a separate computer with software that links with the main plant control system. All the energy optimisation programming is conducted within software on the server and over-rides programmed set-points while in operation. The user would have the ability to turn the server module off if required, at which time the system returns to its original basic mode of operation as programmed at the plant
ii. a remote workstation that communicates with the server via 3G network. All parameter observations and optimisation is carried out at the remote workstation
iii. a prerequisite to the installation of a remote optimisation system is a modern, high level computer installed at the plant.
There are currently several vendors offering systems with remote optimisation capability to the market. Figure 7 illustrates the setup of a remote optimisation system.

**Remote optimisation system**

![Diagram of a remote optimisation system](image)

**Plant benefits**

**Installation of a remote optimisation system:**

i. achieves optimum operation of the plant and thus improves energy efficiency to the maximum extent possible

ii. assists in quantifying energy savings. If energy consumption data prior to implementing the project is not available, the remote optimisation system can simply be turned off, energy performance logged, turned back on, energy performance logged again and the results analysed to quantify the savings. Hence most of the energy saving technologies mentioned in this document can be remotely monitored and their savings accurately quantified

iii. allows plant engineers to obtain plant performance reports via the control management system (CMS), without the need to download and process data from the plant’s supervisory control and data acquisition computer supervisory control and data acquisition system (SCADA).

**Achievable savings**

It is estimated that around 30% of the predicted savings of some control techniques are achievable only through remote optimisation, whilst 70% could be achieved through static implementation at plant level. Exact proportions will depend on the nature of the refrigeration plant and operating conditions. Hence implementation of a remote optimisation system can generate annual savings including:

i. additional energy savings due to remote optimisation of variable head pressure control logic. This could be around 3%

ii. additional energy savings due to remote optimisation of compressor staging and capacity control logic. This could be between 3% and 5%

iii. additional energy savings due to remote optimisation of variable cold store temperature control logic and load shifting during peak periods. This could be between 1% and 3%

iv. energy savings due to remote optimisation of inter-stage pressure on a 2-stage refrigeration plant. This could be around 1%.

**Estimated capital costs**

A remote control optimisation system could cost between $35,000 and $60,000 depending on the specific plant and its size, and on the number of control modules implemented. It should be noted that the larger the size of the plant, the greater are the energy savings due to remote optimisation.
On screw compressors the two main sources of heat rejection are the discharge gas and the cooling of compressor oil via an oil cooling process. On reciprocating compressors heat is rejected primarily via the discharge gas. If left unexploited, these quantities of heat are rejected to the main heat rejection devices of the plant, i.e. the condensers.

Energy used to heat water, whether it is electricity, gas or other fuel, can be reduced by using heat recovery to preheat water to between 50°C and 60°C. This provides a considerable saving in energy consumption. It should be noted that subject to case-specific engineering studies, heat recovery is typically better suited to reciprocating compressors and natural refrigerants, given their inherent thermo-mechanical characteristics.

**Principles**

On screw compressors, optimum heat recovery is achieved by installing:

i. a common discharge gas de-superheating heat exchanger (de-superheater), either on the low stage or high stage discharge

ii. a secondary oil cooler in series with the existing oil cooler on each high stage compressor. Furthermore, the original oil cooler should be retained so that the critical function of oil cooling is not compromised and is independent of hot water demand.

Extra oil cooling cannot be applied to reciprocating compressors.

Cold water is first passed through the de-superheater where it is pre-heated and then passed into the oil cooler where it is heated further to between 50°C and 60°C. This water is then stored in insulated hot water tanks and used as required. Typical applications for water in this temperature range include, but are not limited to:

- domestic hot water:
- ‘washdown’ water for sanitisation purposes
- bottle warming or other process heating requirements
- use as pre-heated feedwater to a hot water generator to supply sterilisation and pasteurisation processes, where 80°C to 95°C water is required. By increasing the temperature of water entering a hot water generator, considerable energy savings are possible.

The installation of a de-superheating heat exchanger to handle discharge gas results in a pressure drop on the refrigerant side that penalises plant energy consumption. However, a correctly engineered de-superheater decreases the load on the main condensers, allowing the system to run at head pressures low enough to:

i. offset the effect of the pressure drop caused by the heat exchanger

ii. may even provide a net improvement in compressor efficiency.

This can often be achieved simply by locating the pressure sensor used for variable head pressure control (technology 1) upstream of the de-superheater, rather than at the condensers.

Installing the de-superheater on the low stage discharge provides electrical (kWh) and gas (GJ) savings as follows:

- electrical (kWh) savings flow from the removal of some of the discharge gas superheat, which means less heat gets through to the high stage and eventually to the condensers
- gas (GJ) savings flow from installation of de-superheater on the high stage gas discharge, which gives greater heat recovery effect and therefore greater gas savings. Power savings, however, are practically nil. The reason for the higher gas savings with the high stage de-superheater is due to the greater heat rejection load on the high stage than on the low stage and also due to the slightly elevated discharge temperature.

A site with relatively less demand for heat recovery and higher electrical tariffs ($/kWh) is better suited to adopting a low stage de-superheater. Conversely, a site with greater heat recovery demand for heat recovery and lower electrical tariffs ($/kWh) is better suited to adopting a high stage de-superheater. Installation of de-superheaters on both the low stage and high stage provides greater overall savings, albeit at greater capital cost.
Plant and equipment requirements

Equipment requirements

The following equipment is required to facilitate heat recovery by discharge gas and oil cooling:

i. discharge gas de-superheating heat exchanger: a shell and tube type is good as it facilitates easy mechanical cleaning of the tubes. Refrigerant would be on the shell side and water on the tube side. Plate heat exchangers can also be used, but chemical cleaning may be required due to fouling over time. Plate heat exchangers are less expensive and require less installation space relative to shell and tube heat exchangers of similar duty

ii. oil cooling heat exchanger; again, a shell and tube type is better, for the same reason as above. Oil would be on the shell side and water on the tube side

iii. insulated hot water tanks; the size of the tanks should reflect hot water requirements, insulation and construction materials (typically stainless) and associated capital costs. If there is already a hot water tank on site a combination of a hot tank and a cold tank is a robust solution, where cold town water is fed into the cold tank as make-up and hot water to the application is drawn from the top of the hot water tank

iv. water circulation pump with variable speed drives to circulate hot water between the tanks and the heat exchangers. The pump is controlled by a variable speed drive based on the hot water temperature at the outlet of the oil cooler

v. hot water circulation pumps to supply the application

vi. three-way mixing valves are optional and may be required to control the supply temperature to the application, especially in a facility which requires hot water at various temperatures.

Figures 8 and 9 illustrate various heat recovery arrangements:

Heat recovery system with low stage de-superheaters

Figure 8: Schematic of heat recovery system with low stage de-superheater
Information requirements

The following minimum information is required to implement a heat recovery system:

i. the number of compressors
ii. the make, model and type of each compressor (screw or reciprocating)
iii. the number of condensers
iv. the type of condenser (air cooled, water cooled or evaporative)
v. amount of water required through the year
vi. the heating equipment available in the system. For example, hot water generators, hot water tanks and pumps
vii. available space for equipment installation.

Plant benefits

i. Energy savings due to reduced direct water heating. The savings could be considerable on facilities using large amounts of hot water
ii. Installing a de-superheating heat exchanger reduces the load on the main heat rejection devices of the plant, enabling the system to run at a slightly reduced head pressure. This would offset the effect of refrigerant pressure drop through the heat exchanger, if the equipment is selected carefully
iii. Power saving due to reduced high stage compressor power consumption if a de-superheater heat exchanger is fitted to the low-stage discharge.

Achievable savings

Energy modelling at instantaneous plant loads indicated heat recovery of up to 20%. The total annual energy savings were calculated by aggregating the instantaneous energy savings over the annual plant load profile presented in APPENDIX I. Heat recoverable from the low-stage de-superheater and oil coolers of the screw compressors is around 16% of the low stage load. If a high stage de-superheater was used in place of a low stage de-superheater, the heat recovery is around 20% of the low stage load.

The achievable savings depend on:

i. the plant having continuous requirement for hot water such that discharge and oil cooling heat can be used continuously, as opposed to a plant requiring large quantities of water for a short period each day
ii. the load on the refrigeration system, as hot water production varies in direct proportion with this load
iii. the hot water storage capacity. A generous storage capacity increases savings at plants requiring large amounts of hot water.

The viability of the project also depends on the plant’s energy costs.

Figure 9: Schematic of heat recovery system with high stage de-superheater

NB: The above schematics are indicative of the concept. Industrial refrigeration systems usually contain multiple high stage compressors such that the solution would involve a common discharge gas de-superheater and a new oil cooler for each high stage compressor.
Estimated financial returns

Capital costs to implement heat recovery depend on the following:

i. the size of the de-superheater and oil coolers required
ii. the number of oil coolers required
iii. whether tanks are required
iv. the amount of pipe-work (ammonia, hot water) required.

The following examples have been considered to estimate gas and electrical power savings due to installation of de-superheaters and oil coolers:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low stage refrigeration load</td>
<td>500 kW</td>
</tr>
<tr>
<td>Low stage suction temperature</td>
<td>-35°C</td>
</tr>
<tr>
<td>Inter-stage temperature</td>
<td>-2°C</td>
</tr>
<tr>
<td>Condensing temperature</td>
<td>35°C</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Low stage de-superheater + high stage oil cooler</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item</td>
</tr>
<tr>
<td>Equipment</td>
</tr>
<tr>
<td>Labour</td>
</tr>
<tr>
<td>Engineering</td>
</tr>
<tr>
<td>Programming</td>
</tr>
<tr>
<td><strong>Total</strong></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas savings [GJ/year]</td>
<td>1,670</td>
</tr>
<tr>
<td>Gas cost savings [$/year]</td>
<td>$10,020</td>
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<tr>
<td>Power savings [kWh/year]</td>
<td>23,000</td>
</tr>
<tr>
<td>Power cost savings [$/year]</td>
<td>$2,300</td>
</tr>
<tr>
<td><strong>Total cost savings [$/year]</strong></td>
<td><strong>$12,320</strong></td>
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<tr>
<td>Project cost ($)</td>
<td>$42,000</td>
</tr>
<tr>
<td>Payback (years)</td>
<td>3.41</td>
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</table>

<table>
<thead>
<tr>
<th>High stage de-superheater + oil cooler</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item</td>
</tr>
<tr>
<td>Equipment</td>
</tr>
<tr>
<td>Labour</td>
</tr>
<tr>
<td>Engineering</td>
</tr>
<tr>
<td>Programming</td>
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<tr>
<td><strong>Total</strong></td>
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</table>

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Gas savings [GJ/year]</td>
<td>2,200</td>
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<tr>
<td>Gas cost savings [$/year]</td>
<td>$13,200</td>
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<tr>
<td>Power savings [kWh/year]</td>
<td>–</td>
</tr>
<tr>
<td>Power cost savings [$/year]</td>
<td>–</td>
</tr>
<tr>
<td><strong>Total cost savings [$/year]</strong></td>
<td><strong>$13,200</strong></td>
</tr>
<tr>
<td>Project cost ($)</td>
<td>$47,000</td>
</tr>
<tr>
<td>Payback (years)</td>
<td>3.56</td>
</tr>
</tbody>
</table>

1. The equipment allowed in the above costing does not include hot water tanks or field hot water pumps.
2. A single high stage compressor has been assumed for estimating the costs and hence a single oil cooler has been indicated in the costing.
3. The usage profile as illustrated in APPENDIX I has been adopted for this analysis.
4. Average cost of power paid by facility is assumed to be $0.10 per kWh.
5. Average cost of gas paid by facility is assumed to be $6 per GJ.
Figure 10 indicates the variation in possible gas cost savings at different gas prices for the above example of high stage heat recovery.

