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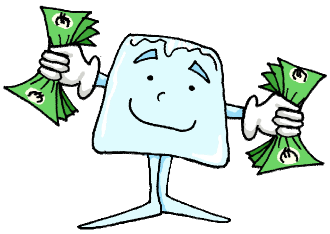

Guide 3

Operational Efficiency Improvements for Refrigeration Systems

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Operational Efficiency Improvements for Refrigeration Systems

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The Food & Drink Industry Refrigeration Efficiency Initiative

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Carbon Trust Networks Project

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Operational Efficiency Improvements for Refrigeration Systems

This guide will help you check the performance of refrigeration equipment to:
1. Make savings by maximising operating efficiency.
2. Identify and correct energy wasting plant faults.

1. Introduction

Refrigeration systems often represent the largest electricity user in food and drink factories. Keeping refrigeration operating at best efficiency is very important but is often ignored because users are often unsure how to check their refrigeration plant performance.

To most end users the primary concern is plant reliability and, in particular, the ability to cool products to the desired temperature. If this is being achieved then it is often assumed that the plant is operating in a satisfactory manner. But is the plant running efficiently?

The Importance of Measuring Plant Performance

Compare your refrigeration plant to a car. If a car manages to complete a journey from A to B on time then it has "delivered" its primary output. But you might have used 2 gallons of petrol when 1 gallon should have been enough. If you don't measure the m.p.g. for the journey it is impossible to ensure that the car is running well and has been driven properly. If the m.p.g. is lower than expected you will take steps to improve performance. If you don't regularly measure the "m.p.g." of your refrigeration plant it is impossible to ensure that it is running well. As with a car, a secondary benefit relates to improved reliability. You can spot faults before they become so bad that the plant breaks down!

This Guide is aimed at users of industrial refrigeration systems in the food and drink sector. It provides advice about how to measure performance and how to make a refrigeration plant run as efficiently as possible. You do not need to be a refrigeration engineer to use this Guide. The technical terms used are explained in the glossary in Appendix 1 and sources of further information are given in Appendix 2.

Measuring and Improving Refrigeration Plant Performance is not a Trivial Task!

The assessment of refrigeration plant performance and the diagnosis of specific energy wasting faults is not easy. Refrigeration equipment is amongst the most complex found in food and drink factories – without a reasonably in-depth understanding of refrigeration it is hard to identify all the possible faults. This Guide is not intended to provide detailed background about how refrigeration plants work – although we do provide references to more detailed documents that do this. The Guide is intended to provide an overview of the key issues to enable you to either:

- Employ outside experts to help you ensure your plant is running at high efficiency.
- Identify and fill the knowledge gaps within your organisation.

This Guide is one of a series of eight being produced under the Food & Drink Industry Refrigeration Efficiency Initiative, a project sponsored by the Carbon Trust and supported by the Food and Drink Federation, the British Beer and Pub Association, the Cold Storage and Distribution Federation, Dairy UK and the Institute of Refrigeration.

2. Two Different Approaches to Performance Measurement

Measuring plant performance can be difficult, and experience has shown that two complementary monitoring strategies give the best results in ensuring maximum plant efficiency. These are:

Strategy 1: Indirect Assessment of Plant Faults. This involves assessment of the performance of individual items of plant, such as condensers, to identify specific types of fault that need to be remedied. This approach usually involves taking a "snapshot" of instantaneous data (e.g. temperatures and pressures) and comparing these data with "expected values". By understanding the interrelationship of different parameters being measured you can diagnose different types of plant fault. This strategy can be very powerful as it helps spot the exact cause of the wasted energy. The types of faults that can be diagnosed are described in Section 4 of this Guide.

Strategy 2: Direct Monitoring of Performance. This involves measuring the power input into the plant over fairly long periods of time (e.g. weekly) and estimating the amount of cooling done in the same period, either by direct measurement or through calculation. This strategy allows you to build a comprehensive picture of plant performance over time. Further details are given in Section 5 of this Guide. Ideally, you should adopt both strategies, although Strategy 1 can often be adopted more quickly and with less investment in metering.

Two strategies are often used for the measurement of boiler performance.

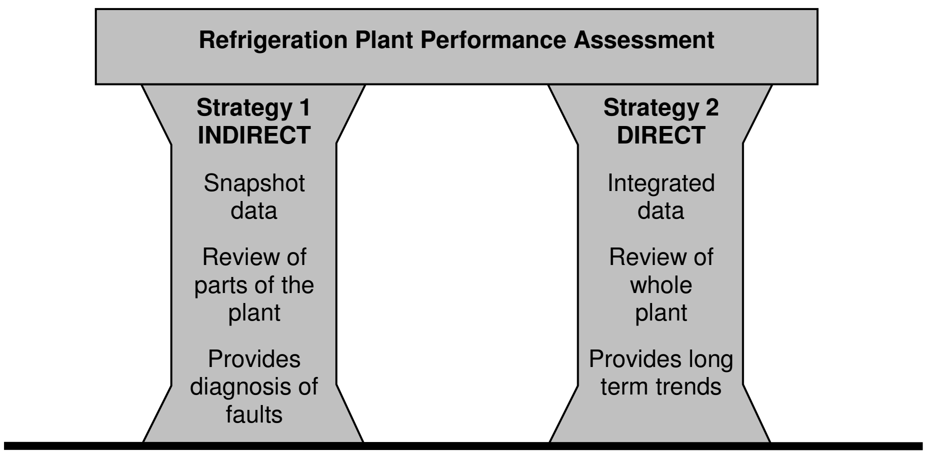
The two strategies described above involve similar principles to the strategies often used to monitor boiler performance.

Strategy 1 is equivalent to indirect measurements such as a boiler "losses test". You measure a snapshot of the flue gas temperature and oxygen content and you can infer the boiler efficiency. The values measured can point towards a fault e.g. high oxygen content implies too much combustion air whereas a high flue gas temperature could mean there is fouling inside the boiler.

Strategy 2 is equivalent to measuring the steam output and boiler fuel input in order to calculate the boiler efficiency. These measurements are often made over fairly long periods (e.g. weekly) using appropriate steam and fuel meters. If the efficiency is lower than expected then remedial action is indicated.

It is important to note that the boiler strategies are complementary – it is better to use them both than just to select one. The same applies to refrigeration plant!

Figure 1 The Twin Pillars of Refrigeration Plant Performance Assessment



3. What Data Should Be Measured?

A key starting point is to ensure that you are measuring the appropriate data. The requirements are quite different for the two strategies described in Section 2. A well instrumented refrigeration plant will provide access to a number of instantaneous values (such as temperatures, pressures, amps etc.) and some long term values (such as compressor kWh). Instantaneous values are required for Strategy 1, whilst Strategy 2 requires various long term values to be measured.

For Strategy 1, Indirect Assessment of Plant Faults, the minimum measurement requirements are illustrated in Figure 2. The key parameters are listed in Table 1.