**Cost savings based on gas cost for high stage heat recovery**

![Graph showing cost saving vs gas cost](image-url)

**Figure 10:** Cost savings associated with heat recovery
Technology 5: Defrost management

Most facilities have cool rooms or freezer rooms that use fan coil evaporator units. Over time, ice forms on these coils and in their condensate trays due to airborne moisture.

The rate at which ice is formed on a coil depends on several factors such as room temperature, cooling load, ambient conditions and number of door openings. As the ice layer on the coils increases, the coil’s heat exchanging capacity decreases. Hence, there is a need to defrost coils.

The conventional defrosting methods are hot gas, electric, air and water. Many ammonia plants use hot gas defrost as there is sufficient heat available from the refrigeration plant and it is cheaper than electricity. The process of defrosting introduces heat into the refrigerated space, increasing the plant’s workload and energy consumption. The defrost process also consumes energy. Consequentially, if defrosting methods are not optimised, refrigeration plant efficiency will suffer.

Principles

Defrost management refers to managing the interval between defrosts, the duration of the defrost process and termination of defrosts. By convention, the frequency and duration of defrosts are fixed, regardless of room temperature and work load between defrosts. If there are too many defrosts or they last too long, heat will be needlessly added to the cool room. Hence optimised defrost management reduces overall refrigeration plant energy consumption.

The defrost management method proposed involves monitoring the time over which the coil has been operating at full capacity. The interval between defrosts will be shorter if the coil has been running continuously at full capacity and longer if it has been running at lower capacity. The capacity of the coil can be found by assessing the amount of cooling supplied to it over a period of time. In the case of a flooded ammonia evaporator coil, the capacity is the time for which the liquid solenoid valve is open.

This process requires continuous observation and tuning of parameters in the control system and is, therefore, suitable for remote optimisation – refer to the section on TECHNOLOGY 3: Remote Controlled Optimisation of Refrigeration Plant.

Plant equipment and information requirements

Equipment requirements

The following equipment is required to facilitate a sound defrost management strategy:

i. temperature sensing device attached to the face of the evaporator coil. When coil temperature reaches 10°C, the defrost is complete and can be terminated
ii. a means of tracking the room cooling load, such as a monitoring solenoid or control valve
iii. a high level computer (or remote CMS unit) which controls the defrost system.

Information requirements

i. Make and model of fan coil units in each of the rooms
ii. Defrost technique applied
iii. Coil arrangement within plant and circuit design
iv. Room temperature.

Plant benefits

i. Managed defrosting intervals prevent wasteful defrosting during low load periods
ii. Optimised defrost duration and termination of defrost prevent excess heat entering the cool room, easing the load on the refrigeration system and preventing energy waste
iii. Quick termination of defrost by electric defrosting units also saves unnecessary power consumption by the electric heaters
iv. Reduced need for defrosting during low-load situations means the refrigeration plant will be stable for longer periods of time. Similarly with an ammonia evaporator coil, fewer defrosts means fewer suction pressure fluctuations, leading to more stable operating.
Achievable savings

A freezer store with a maximum load of 500 kW operating 24/7/365 at an average load of 50% and an average energy cost of $0.10 per kWh will yield energy savings of approximately 100 MWh/year or $10,000 following implementation of a sound defrost management strategy. This is based on the assumption that the defrosting is either done by hot gas or electricity.

The achievable annual savings depend on:

i. the number, type and size of fan coil units in the room
ii. the room temperature
iii. the cooling load profile
iv. the type of defrost: Hot gas, electric, air or water
v. the number of defrosts and the defrost interval currently employed
vi. the defrost relief point: low temperature, intercooler or economizer vessel – refer to TECHNOLOGY 9(ii): Defrost Relief Piping to Intercooler or Economizer.

Figure 11 illustrates the effect of employing fixed defrost intervals and durations. As the average freezer load reduces, there is an unnecessary penalty on the refrigeration plant that can be avoided by employing a defrost management system.

Energy efficiency penalty with freezer load

![Energy efficiency penalty with freezer load](image)

Figure 11: Energy penalties associated with conventional defrosts

Estimated financial returns

This project is mainly achieved by implementing control algorithms. The only equipment that would be required is a coil temperature thermostat or sensor to terminate the defrost cycle. The estimated cost of equipment and initial definition of control algorithm is approximately $1,500 per evaporator, assuming that the site has a control system to which the thermostat can be connected. Optimisation costs would be additional based on the level of optimisation required.

For a site with 10 evaporators of 50 kW capacities each, typical capital costs and payback could be as follows:

<table>
<thead>
<tr>
<th>Energy Savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
<th>Payback [years]</th>
</tr>
</thead>
<tbody>
<tr>
<td>100,000</td>
<td>10,000</td>
<td>15,000</td>
<td>1.5</td>
</tr>
</tbody>
</table>
Technology 6: Variable cold store temperatures

Cold stores generally run to a fixed set-point room temperature throughout the day. This causes the refrigeration plant load to increase during peak periods to maintain temperature, and to run largely unloaded during off-peak periods.

For products that require cold storage and can tolerate slight variations in storage temperature, such as meat products and frozen vegetables, variable cold store temperatures can exploit off-peak efficiencies, lower off-peak power costs and minimize peak demand costs.

**Principles**

The workload of the refrigeration plant, and its energy consumption, can be reduced during the day by allowing temperatures to rise slightly. The plant can then cycle down to a lower temperature during the night, when it is cheaper to achieve lower temperatures.

The main sources of heat in a cold store are transmission and infiltration loads, introduced via walls, the floor and roof, which allow heat and outside ambient air into the cold store. Heat also enters when doors are opened. This means plant load is higher during the day – when energy charges are higher and lower at night.

This variable temperature strategy involves reducing temperatures during off-peak periods and raising them during the day. This shifts the load on the plant by running a higher suction pressure during off-peak hours, thus possibly reducing overall energy consumption and costs and avoiding demand peaks by intelligent shifting of load.

The strategy requires continuous monitoring of the plant, which makes it suitable for remote optimisation – refer to the section on TECHNOLOGY 3: Remote Controlled Optimisation of Refrigeration Plant.

**Plant benefits**

i. Possible energy savings during day-time by allowing higher cold store temperatures. A cold store usually running at -20°C can be reduced to -22°C during night time and run at -18°C for part of the day

ii. Possible overall reduction in energy costs

iii. Possible reduction in peak demand costs due to intelligent load shifting.

It should be noted that the above benefits are variable depending on the specific facility and the variation allowed by the particular products in the cool room. It is also likely that energy consumption may slightly increase due to the higher workload at night, but energy costs and peak demand costs may reduce, hence modelling has to be conducted based on specific operating conditions.

**Achievable savings**

The achievable annual savings depends on:

i. the amount of temperature variation allowed by the products

ii. the extent to which suction pressures can be varied

iii. peak and off-peak energy costs ($/kWh) and peak demand costs ($/kVA) paid by the facility

iv. whether the refrigeration system is dedicated to the cold stores or runs other production loads. If the system runs other loads at lower temperatures such as a blast freezer, no benefit may be possible unless the suction to the cold stores is split: refer to later section on plant design.

**Estimate of capital costs**

This project is mainly achieved by implementing control algorithms. The estimated cost of initial definition of control algorithm would typically be between $5,000 and $10,000 depending on the size of the installation, and assuming a modern computer is already in operation on the site. Optimisation costs would be additional based on the level of optimisation required.
Technology 7: Variable evaporator fan speeds

Fan-coil evaporators are used in most industrial cold storage applications. These evaporator fans are rarely speed controlled, running at full speed and turning on and off as required. The evaporator fan motor is a continuous heat load on the cool room which in turn, is an additional load on the refrigeration plant.

Principles

The evaporator fan is a major part of the overall cooling load on a cold space, especially in off-peak periods. They tend to be run unnecessarily at full speed, even when workload is low. By varying the fan speeds to suit load, energy savings are possible because lower fan speed translates to less heat being introduced to the cool room.

Reduction in fan speed is directly proportional to evaporator capacity. However, a fan’s power consumption is proportional to the cube of its speed. Therefore, reducing fan speed by 20% reduces fan power consumption by approximately 50%. Hence considerable energy savings are achievable using variable evaporator fan speeds. In applications using multiple evaporators, this operation best achieved by simultaneously varying the speed of all evaporator fans to suit the load, so as to ensure uniform air distribution in the cold store.

NB: The concept of variable evaporator fan speed control is not necessarily applicable to blast freezing as evaporator fans may need to run at full air flow to achieve full freezer performance.

Equipment requirements

The following equipment is required:

i. variable speed drives (VSD) for the evaporator fans – typically one VSD per evaporator. The VSD would control all fans in the evaporator unit and needs to be selected for the maximum current that could be drawn when all fans run at full speed

ii. sufficient programming capability in the control system to facilitate effective speed control logic.

Achievable savings

Figure 12 illustrates the power savings with respect to room load by virtue of installing variable speed drives on evaporator fans. As the graph indicates, the savings are greatest at low room loads. The horizontal line between loads of 10% and 50% indicates that a minimum fan speed of 50% has been considered in the modelling, which is typical for most evaporator fans.
The achievable annual savings depends on:

i. the load profile
ii. the size and number of evaporator fans (kW)
iii. the number of cold stores in the facility.

Estimated financial returns

Capital costs to implement variable evaporator fan speeds depend on the following:

i. the number of evaporators in the facility
ii. the size of the evaporator fan motors.

For a cold store with a total evaporator capacity of 500 kW (10 x 50 kW units), typical capital costs could be as follows:

<table>
<thead>
<tr>
<th>Item</th>
<th>Estimated cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment</td>
<td>$45,000</td>
</tr>
<tr>
<td>Labour</td>
<td>$8,000</td>
</tr>
<tr>
<td>Engineering</td>
<td>$4,000</td>
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<tr>
<td>Programming</td>
<td>$8,000</td>
</tr>
<tr>
<td>Total</td>
<td>$65,000</td>
</tr>
</tbody>
</table>

Typical achievable energy savings for the above example have been estimated to be as follows:

<table>
<thead>
<tr>
<th>Energy consumption of conventional system [kWh/year]</th>
<th>Energy consumption of improved system [kWh/year]</th>
<th>Energy savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
<th>Payback (years)</th>
</tr>
</thead>
<tbody>
<tr>
<td>370,000</td>
<td>80,000</td>
<td>290,000</td>
<td>$29,000</td>
<td>$65,000</td>
<td>2.24</td>
</tr>
</tbody>
</table>

1. Equipment allowed in the above costing includes ten variable speed drives (4 kW each), shielded cabling, electrical wiring and enclosures for VSDs.
2. The usage profile as illustrated in APPENDIX I has been considered for the purpose of this analysis.
3. Average cost of power paid by facility is considered as $0.10 per kWh.
Technology 8: Condensate sub cooling

Refrigerant that has condensed in the condensers of a refrigeration plant is commonly stored in liquid receiver(s) and supplied to either the field as high pressure liquid or the low pressure vessels of the plant.

In most cases, the liquid is either at saturation temperature or only slightly sub cooled. Further subcooling of the liquid improves the thermodynamics of the refrigeration plant, permitting energy savings due to reduced flash gas and therefore a higher coefficient of performance (COP).

This further subcooling can be achieved in different ways, and two methods are considered here:

i. subcooling using cold town water supply to the condensers
ii. subcooling using economisers on high stage screw compressors.

The above methods can be used separately and, in some cases, may be feasible in combination. Where only reciprocating compressors are used, town water subcooling is the only option. The more modern (and positive) trend towards using rain water storage may render this option less feasible, as water from rainwater tanks tends to be warmer than town water.

(i) Condensate subcooling with town water supply

Principles

On a system using cooling towers or evaporative condensers, subcooling of high pressure refrigerant liquid can be achieved by exchanging heat from the condensate with incoming town water used as make up water. The incoming town water is generally between 10°C and 15°C depending on the time of the year and the geographical location. Elevating the temperature of incoming town water before it enters the cooling towers or evaporative condensers has a marginal effect on their performance. Thus, the relatively cold town water can be used to maximise efficiency by first passing it through a condensate sub cooler, then routing it to the cooling towers or evaporative condensers. This sub cooler is typically a refrigerant-to-water plate heat exchanger or a water jacket sub-cooling arrangement around the refrigerant liquid supply line from the liquid receiver.

Plant equipment and information requirements

Equipment requirements

The following equipment is required:

i. a refrigerant-to-water plate heat exchanger. Alternatively a tube-in-tube section could be installed on the liquid supply line from the liquid receiver assuming that the required straight line length is available
ii. insulation for the sub-cooled refrigerant liquid line if the pipe length of the sub-cooled line is considerable.

Information requirements

i. Make and model details of compressors, cooling towers and evaporative condensers
ii. Size (diameter) of main high pressure refrigerant liquid line.

Plant benefits

Increases the COP of the refrigeration plant without impacting on maintenance costs or plant operation.

Achievable savings

The achievable annual savings depend on:

i. town water supply temperature
ii. condensing (liquid) temperature of the plant.

Figure 13 illustrates the improvement in efficiency of the system due to condensate sub-cooling by town water. As the town water temperature increases, the COP improvement reduces. Similarly, as condensing temperature decreases for given water inlet temperature, the COP improvement reduces. Variation in town water temperature is mainly due to geographical and seasonal variations.
Estimated financial returns
Capital costs to implement condensate sub-cooling depend on the following:

i. the size of heat exchanger required depending on refrigerant flow
ii. the type of heat exchanger – plate or tube-in-tube.

For a refrigeration system with a peak heat rejection load of 500 kW, typical capital costs could be as follows:

<table>
<thead>
<tr>
<th>Item</th>
<th>Estimated cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment</td>
<td>$3,000</td>
</tr>
<tr>
<td>Labour</td>
<td>$3,000</td>
</tr>
<tr>
<td>Engineering</td>
<td>$2,000</td>
</tr>
<tr>
<td>Total</td>
<td>$8,000</td>
</tr>
</tbody>
</table>

For a refrigeration system with a peak heat rejection load of 500 kW, the estimated savings and payback are as follows:

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>502,000</td>
<td>487,000</td>
<td>15,000</td>
<td>$1,500</td>
<td>$8,000</td>
<td></td>
<td>5.33</td>
</tr>
</tbody>
</table>

1. The annual energy savings are calculated based on an average water temperature of 12°C and average condensing temperature of 30°C.
2. The usage profile presented in Appendix I has been used for this analysis.
3. Equipment allowed in the above costing includes a plate heat exchanger and refrigerant and cooling water pipe-work.
4. Average cost of power paid by facility is considered as $0.10 per kWh.
(ii) Condensate subcooling with economisers

Principles
On any single or 2-stage ammonia plant containing screw compressors on the high side, condensate can be sub-cooled by the refrigerating effect of the so-called economiser port of these compressors.

The economiser port provides a suction pressure intermediate between that of main suction pressure of the high-stage compressor, and the discharge pressure (system condensing pressure in this case). For most typical high-stage applications, the intermediate suction pressure is typically equivalent to evaporation temperatures in the range of 0°C to 10°C.

By expanding some of the high pressure liquid refrigerant from the liquid receiver to this intermediate suction pressure, the cooling effect of the evaporating refrigerant at this pressure can be utilised to sub-cool the remaining high pressure liquid refrigerant to a lower temperature before it is then expanded to the intercooler or high stage suction pressure, thus reducing the amount of flash gas developed in the intercooler or at high stage suction pressure. Typically this cooling effect is utilised in several different ways:

1. open flash economiser vessel, in which all refrigerant is first expanded into the economiser, and then expanded further into the intercooler / high-stage suction vessel
2. economiser vessel with flooded sub-cooler coil, in which the remaining high pressure refrigerant is sub-cooled by immersion within the evaporating refrigerant at intercooler pressure
3. sub-cooler plate heat exchanger, in which the remaining high pressure refrigerant is sub-cooled by evaporating refrigerant within the heat exchanger. Several different arrangements are possible.

Energy savings are achieved due to the reduced flash gas introduced into the intercooler vessel and hence increased refrigeration plant efficiency.

This concept could be beneficial on sites that use recovered rainwater as feed water to the evaporative condensers, as this reduces or eliminates the capability to sub-cool the refrigerant liquid. On some sites, a combination of the two technologies could yield significant energy savings, although at a high capital cost. For example, the system could benefit from low town water temperatures in winter, and then largely rely on economiser operation in summer, with a combination of modes during other seasons.

Plant equipment and information requirements

Equipment requirements
The following equipment is required:

i. either of the above equipment combinations achieve liquid sub-cooling, such as:
   b. an open flash economiser vessel able to service some or all of the high stage compressors in the plant
   c. an economiser vessel fitted with a liquid sub-cooling coil
   d. a liquid sub-cooling plate heat exchanger operated either in flooded or direct expansion mode.

ii) insulation for the sub-cooled refrigerant liquid line if it is long.

Information requirements

i. The make and model details of high stage compressors in the plant
ii. The available space in the plant room to install the new economiser vessel/heat exchanger
iii. The size (diameter) of main high pressure refrigerant liquid line.

Plant benefits

i. It increases the COP of the refrigeration plant without impacting on maintenance costs or plant operation.

Achievable savings

The achievable annual savings depend on:

i. inter-stage and/or high stage suction and the condensing and high pressure liquid temperatures of the plant, which have an impact on the intermediate pressure at which the economiser vessel operates. This finally governs the temperature to which the refrigerant liquid can be sub-cooled.
Figure 14: COP improvement due to condensate sub cooling by high stage economiser

Figure 15: Increase in capacity due to condensate sub cooling by high stage economiser.

Figure 14 illustrates the improvement in the coefficient of performance (COP or efficiency) of the system and figure 15 illustrates the increase in available capacity of the system due to condensate sub-cooling using an economiser vessel on the high stage compressors, for a range of suction temperatures (Ts) of the system. This is the suction temperature of single stage systems (dairies, breweries, wineries) or the interstage temperature of two-stage systems (abattoirs, cold stores, chicken processors, etc) The lower this suction temperature, the greater is the increase in COP. For a typical system with a suction temperature of -10°C and a condensing temperature of 35°C, the increase in COP is around 4.5% and the increase in refrigeration capacity is around 7%.
Estimated financial returns

Capital costs to implement this technology depend on the following:

i. the size of economiser vessel/heat exchanger required
ii. the number of high stage compressors to be economised.

For a refrigeration system with a high stage load of 500 kW, typical capital costs (open flash option) could be as follows:

<table>
<thead>
<tr>
<th>Item</th>
<th>Estimated cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment</td>
<td>$8,000</td>
</tr>
<tr>
<td>Labour</td>
<td>$6,000</td>
</tr>
<tr>
<td>Engineering</td>
<td>$5,000</td>
</tr>
<tr>
<td>Total</td>
<td>$19,000</td>
</tr>
</tbody>
</table>

For the example considered above, typical achievable savings could be as follows:

<table>
<thead>
<tr>
<th>Energy consumption of conventional system [kWh/year]</th>
<th>Energy consumption of improved system [kWh/year]</th>
<th>Energy savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
<th>Payback [years]</th>
</tr>
</thead>
<tbody>
<tr>
<td>586,000</td>
<td>558,000</td>
<td>28,000</td>
<td>$2,800</td>
<td>$19,000</td>
<td>6.78</td>
</tr>
</tbody>
</table>

1. The usage profile presented in Appendix I has been used for the purpose of this analysis.
2. Equipment allowed in the above costing includes a vertical economiser vessel, pipe-work, fittings and insulation.
3. Average cost of power paid by facility is considered as $0.10 per kWh.
Technology 9: Plant design review

Many plants may be inefficient due to deficiencies in design and installation. A thorough review of the refrigeration plant could result in the identification of shortcomings or bottlenecks that result in reduced plant efficiency. These can include:

i. undersized or poorly located heat exchangers
ii. fouling of heat exchanger surfaces, due to corrosion, ice formation, oil accumulation or other causes
iii. undersized suction or discharge lines or associated line components resulting in excessive pressure losses
iv. incorrect routing of suction lines, in particular wet return lines, resulting in excessive pressure drop, excessive liquid retention or unstable flow.

A design review would also address other issues such as:

i. the combination of loads requiring different evaporation temperatures onto a single suction pressure level, with the aim of splitting suction pressure levels. This typically occurs where blast freezing and cold storage, or air-conditioning and process room cooling co-exist.
ii. the set levels of operating suction pressures in the plant, with the view to raising compressor suction pressure settings
iii. A defrost relief strategy employed in the plant designed to return defrost relief to the highest possible suction pressure level in the plant
iv. eliminating liquid injection oil cooling in preference to thermostop or water cooling on screw compressors.

Addressing bottlenecks is often a necessary precursor to the above review items.

Figures 16 and 17 indicate the effect of suction temperatures on the COP of 2-stage and single stage refrigeration plants respectively. These have been referred to in the following sub sections. Figure 18 illustrates the effect of discharge line pressure drop and therefore, increased equivalent plant condensing temperature on the COP of a refrigeration plant. This has been referred to in the following sub sections.

Effect of suction temperature on COP of a 2 stage plant

Figure 16: Effect of suction temperature on COP of a 2 stage plant
Effect of suction temperature on COP of a single stage plant

![Figure 17: Effect of suction temperature on the COP of a single stage plant](image)

Effect of discharge line pressure drop on COP of a refrigeration plant

![Figure 18: Effect of discharge line pressure drop on COP of a refrigeration plant](image)
(i) Bottleneck removal

Bottlenecks can occur in items such as heat exchangers, suction lines or other devices configured or in a condition such that there is a design flaw which must be overcome by running the refrigeration plant inefficiently.

Principles

The identification of bottlenecks requires careful review of the plant design documentation, process diagrams, arrangement drawings and operating manuals, in conjunction with a plant inspection by a competent engineer.

Once identified and the effect on the plant established, the removal of bottlenecks generally requires the operator to:

• replace or relocate undersized or poorly located heat exchangers
• address the causes of fouling or replace corroded heat exchangers where fouling has been identified
• replace suction or discharge lines or line components that are undersized
• rectify incorrect line routing.

Often it is found that once bottlenecks are removed, suction pressures can be raised or controlled at variable levels, where previously they had to be kept low to achieve plant performance. This leads to the improvements indicated in Figures 16 to 18. These energy savings are achieved by:

• raising suction pressure set-points whilst maintaining process or room temperatures
• enabling variable suction pressure set-points, such as increased low side suction pressures at low-load conditions, or floating inter-stage pressures to minimize overall compressor power – refer to Technology 1: Variable Plant Pressure Control.

Plant benefits

Plant benefits include:

i. higher suction and/or lower discharge pressures which reduce compressor power consumption, for a given cooling capacity
ii. floating or optimised inter-stage pressures allow the total power consumption of all compressors to be minimised
iii. maximising the operating suction pressures (low suction or inter-stage), which will maximise the availability of the plant and could result in capital cost savings by avoiding or reducing additional compressor capacity requirements where plant capacity needs to be extended.

Achievable savings

The achievable annual savings depends on:

i. the original design and current age of heat exchangers
ii. the current suction and discharge line design and routing

The effect of suction temperature on the efficiency of 2-stage and single stage refrigeration plants has been illustrated in figures 16 and 17. The effect of discharge line pressure drop on the efficiency of a refrigeration plant has been indicated in figure 18. All these are considered as bottlenecks.

The energy savings indicated in Figure 17 assume a 500 kW refrigeration load and a possible elevation in suction temperature by 5°C due to bottleneck removal, aggregated over the load profile presented in APPENDIX I.

<table>
<thead>
<tr>
<th>Energy consumption of conventional system (kWh/year)</th>
<th>Energy consumption of improved system (kWh/year)</th>
<th>Energy savings (kWh/year)</th>
<th>Energy cost savings ($/year)</th>
</tr>
</thead>
<tbody>
<tr>
<td>677,000</td>
<td>577,000</td>
<td>100,000</td>
<td>$10,000</td>
</tr>
</tbody>
</table>

1. Average cost of power paid by facility is considered as $0.10 per kWh.
2. The usage profile indicated in APPENDIX I has been used for the purpose of this analysis.
**Estimated capital costs**

The capital costs depend on the specific bottleneck being addressed such as:

i. cleaning or replacing heat exchangers
ii. suction and/or discharge lines, depending on sizing and extent of replacement required.

Due to various possibilities and complexities of upgrade involved to achieve the above-mentioned energy savings, the capital costs have not been indicated.

**(ii) Suction splitting**

Facilities that require blast freezing usually also have holding freezers to store frozen product. Often, the blast freezers and holding freezers are operated from the same suction vessel. The inherent inefficiency in this approach is that the higher temperature in a cold storage plant (typically at -30°C suction) is generated at an excessively low temperature designed for blast freezing (typically at -40°C). Furthermore, and as indicated in Figures 16 and 17, raising suction temperatures where possible will improve refrigeration plant efficiency.

A similar situation exists in facilities that have cool rooms or process areas at intermediate temperatures (0°C to 10°C) and comfort air-conditioned loads (18°C to 25°C) both connected to the intermediate suction temperature (typically -10°C).

**Principle**

Suction splitting on low temperatures involves separating the suction of the blast freezers and the holding freezers by introducing a new suction level in the plant to independently serve the holding freezers. This involves raising the suction temperature for the holding freezer, thus improving refrigeration plant efficiency.

Suction splitting on medium temperatures may involve diverting the high temperature (18°C to 25°C) loads to a separate suction level. In plants with economised screw compressors, this can be achieved by using a side load connection, which handles dual suction pressures without the need to dedicate separate compressors to both applications.

**Plant benefits**

Plant benefits include:

i. energy savings due to running holding freezers at appropriate suction temperature rather than at blast freezer suction. This allows suction temperature to be elevated by about 15°C.

ii. in plants where compressor duty is near design capacity, splitting the suction relieves compressor capacity so that throughput in the blast freezers can be increased if required, assuming suction lines are adequately sized.

iii. energy savings and suction splitting as above, but at medium temperature levels.

**Achievable savings**

For a refrigeration system with a peak freezer load of 500 kW and aggregated over the load profile presented in APPENDIX I, the estimated cost savings and payback are as follows:

<table>
<thead>
<tr>
<th>Energy consumption of conventional system [kWh/year]</th>
<th>Energy consumption of improved system [kWh/year]</th>
<th>Energy savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
</tr>
</thead>
<tbody>
<tr>
<td>545,000</td>
<td>471,000</td>
<td>74,000</td>
<td>$7,400</td>
</tr>
</tbody>
</table>

1. Average cost of power paid by facility is considered as $0.10 per kWh.
2. The usage profile indicated in APPENDIX I has been used for the purpose of this analysis.

Figure 17 illustrates the effect of suction splitting on the efficiency of the plant, which contains blast freezers and holding freezers, for various holding freezer suction temperatures. The efficiency improvement is in the range of 13% to 16% for the range of suction temperatures mentioned above.

The achievable annual savings depends on:

iv. the actual difference in temperature between the two suction levels in consideration. The greater the difference in temperatures, the greater the savings

v. the load profile

vi. the number of days of production – the greater the number of days, the larger the savings.
Plant equipment requirements

Various technical solutions as described below are possible for a project of this nature. The costs could vary widely based on the solution chosen and hence have not been indicated in this document.

**Low cost solution:**

Use existing equipment if available:

i. assume the plant has an existing pressure vessel that could be used for the new suction temperature level

ii. on systems with blast freezers and holding freezers, use existing low temperature liquid pumps to feed liquid from the low temperature vessel to the holding freezer. This would lead to a slight reduction in energy savings as compared to using dedicated liquid pumps

iii. assume the plant has a compressor that can be dedicated to serve the new suction temperature level.

**Medium cost solution:**

If there is no dedicated pressure vessel available in the plant:

i. a simple "slop pot" solution could work where the wet return vapour from the freezer evaporators could be directed and the dry suction gas provided to the respective compressors. This is a less expensive solution than a full scale pressure vessel

ii. use existing low temperature liquid pumps to feed liquid from the low temperature vessel to the holding freezer. This would lead to a slight reduction in energy savings as compared to using dedicated liquid pumps

iii. assume the plant has a compressor that can be dedicated for holding freezer operation.

**High cost solution:**

This would involve all new equipment:

i. a full scale pressure vessel

ii. new ammonia circulation pumps

iii. a new compressor for dedicated operation at the new suction temperature level.

**NB:** The number of compressors required will rise with an increase in the number of different suction levels. Evaluation is required to gauge the additional plant complexity and hence the practical feasibility of implementing such a project.

As an alternative to using new compressors, the side load connection on existing screw compressors can be used to provide capacity to the new suction temperature level. This depends on:

i. available side load capacity as compared to the required capacity for the application

ii. screw compressor load status. A compressor running on slide valve control would provide reduced slide load capacity as the compressor unloads and, at a certain point, would not provide any side load capacity at all. A variable speed compressor, however, could provide side load capacity even at its lowest speed as far as the slide valve is maintained in the 80% to 100% range

**Plant benefits**

**Plant benefits include:**

i. energy savings due to reduced load on the refrigeration plant if the liquid relieved during defrost is directed into a higher temperature vessel rather than the low temperature vessel

ii. low-stage suction pressure is not influenced by flash gas surges from defrost processes. This has a beneficial effect on the temperature control of other rooms connected to the same suction.

**Plant equipment requirements**

The following equipment is required:

i. separate defrost relief pipe-work to the intercooler or economiser. The pipe-work could be directly branched into a wet return or liquid inlet nozzle on the respective vessel.

(iii) **Defrost relief piping to intercooler or economiser**

When an evaporator undergoes a defrost cycle, the hot gas entering the evaporator condenses into liquid and is relieved through the suction line to return the liquid to the system. In the process the liquid partly evaporates and generates "flash gas" which must then be compressed by the compressors. In a two-stage refrigeration plant with the low stage operating at around -30°C, defrost relief often occurs through the low stage suction line and into the low temperature accumulator vessel. Therefore, there is an additional load on the refrigeration plant during defrost as a result of the relieved liquid flashing into the vessel.
Figures 19 and 20 illustrate the conventional and the energy-efficient modes of defrost relief.

Conventional defrost

![Schematic of conventional defrost relief technique](image)

**Figure 19:** Schematic of conventional defrost relief technique

Energy efficient defrost

![Schematic of energy efficient defrost relief technique](image)

**Figure 20:** Schematic of energy efficient defrost relief technique

**Achievable savings**

Modelling has suggested that the typical achievable savings by defrost relief piping to the intercooler or economiser is estimated to be around **0.40 kWh/day per kW of freezer room load** based on an average ice thickness of 0.5 mm, low temperature vessel at -28°C, intercooler/economiser vessel at -10°C and freezer room temperature of -20°C. For a 500 kW average freezer room load and an average energy cost of $0.10 per kWh, the energy savings are around **74 MWh/year** or **$7,400/year**.

The achievable annual savings depends on:

i. the actual thickness of ice on the surface of the evaporator
ii. the air temperature in the freezer store
iii. the vessel temperatures
iv. the number of defrost cycles per day
v. the number of evaporators.
Estimated financial returns

Capital costs to implement energy efficient defrost relief depend on the following:

i. the number of evaporators in the facility
ii. the length of pipe-work required between the evaporators and the intercooler/economiser vessel in plant room.

For the example considered and modelled above, typical costs, energy savings and payback could be as follows:

<table>
<thead>
<tr>
<th>Item</th>
<th>Estimated cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment</td>
<td>$10,000</td>
</tr>
<tr>
<td>Labour</td>
<td>$5,000</td>
</tr>
<tr>
<td>Total</td>
<td>$15,000</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>Energy savings</th>
<th>Energy cost savings</th>
<th>Project cost</th>
<th>Payback</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[kWh/year]</td>
<td>[$/year]</td>
<td>[$]</td>
<td>[years]</td>
</tr>
<tr>
<td></td>
<td>74,000</td>
<td>7,400</td>
<td>15,000</td>
<td>2.03</td>
</tr>
</tbody>
</table>

1. Equipment allowed in the above costing includes for defrost relief pipe-work and insulation from 10 evaporators and 30 metres of pipe-work from the evaporators to the intercooler or economiser vessel.
2. Average cost of power paid by facility is considered as $0.10 per kWh.

(iv) Removal of liquid injection oil cooling on screw compressors

Industrial screw compressors in ammonia plants are frequently cooled by direct injection of liquid refrigerant into the compressor. The expansion and evaporation of this refrigerant cools the gas and oil during compression, but reduces the effective compressor capacity, increases power consumption and renders any heat recovery from the system of little value due to the low compressor discharge and oil temperatures. Converting such systems to water or thermosiphon oil cooling improves efficiency, increases cooling capacity and enables the application of heat recovery.

Principles

Screw compressors used in ammonia refrigeration plants circulate vast quantities of oil through the system to aid with the sealing of the compression space during compression, and this oil absorbs a substantial portion of the heat generated by compression of the gas. The oil is separated out from the discharge gas and returned to the compressor, but requires to be supplied to the compressor at temperatures generally in the range of 40-60°C. Some means of oil cooling is required to achieve these temperatures, and generally designers have the choice of two main options:

1. direct injection of high pressure liquid refrigerant into the compression space (generally at or near the suction port). This refrigerant evaporates, cooling the gas and oil during the compression process and resulting in relatively low discharge and oil temperatures
2. external oil cooling generally via a shell and tube heat exchanger installed in the oil circulation system. These oil coolers are cooled either by means of cooling water or by natural circulation of refrigerant at receiver pressure (thermosiphon).

Direct liquid injection is generally an effective, low cost and low maintenance option, and is widely used. However, the effective cooling capacity of the compressor is reduced, power consumption is increased and, due to the low discharge temperature and low oil temperature in the oil circulation system, heat recovery is rendered effectively of little value. This means upgrading the compressor to external oil cooling has a range of system benefits.

Plant benefits

Replacing liquid injection oil cooling with thermosiphon or water-based oil cooling has the following benefits:

i. an increase in the compressor’s refrigeration capacity by around 5-10%
ii. an increase in refrigeration efficiency by around 10-15%
iii. an increase in discharge temperature from 50°C to around 70°C, thus providing increased heat recovery potential.

Plant equipment requirements

The following equipment is required:

i. a thermosiphon or water-based oil cooler
ii. an ammonia liquid and vapour return pipe-work between the liquid receiver and the oil cooler (if thermosiphon) OR cooling water pipe-work between the evaporative condensers/cooling towers and the oil cooler (if water based).

Achievable savings

The achievable annual savings depend on:

i. the size of screw compressors in the plant.
Figure 21 illustrates the improvement in efficiency of the system due to replacement of liquid injection oil cooling with water-based oil cooling (or thermosiphon oil cooling), at various compressor suction temperatures. Significant increase in efficiency is possible due to implementing this technology and it offers good savings potential at relatively low capital costs. Additionally, considerable increase in cooling capacity is also obtained as indicated in the above graph.

**Estimated financial returns**

The option of replacing liquid injection oil cooling with water-based oil cooling has been considered here:

Capital costs to implement water-based oil cooling depend on the size of the oil cooler.

For an industrial screw compressor with a refrigeration capacity of 500 kW at a saturated suction temperature of -10°C and a condensing temperature of 35°C, typical capital costs could be as follows:

<table>
<thead>
<tr>
<th>Item</th>
<th>Estimated cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment</td>
<td>$10,000</td>
</tr>
<tr>
<td>Labour</td>
<td>$4,000</td>
</tr>
<tr>
<td>Engineering</td>
<td>$4,000</td>
</tr>
<tr>
<td>Total</td>
<td>$18,000</td>
</tr>
</tbody>
</table>

For the example considered above, typical energy savings are as follows:

<table>
<thead>
<tr>
<th>Energy consumption of conventional system [kWh/year]</th>
<th>Energy consumption of improved system [kWh/year]</th>
<th>Energy savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
<th>Payback [years]</th>
</tr>
</thead>
<tbody>
<tr>
<td>665,000</td>
<td>595,000</td>
<td>70,000</td>
<td>$7,000</td>
<td>$18,000</td>
<td>2.57</td>
</tr>
</tbody>
</table>

1. The usage profile presented in Appendix I has been used for the purpose of this analysis.
2. Equipment allowed in the above costing includes a 100 kW shell and tube water cooled oil cooler, oil and water pipe-work and fittings and a relief valve for the oil cooler.
3. Average cost of power paid by facility is assumed as $0.10 per kWh.
Technology 10: 
Ammonia system maintenance

(i) Optimised oil supply to screw compressors

Lubricating oil is used in screw compressors to lubricate the main rotor bearings and provide a hydraulic seal between the suction gas, rotors and the rotor housing which is essential for the gas compression process.

If insufficient oil is provided to the suction gas, the refrigeration capacity of the compressor will diminish due to internal leakage, resulting in excessive oil temperatures. Conversely, excess oil will restrict the compression space within the compressor, resulting in over compression of the gas and therefore excessive power consumption. Inefficiencies in over compression are compounded when a compressor operates at reduced speed, given the increase in oil flow to suction gas ratio.

Many operators tend to set oil flow rates at conservatively high levels. Attention should be given to adjusting oil flow rates to maintain them at optimum levels to maximize compressor efficiency.

Principles

Adjusting the lubricating oil feed rate is generally achieved by adjusting a suitable regulating valve fitted into the feed line to the compressor. Oil absorbs the heat of compression from the discharge gas, therefore the quantity of oil injected into a screw affects the gas and oil discharge temperatures. Higher discharge and oil temperatures are achieved at low oil flow rates, and vice versa.

The optimum oil temperature for a given compressor depends on the application (low stage, high stage, single stage economized) and the operating suction and discharge conditions. It is customary to set oil flow rates to achieve discharge temperatures near 70°C, although settings can vary from this value.

Where heat is recovered from the compressor oil or discharge gas, it is desirable to minimize the oil flow rate in order to maximize discharge and oil temperatures, thus maximizing the achievable hot water temperatures.

Plant benefits

Plant benefits include:

i. optimised oil feed rates will result in the lowest possible power consumption of the compressor under the given operating conditions

ii. a minimised oil feed rate will also result in the most effective heat recovery from the compressor.

Achievable savings

The achievable annual savings depend on:

i. the compressor application; greater savings are achievable with high stage compressors

ii. the extent of oil overfeed.

Where significant oil overfeed exists, energy savings of 5% to 10% during full speed operation, and up to 20% during low speed operation can be achieved by correcting the oil feed rate.

Estimated financial returns

Correcting oil feed rates can be conducted at any time at minimal cost (technician labour costs only).

It should be noted that:

i. it is recommended that particular care to correct oil flow rates be given to variable speed compressors

ii. the compressor manufacturer’s recommendations should be adhered to at all times.

(ii) Water and air purging

When air and water contaminate ammonia refrigerant, system efficiency falls sharply. Moist air enters a refrigeration system under the following circumstances:

• during maintenance; if the portion of the system being attended to is not pulled into a proper vacuum before the system starts again, air remains in the system and accumulates in the condensers.

• in low suction temperature applications; if the system operates in a vacuum, i.e. below -33°C suction temperature in the case of ammonia, air and therefore water (via moisture) enters the system via compressor shaft seals, valve glands and/or pipe joints.

Once in a refrigeration system, air and other non-condensable gases eventually accumulate in the condensers. This reduces the heat rejection surface area of the condensers and the head pressure of the plant rises to compensate, resulting in increased energy use.

Water usually accumulates at the low pressure side of the system and causes the evaporator temperature to rise. To maintain the desired evaporator temperature, the corresponding compressor suction temperature must be reduced, thus reducing compressor capacity. To achieve the same capacity as a system with no water content, the compressors will have to run at higher load or additional compressors must operate, resulting in increased refrigeration plant power consumption.
Principles

Removing air from the ammonia refrigerant (purging) can be done automatically or manually. Manual purging is done using purging points installed when the plant is built. Typically these purging points are located on the liquid outlet from the condensers where the non-condensable gases tend to accumulate. Where there is an increased chance of air accumulation such as on low pressure systems that operate in vacuum, automatic purgers are advisable. Manual purging is sufficient on single stage systems that do not operate in a vacuum.

It is critical to ensure the design of the system effectively removes air and other non-condensable gases. In many instances, the installed purging arrangement does not eliminate all gases, and creates a bottleneck. A competent engineer needs to ascertain whether the purging system will completely eliminate contaminants.

Automatic air purgers do not deal with any water vapour drawn into the system. Water purgers (ammonia anhydrators) are available for this purpose. Some modern purgers remove both air and water but are more expensive than air purgers. For a plant that does not have any air purging or water purging installed, a combination air and water purger makes good economic sense.

Figure 22 illustrates the effect of air and other non-condensable gases in an ammonia refrigeration system. The power consumption of the plant increases by around 3% per degree C rise in equivalent condensing temperature. The greater the amount of non-condensable gases in the system, the greater is the increase in equivalent condensing temperature. If not attended to, this can result in the plant tripping on high head pressure and loss in compressor capacity.

Effect of air in an ammonia refrigeration system

![Figure 22: Effect of air in an ammonia refrigeration system](image-url)
Figure 23 illustrates the effect of water in an ammonia refrigeration system. As the amount of water in ammonia increases, the power consumption increases due to the fact that the plant loses capacity as the suction temperature reduces.

**Effect of water in an ammonia refrigeration system**

![Graph showing the effect of water concentration on power consumption](image)

**Plant benefits**

Plant benefits include:

i. proper purging of air and water from ammonia ensures optimum suction and discharge pressures, thus resulting in power savings

ii. on a plant operating at or near full capacity, removal of air and water from ammonia is critical to ensure that sufficient cooling and heat rejection capacity is available.

Furthermore, water and air have other negative effects on plant operation that cause maintenance and operational problems, and resolution of these problems is a clear benefit to the operation for these reasons as well. As a general rule, it is recommend that all two-stage ammonia plants have a well-designed air-removal system and testing of water content should be conducted at least annually.

**Achievable savings**

The achievable annual savings depends on:

i. the effectiveness of existing air and water removal systems or air and water ingress avoidance procedures

ii. the load profile

iii. the initial air and water content before implementing removal systems.

As it is not possible to predict or, in some cases, even measure the air or water concentration, the actual savings achievable on a particular site are difficult to predict.
## Estimated financial returns

### Air purger

<table>
<thead>
<tr>
<th>Item</th>
<th>Estimated cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equipment</td>
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<tr>
<td>Labour</td>
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<tr>
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</table>

<table>
<thead>
<tr>
<th>Energy consumption of system with air [kWh/year]</th>
<th>Energy consumption of purged system [kWh/year]</th>
<th>Energy savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
<th>Payback [years]</th>
</tr>
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<tbody>
<tr>
<td>695,000</td>
<td>595,000</td>
<td>100,000</td>
<td>$10,000</td>
<td>$20,000</td>
<td>2</td>
</tr>
</tbody>
</table>

1. Equipment allowed in the above costing includes a standard industrial grade automatic multi-point air purger. For a system which operates at suction temperatures above -33°C, manual purging may suffice and the capital costs would be correspondingly lower.
2. Average cost of power paid by facility is assumed at $0.10 per kWh.
3. The energy savings is calculated on the assumption that the plant condensing temperature increases by around 5K due to the presence of air and other non-condensable gases.
4. The usage profile illustrated in APPENDIX I has been used for this analysis.

### Water purger

<table>
<thead>
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</tr>
</thead>
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<td>Labour</td>
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<td><strong>Total</strong></td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Energy consumption of system with air [kWh/year]</th>
<th>Energy consumption of purged system [kWh/year]</th>
<th>Energy savings [kWh/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
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<td>631,000</td>
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<td>36,000</td>
<td>$3,600</td>
<td>$11,000</td>
<td>3.05</td>
</tr>
</tbody>
</table>

1. Equipment allowed in the above costing includes a standard industrial grade automatic water purger.
2. Average cost of power paid by facility is assumed as $0.10 per kWh.
3. The energy savings is calculated on the assumption that the system contains 5% water.
4. The usage profile illustrated in APPENDIX I has been used for this analysis.
Fluid chillers vary greatly in relation to their efficiency and considerable energy savings can be achieved by selecting a chiller well suited to the application.

Considerable development has been undertaken in recent years to develop high efficiency chillers, such that great differences in performance exist within the market, and older chillers are often substantially less efficient than their modern counterparts.

Furthermore, most chillers on the market are designed for air-conditioning applications, where summer operation dominates the annual operating profile. Process chillers have substantially different requirements, rendering some HVAC type chillers less suitable for such applications.

**Principles**

**Energy efficiency**

Several considerations are important to achieving overall high energy efficiency:

i. Fluid temperatures – chillers well suited to standard chilled water operating conditions 6°C / 12°C can be inappropriate for glycol chilling i.e. less than 0°C

ii. Water-cooling vs. air-cooling – water cooling generally results in lower overall/annual power consumption, but requires an additional cooling water system. In addition, maintenance costs can be higher

iii. Choice of refrigerant and chiller design – modern low-charge ammonia chillers and oil-free R134a chillers have substantial efficiency benefits over conventional HFC-based chillers

iv. Part load efficiency of the chiller – well designed chillers offer higher efficiency at partial loads, and in most cases chillers operate at part load for the greater part of their operating life. Chillers with reduced part load efficiency should be avoided

v. Low ambient efficiency of the chiller – well designed chillers are able to capitalise on low ambient (winter) conditions, and operate at increased efficiency under these conditions

vi. Environmental conditions – air cooled chillers often deteriorate rapidly in coastal environments, or in the vicinity of certain chemical plant or mining activities, causing power consumption to increase. Chillers need to be able to withstand the conditions under which they are expected to operate.

**Industry practice**

It is common industry practice to evaluate and compare chiller performance at standard conditions, i.e. chiller water inlet/outlet of 6°C / 12°C and cooling water inlet/outlet temperatures of 29.5°C / 35°C (water cooled chillers) or in the case of air-cooled chillers, at design summer ambient conditions. These are conditions widely used in the air conditioning industry, which represents the largest single market for fluid chillers. Consideration is given to part load operation, and integrated part load values (IPLVs) are published to allow comparison of chillers when they will frequently operate in low-load conditions.

In industrial applications, chillers are subjected to substantially different conditions and load profiles than in air-conditioning applications, hence the chiller design criteria are different. Using an “air conditioning” style of chiller for process applications frequently comes at a high energy cost.

Further consideration should be given to the following:

i. Most process applications require fluid temperatures lower or higher than the standard design values of 6°C (supply) and 12°C (return). Most food processing applications generally require ice-water (1°C), or chilled water/glycol (typically -2°C to -8°), whilst plastics processing applications, for example, require water between 10°C and 12°C.

• Chillers designed for low-pressure refrigerants (e.g. R134a or R600a) are substantially less well suited to low temperature applications. Under these conditions refrigerants such as ammonia (R717) generally offer the greatest energy efficiency. Ammonia-based process chillers have been specifically developed to serve this market.

• Conversely, low pressure refrigerants can be beneficial for high temperature applications, but special attention must be given to condenser capacities (next point) and low ambient conditions to ensure that these benefits are properly utilised. Standard R134a chillers often do not meet these requirements and hence the benefit of the low-pressure refrigerant is lost.
• Chillers designed for standard fluid temperatures will operate at higher capacities if fluid temperatures rise, and therefore they require larger condenser capacity to cope with the greater heat rejection. Chillers therefore either need to be fitted with larger condensers or down rated. This would involve fitting smaller compressors to maintain efficient operation. Selecting an oversized chiller with good part load performance can overcome the restrictions of standard chiller design in many cases, but at additional capital cost.

ii. Chillers using centrifugal compressors, or direct expansion (DX) evaporators, often cannot be operated at condensing pressures substantially below design values. Centrifugal chillers can experience instability (surge), whilst DX chillers can experience capacity reductions due to reduced performance by the expansion valve. Such chillers use various techniques to maintain high condensing pressure, even under low ambient temperature conditions. This result is an energy penalty under low ambient conditions. As chillers for process applications often operate at unchanged cooling loads in summer and winter (or only slightly reduced loads in winter in some cases), the inability of a chiller to vary the condensing temperature in response to low ambient temperature conditions represents a significant penalty for process applications.

• Positive displacement compressors (reciprocating, screw) can generally operate without difficulty at low discharge pressures, such as when chillers operate at low condensing temperatures, and are hence preferred for process applications.

• Flooded evaporators are unaffected by lower condensing pressures (unlike direct expansion evaporators), and are therefore also preferred for process applications. Where direct expansion chillers are fitted with electronic expansion valves (EEVs), the capacity limitations at low condensing pressures can be largely overcome. Flooded evaporators generally require a greater charge of refrigerant, which poses environmental risks in the case of HFC and HCFC refrigerants or safety considerations in the case of ammonia. It should be noted, however, that low-charge flooded ammonia process chillers that comply with Australian state and federal safety codes are now available.

iii. Chillers are often selected for their capacity at full load, without fully considering the operation of the chiller at cooling loads less than 100%. Low-cost chillers, in particular, often display substantial efficiency penalties at lower loads.

• Some chillers use a combination of evaporators each running on a separate refrigeration circuit or compressor, and then turn the compressors off as load diminishes. This can reduce chiller efficiency unless the water flow through the unused evaporators is also stopped. Hence such chillers will show a distinct reduction in efficiency during part load operation.

• Some chillers use hot-gas bypass techniques to falsely load compressors during low-load operation to prevent compressor short-cycling under these conditions. Such techniques substantially reduce chiller efficiency and chillers using such techniques should be avoided.

• Efficient chillers generally use a single large evaporator, which then operates at smaller temperature differentials between the fluid and refrigerant at part load. This allows the refrigerant evaporation temperature to increase without affecting the fluid temperature. Such designs, if coupled with an efficient compressor control system (such as speed control), can improve chiller efficiency at part load.

Chiller selection

There is often a disparity between chiller designs. Chillers suited to a particular application, such as commercial air-conditioning may be inappropriate in other applications, such as chilled glycol/water for dairy process cooling.

When selecting the fluid chiller, consideration must be given to:

i. Fluid temperatures – If the chiller is expected to operate at non-standard conditions, ensure the chiller is suited to these conditions, or order a specially designed unit for the non-standard conditions to avoid energy penalties.

ii. Annual load profile – If the chiller is expected to provide cooling throughout the year, even during low ambient (winter) conditions, ensure the chiller can capitalise on the cooler conditions by operating at low condensing temperatures.

iii. Daily load profile – Unless the chiller is expected to operate at full load at all operating times, select a chiller with good part load performance. A well-designed chiller should exhibit substantially higher efficiency at part load than at full load, not the other way round.

iv. Refrigerant choice – Natural refrigerants such as ammonia (R717), carbon dioxide (R744) and hydrocarbons (R600a, R290, R1270) offer a future-proof option as these refrigerants have no environmental impacts and will be unaffected by any carbon price in the future. HCFC based chillers (R22) should not be considered for new applications at all, as these gases are ozone depleting and scheduled for phase-out under the Montreal Protocol. HFC-based refrigerants (R134a, R407C, R404A, R507) are all high global warming potential gases, and will be severely affected by any future price on carbon, which will potentially increase the maintenance costs of these units.
Achievable savings

Chiller efficiency is substantially influenced by the fluid temperatures, part-load efficiencies and the ability of the chiller to benefit from low ambient temperatures. It is impossible to generalize on energy savings potential, other than to offer the following points of guidance.

i. Modern, high-efficiency chillers exhibit a coefficient of performance (COP) under full load at standard conditions as high as 6, with part-load values up to 12 as standard. Some low-cost chillers do not exceed COP values of 4 at full load, and these values reduce to below 2 at part-load in some cases. An inefficient chiller can incur power penalties between 25% and 400%, depending on usage, when compared to an efficient counterpart. This extreme variance on the market is often not well understood, or even believed.

ii. A chiller designed to fully utilise low ambient temperatures can exhibit power consumption reductions as high as:
- 30% at typical Sydney winter conditions (10°C), compared to chillers that may limit condensing temperature to above 22°C, which represents a typical lower limit for centrifugal chillers:
  - 50% compared to chillers that may limit condensing temperature to above 25°C, which represents a typical limit for direct expansion screw chillers using R134a.

iii. An ammonia-based process chiller operating at a supply temperature of -5°C will typically consume 30% to 50% less power than an equivalent R407C, R404A or R507 chiller operating at summer conditions, and 50% to 60% less at winter conditions (10°C). Refer also to Figure 25.

Very significant energy savings can be achieved with appropriate chiller selection, and the additional capital cost of dedicated process chillers can be amortised in a short time in many process cooling applications.

To illustrate the effect of refrigerant choice, a comparison of process chillers, one using ammonia and the other R404a, was conducted using the assumptions as detailed in the table below. The results for Sydney ambient conditions are shown in Figure 24.

<table>
<thead>
<tr>
<th>Property</th>
<th>R717 Chiller</th>
<th>R404A Chiller</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigeration Capacity [kW]</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>Glycol supply/return temperature [°C]</td>
<td>-5/0</td>
<td>-5/0</td>
</tr>
<tr>
<td>Load profile</td>
<td>100% during 24/7</td>
<td>100% during 24/7</td>
</tr>
<tr>
<td>Cooling water approach to WB [°C]</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Evaporation temperature[°C]</td>
<td>-7</td>
<td>-9</td>
</tr>
<tr>
<td>Superheat[°C]</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Subcooling [°C ]</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td>Compressor</td>
<td>Widely used industrial ammonia reciprocating compressor</td>
<td>Widely used commercial reciprocating compressor</td>
</tr>
<tr>
<td>Minimum condensing temperature [°C]</td>
<td>15</td>
<td>25</td>
</tr>
<tr>
<td>Condensing approach to avg water [°C]</td>
<td>3</td>
<td>5</td>
</tr>
</tbody>
</table>
Figure 24: Effect of refrigerant choice on COP of process chiller
Technology 12: Chilled water/glycol circuit design and control

The performance of systems using chilled water or chilled glycol and water depends on the suitability of the chiller and on the design and operation of the circuit (often referred to as the ‘field’).

Where the field design is ineffective, part-load and even full load operation of the chiller can be compromised, resulting in reduced available capacity and increased power consumption.

Fundamentally the field design and control needs to match the chiller to the load in the most effective and efficient manner. If fluid flows in the system are not modulated or balanced, mixing of cold and warm fluid streams can result. As a result, the fluid return temperature to the chiller is reduced unnecessarily, compromising both the effective cooling performance and the energy efficiency of the chiller.

Principles

Chillers are designed such that, in normal operating conditions, the temperature differential across the chiller (the temperature difference between fluid return and supply temperature) is generally in the range of 4°C to 6°C with the fluid flow rate at the design value. The refrigeration capacity of the chiller is then given by the following relationship:

\[ Q = m' \times C_p \times \Delta T \]

where:
- \( Q \) = chiller refrigeration capacity [kW]
- \( m' \) = mass flow rate of chilled fluid [kg/sec]
- \( C_p \) = specific heat capacity of chilled fluid [kJ/Kg K]
- \( \Delta T \) = temperature differential between fluid entering and leaving the chiller [°C or K]

The temperature differential is essential to maintain chiller efficiency; if the chiller operates at lower differential temperatures, by increasing the fluid flow rate, for example, the mean fluid temperature in the chiller is reduced. This in turn causes the chiller to operate at lower evaporation temperature to maintain refrigeration capacity, thereby reducing chiller efficiency and raising power consumption.

Figure 25: Example of 3 way valve control mixing warm and cold fluid streams
At all times, the overriding principle of field design and control is to maximise the fluid return temperature to the chiller under all operating conditions. This has several significant implications for system design, as detailed below.

i. The flow rate through the chiller must be reduced in proportion to the actual cooling load. If the load drops by 50%, the flow rate is also reduced by 50%, so as to maintain the return temperature to the chiller. This maintains or increases efficiency, given that good chillers have higher part load efficiency. If the flow rate is not adjusted appropriately, lower return temperatures negate the benefit(s) of increased chiller part load efficiency.

ii. The flow of chilled fluid to devices that use cold fluid should be controlled, by turning the flow on or off when these devices are in use or off, respectively, or by modulating the flow to the device in proportion to the instantaneous cooling load of the device.

iii. Eliminating all forms of bypass flow in the system. Bypass flow control allows “unused” cold fluid to return to the chiller or buffer tank via the return line, effectively causing the cold fluid to mix with the heated fluid in the return line, resulting in a reduction of return temperature and therefore chiller performance as mentioned above. Refer to the example shown in Figure 25.

Ideally, as the field load reduces, so should the fluid flow rate. This maintains the temperature differential across the chiller, maintaining chiller efficiency. This can be achieved with a single circuit arrangement (Figure 26) or with a primary/secondary arrangement (Figure 27) with the following key modifications, as detailed below.

- Prioritising use of 2-way modulating or on-off control and avoiding use of 3-way bypass control or constant flow arrangements. This achieves variable system flow in proportion to the load.
- Use of a variable-speed pump to deliver a variable flow proportional to the load.
- Use of an end-of-line pressure sustaining valve to maintain pressure when the pump reaches minimum speed (generally 20 Hz). This causes some unavoidable mixing of supply and return flow.

---

**Figure 26:** Typical basic chilled water system with different commonly used flow control arrangements
In some cases it is not possible or practical to regulate the flow through the chiller to match the flow through the field. Situations where this can occur include:

i. if the field load can reduce to very low loads, say 10%-20% of design capacity, but where the chiller cannot operate at such low flow rates, say only to 40% of design

ii. if the chiller efficiency falls at reduced load, as applicable to some chillers. Rather than operate the chiller unloaded, it would be preferable to operate the chiller intermittently at full load, so as to operate at optimum efficiency.

Under these circumstances, and keeping to the principle of maximising fluid return temperatures, a stratified buffer tank, or a number of smaller tanks connected top to bottom in series are required, as shown in Figure 27. Primary and secondary pumps are used to separately provide controlled fluid flow to the chiller and field, respectively. Efficiency is maintained by cycling the chiller and primary pump on and off in response to the accumulated temperature levels in the buffer tank. In effect the field consumes the chilled fluid at a variable rate to suit the actual load condition, and the chiller operates at optimum efficiency and cycles on as needed to replenish the supply of chilled fluid in the tank.

Several variations on this design are feasible, single stratified tank, multiple smaller tanks in series, etc. Selecting the best tank arrangement depends on various considerations, including site space constraints.

**Primary/secondary circuit arrangement**

![Diagram of Primary/secondary chilled water system with different commonly used flow control arrangements](image)

**Figure 27: Primary/secondary chilled water system with different commonly used flow control arrangements**

**Plant benefits**

Plant benefits include maximised efficiency of the chilled water system at all load conditions.

**Achievable savings**

Often, the field is designed such that a chiller is subject to constant fluid flow rates and must use bypass flow control at the cooling stations to maintain chiller flow. For reasons explained above, this is inefficient. Substantial energy savings are achievable by converting the system to variable flow and two-way flow control at the cooling stations. The savings achievable depend on:

i. the part-load efficiency of the chiller. Higher savings are achieved if the chiller exhibits higher efficiency at part load than at full load

ii. the extent of bypass flow control and constant flow used at the cooling stations in the system

iii. annual and daily load profiles.

Typically, chiller power savings in the range of 5% to 20% are achievable by maximising fluid return temperatures.

Implementation costs are expected to be minimal in most cases, but may require some investment in improved control systems or extensive modifications to the flow control devices in others. Under some circumstances the system may need to be converted to a primary/secondary system due to the operational limits of the chiller.

Given the scope of this technology, it is not possible to generalize on the extent of savings achievable, the costs associated, nor the payback period of such projects.
Technology 13:
Heat recovery from chillers and chiller-heat pump units

Chillers conventionally are used to generate chilled water or a chilled mixture of glycol and water. They reject the heat generated by condensation to the environment via air-cooled condensers or cooling towers. Any hot water requirements are generally met by means of a separate hot water generator arrangement.

Modern chillers, in particular those using ammonia (R717) or carbon dioxide (R744) refrigerant, offer significant potential to recover otherwise wasted heat at useful temperature levels, i.e. greater than 50°C. This recovered heat can be used to offset the energy consumption of other operations, thus reducing site energy consumption.

Combined chiller and heat pump units generate chilled fluid and hot water simultaneously by elevating the heat rejection temperature sufficiently to heat cold water to temperatures greater than 60°C. A chiller-heat pump unit will reject heat to the environment only if the instantaneous heat produced by the chiller-heat pump exceeds the instantaneous heating demand.

Chiller/heat pump units

Combined chiller and heat pump units are commercially available either as two-stage units using ammonia as refrigerant, or as single-stage transcritical units using carbon dioxide as refrigerant. Ammonia chiller-heat pump units can generate water up to 70°C, whilst CO2 heat pumps can provide hot water at up to 90°C. However, heating and cooling efficiencies, as well as capital costs differ substantially for the two designs. A careful analysis of the cooling and heating demands as well as power and heating costs on a given site must be evaluated to determine the solution best suited to the application in question.

Plant benefits

Plant benefits include reduced site heating demand as recovered heat can offset demand for hot water generation, generally less than 60°C.

Achievable savings

The achievable annual savings depends on:

i. refrigerant used in the chiller. Ammonia or carbon dioxide chillers offer more heat recovery at higher temperatures than, for example, R134a or R407C chillers

ii. required water temperature. Warm water needs at 40°C are easily met with heat recovery from most chillers, but where hot water at greater than 55°C is routinely required, ammonia process chillers are better suited than HFC-based alternatives.
### Estimated financial returns

<table>
<thead>
<tr>
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<th>Estimated cost</th>
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<td><strong>Total</strong></td>
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<table>
<thead>
<tr>
<th>Energy savings [GJ/year]</th>
<th>Energy cost savings [$/year]</th>
<th>Project cost [$]</th>
<th>Payback [years]</th>
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<td>1,650</td>
<td>$9,900</td>
<td>$26,000</td>
<td>2.6</td>
</tr>
</tbody>
</table>

1. The equipment allowed in the above costing includes a discharge line de-superheater (heat recovery heat exchanger), a variable speed controlled hot water pump, a 5000 litre insulated stainless steel hot water tank, insulated hot water pipe-work, fittings and controls.
2. The indicated energy savings is based on a 500 kW chiller with reciprocating compressors that can provide up to 60 kW of heat recovery at 35°C condensing temperature.
3. The chiller is assumed to run at an average load of 70% and that all recovered heat can be used.
4. Average cost of gas paid by facility is assumed as $6 per GJ.
Technology 14: Variable chiller fluid temperatures

It is common industry practice to operate chilled water or glycol/water systems at fixed temperature set points, without consideration of ambient conditions or cooling loads.

In some situations these set point temperatures can be increased without a negative effect on the process being cooled, which has the effect of increasing the chiller efficiency and reducing power consumption.

Principles
Refrigeration systems use substantially less power if the evaporation temperature at which the system operates can be increased. Typically, power consumption reduces by 3% for every 1°C increase in evaporation temperature and hence, fluid temperature. A 3°C increase in evaporation temperature can result in energy savings of 9%. Some applications, however, do not permit significant variation in fluid temperature. For example, many food processing applications, such as milk processing, use chilled water or water/glycol to cool product via a heat exchanger where steady fluid temperatures are essential in maintaining product quality and hygiene.

However, many other applications can tolerate seasonal or load-dependent variation of fluid temperature. For example, when chilled water is used for cooling of processing rooms, the reduced cooling loads and lower ambient humidity levels allow chilled water temperature to be set higher during winter. In the case of fruit storage rooms cooled by glycol/water systems, the reduced cooling load during winter generally allows the fluid temperature to be increased by 2-3°C in winter as compared to summer operation.

Achievable savings
Where an existing chiller operates at essentially constant set point temperatures and the temperature can be made to vary under the range of operating conditions, substantial energy savings can be achieved. These depend on the:

i. nature of the application
ii. annual and daily load profile
iii. ambient conditions.

In some situations, significant variation of fluid temperatures is achievable with annual power savings as high as 20%, in others this cannot be tolerated and little potential exists.

Implementation costs are expected to be minimal in most cases (reprogramming of controls), but may require some investment in improved control systems in others.

Given this technology, it is not possible to generalize on the extent of savings achievable, the costs associated, nor the payback (ROI) of such projects.
Technology 15: Variable cooling water temperatures

It is common industry practice to design and operate cooling water systems at standard conditions such as 29.5°C inlet and 35°C return temperatures, without consideration of ambient conditions.

This often leads to cooling water systems operating at temperatures of 25°C or above, even under winter conditions, with ambient air temperatures of 10°C or lower. This represents a substantial lost opportunity for energy savings.

Principles
Refrigeration systems use substantially less power if the condensing temperature at which the system operates can be reduced. Typically, power consumption reduces by 3% for every 1°C reduction in condensing temperature. Hence a 10°C reduction in condensing temperature can result in energy savings of 30%. Refer to Figure 18 in regard to this.

Water cooled chillers require a cooling system to remove the condensing heat. Generally this consists of a water pump, a cooling tower and piping to circulate water through the heat exchanger on the chiller and the cooling tower. The cooling water temperature is dependent on the ambient wet bulb temperature and the rate of air circulation through the cooling tower. In summer conditions (in Sydney this could be 25°C wet bulb temperature), cooling water temperatures of 27°C or higher are achievable, whilst in winter conditions (<5°C wet bulb temperature), cooling water temperatures below 10°C could be achieved at full air flow rate. However, in most systems the air flow rate is modulated by controlling the cooling tower fan speed or by bypassing the cooling tower to maintain cooling water temperature above a set minimum, typically 15°C to 20°C, sometimes higher.

Where the chiller design permits the condensing temperature to float to low values without negative impact on the chiller reliability or refrigeration capacity, considerable energy savings can be achieved by allowing the cooling water temperature to reduce to very low values, although this has to be well above 0°C to prevent any problems with ice formation during winter conditions.

In some cases, cooling water systems are shared within a facility, and cooling water is provided to a range of devices requiring cooling. In some cases these other devices require constant cooling water temperatures and hence the design of the cooling water system on site may need to be reviewed to ensure that these other devices are not affected by excessively low cooling water temperatures.

Achievable savings
Where an existing water-cooled chiller operates at essentially standard cooling water conditions, greater than 25°C, the cooling water temperature can “float” to suit ambient conditions; hence substantial energy savings are achievable. These savings also depend on:

i. the annual and daily load profile
ii. the suitability of the chiller for floating condensing temperature operation.

Typically annual savings of 5% to 15% can be achieved for process cooling applications if cooling water temperatures are fully floated. Implementation costs could be minimal in some cases, such is the reprogramming of controls, but may require some changes to the chilled water circuit in other cases, adding further costs.

Given this technology, it is not possible to generalise on the extent of savings achievable, the costs associated, nor the payback period of such projects.
Conclusion

With rising energy costs and increasing environmental awareness, there is a growing need to reduce power consumption in the industrial refrigeration sector in NSW. The Office of Environment and Heritage has established Energy Saver Industrial Refrigeration which aims to promote the widespread adoption of cost-effective, commercially proven and energy efficient technologies throughout key industries, such as abattoirs, bakeries, cold stores, dairy processors, food and beverage manufacturers, meat packers and processors, pet food manufacturers and wineries.

This report shows that the energy costs of a refrigeration plant could be reduced by around 40 per cent, possibly more, through adoption of the most energy efficient equipment and techniques. As shown, the various energy saving technologies defined in this document are applicable to most industrial refrigeration facilities, and many of these technologies comprise control related upgrades and are best implemented together to maximise energy savings.

Appendix I:
Usage profile considered for energy savings calculations

The usage profile illustrated in Figure 28 is typical of industrial refrigeration facilities. It has been considered to calculate annual energy savings for many of the technologies mentioned in this document. The usage profile considers the portion of time of the year that the plant operates at a specific part load, at 10% part load increments.

Annual usage profile

Figure 28: Annual refrigeration plant usage profile
## Appendix II: Glossary of terms

<table>
<thead>
<tr>
<th>Term</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bottleneck</td>
<td>A plant design feature that forces the rest of the plant to run inefficiently</td>
</tr>
<tr>
<td>CMS</td>
<td>Control management system</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance = ratio of cooling duty to consumed power</td>
</tr>
<tr>
<td>DECCW</td>
<td>Department of Environment, Climate Change and Water (NSW)</td>
</tr>
<tr>
<td>DX</td>
<td>Direct expansion</td>
</tr>
<tr>
<td>EEV</td>
<td>Electronic expansion valve</td>
</tr>
<tr>
<td>Field</td>
<td>Heat exchangers outside of plant room doing useful cooling work, and connected to plant via piping system</td>
</tr>
<tr>
<td>HCFC</td>
<td>HydroChloroFluoroCarbon (a synthetic, ozone-depleting and high global warming refrigerant)</td>
</tr>
<tr>
<td>HFC</td>
<td>HydroFluoroCarbon (a synthetic, high global warming refrigerant)</td>
</tr>
<tr>
<td>IPLV</td>
<td>Integrated part load value</td>
</tr>
<tr>
<td>ISPC</td>
<td>Inter-stage pressure control</td>
</tr>
<tr>
<td>OEH</td>
<td>Office of Environment and Heritage (formerly DECCW)</td>
</tr>
<tr>
<td>PLC</td>
<td>Programmable logic controller</td>
</tr>
<tr>
<td>Purging</td>
<td>Removal of non-condensable gases or water from a refrigeration system</td>
</tr>
<tr>
<td>R134a</td>
<td>HFC refrigerant commonly used for medium/high temperature applications</td>
</tr>
<tr>
<td>R404A</td>
<td>Blended HFC refrigerant commonly used for low temperature applications</td>
</tr>
<tr>
<td>R407C</td>
<td>Blended HFC refrigerant commonly used for medium temperature applications</td>
</tr>
<tr>
<td>R507</td>
<td>Blended HFC refrigerant commonly used for low temperature applications</td>
</tr>
<tr>
<td>R717</td>
<td>Ammonia, a commonly used natural refrigerant</td>
</tr>
<tr>
<td>R744</td>
<td>Carbon dioxide, a natural refrigerant increasingly being used for low and medium temperature applications</td>
</tr>
<tr>
<td>SCADA</td>
<td>Supervisory control and data acquisition system</td>
</tr>
<tr>
<td>SCT</td>
<td>Saturated condensing temperature</td>
</tr>
<tr>
<td>SST</td>
<td>Saturated suction temperature</td>
</tr>
<tr>
<td>Static</td>
<td>Single point / one time; as in static implementation: control system with logic adjustable only by local program changes, static optimisation: control system with logic optimised for one operating condition only</td>
</tr>
<tr>
<td>VHPC</td>
<td>Variable head pressure control</td>
</tr>
<tr>
<td>VPPC</td>
<td>Variable plant pressure control</td>
</tr>
<tr>
<td>VSD</td>
<td>Variable speed drive</td>
</tr>
</tbody>
</table>