Figure 2 Key Parameters for Assessment of Plant Faults

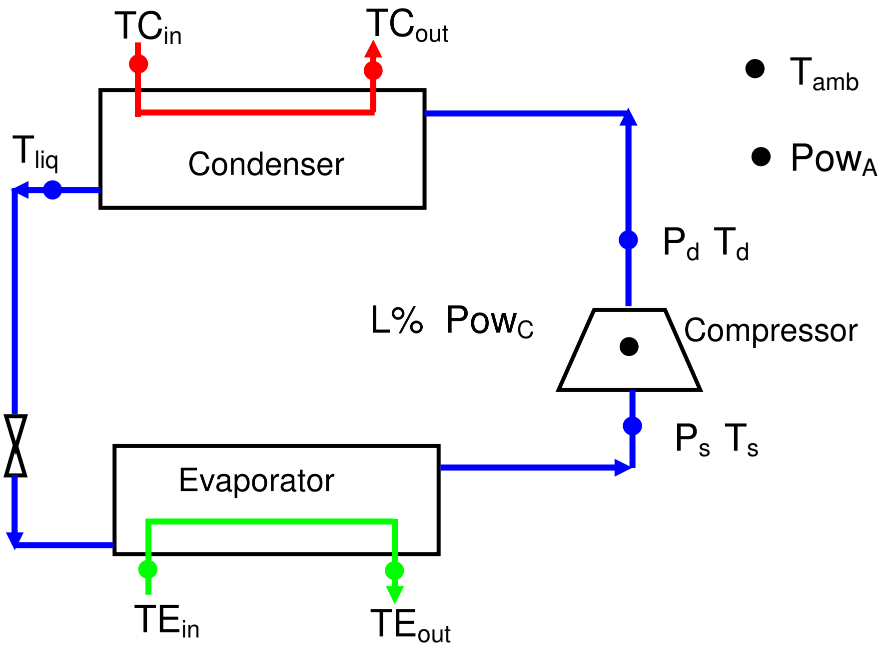


Table 1 "Instantaneous" Measurements Required for Strategy 1

Parameter	Symbol in Figure 2	Comments
Cooling load temperatures	TE_{in} and TE_{out}	Process stream temperatures at inlet and exit of the evaporator (e.g. air on/off for a cold store cooler or liquid in/out for a liquid chiller)
Heat rejection temperatures	TC_{in} and TC_{out}	Fluid temperatures at inlet and exit of the condenser (air or cooling water)
Ambient temperature	T_{amb}	Dry bulb for air-cooled condensers, wet bulb for evaporative or water cooled condensers.
Compressor suction and discharge pressures and temperatures	P_s and P_d T_s and T_d	Gauge pressures at compressor suction and discharge. Actual refrigerant vapour temperature at same locations.
Liquid temperature of refrigerant leaving condenser	T_{liq}	Temperature of liquid in liquid line, after the condenser and receiver but before the expansion valve.
Plant load	L%	Measured in terms of compressor loading.
Power or Amps	Pow_C and Pow_A	The power being absorbed by compressor/s and main auxiliaries (pumps / fans for evaporator and condenser).

The data listed above is the minimum required. Other data that is often of help includes the pressure and temperature conditions at the evaporator outlet and the condenser inlet.

To be able to interpret this data in a satisfactory way you will also need plant design and commissioning data to help establish “expected values” of each parameter under a range of operating conditions. This is discussed in Section 4 and Appendix 5.

The pressure gauges showing the refrigerant pressure at the compressor inlet and outlet are especially important for the assessment of possible plant faults. Experts often refer to evaporating and condensing temperatures even though they are measuring the values with a pressure gauge. Appendix 3 explains how a temperatures and pressures are sometimes equivalent in a refrigeration system.

For Strategy 2, Direct Monitoring of Performance the measurement requirements are:

- Compressor energy input. kWh meter/s to measure compressor energy input.

- Auxiliary energy input. kWh meters or an alternative way of measuring auxiliary energy use. kWh meter are best, but many auxiliary devices such as pumps and fans run at fixed load, so a cheap alternative option is to fit hours run meters to them and make an estimate of energy consumption (e.g. by measuring the electrical current with a portable tester and converting that into power and using hours run to convert to energy).

- Total energy input. Sometimes it is simpler to measure compressor energy and the auxiliaries together. This depends upon the local cabling arrangements, although it is usually also worthwhile measuring the energy consumed by the compressor itself.

- Weather conditions. You must estimate the average ambient temperature. If you have an air cooled condenser it is the dry bulb temperature that should be measured. However, if you have an evaporative condenser or a water cooled condenser with a cooling tower, then the average wet bulb temperature is the relevant parameter to measure.

- Cooling load. You need a way to estimate the amount of cooling being carried out. This is often quite difficult and may require some form of “proxy” variable to be measured. Further details about measuring cooling loads are given in Appendix 4.

4. Strategy 1: Indirect Assessment of Plant Faults

The indirect assessment of faults uses instantaneous values; hence it can be done on an ad hoc basis as required. This could be in response to adverse data from direct monitoring of performance. Alternatively, if direct monitoring is not being carried out you should carry out indirect assessments on a scheduled basis e.g. weekly or monthly.

Taking a Data Snapshot

A snapshot of plant operating data, as illustrated in Figure 2 and Table 1 should be collected. Ideally the plant should be running under steady state conditions and, if possible, the plant should be run at 100% load as it is easier to interpret the data at full load. Avoid taking measurements within a few minutes of plant start up as the parameters may still be climbing towards a steady state value. If the compressor keeps varying in capacity, in response to load variations you may need to adjust a control setting to force the plant onto a steady full load. For example, reduce the temperature set point of a cold store by a few degrees. If you do this, do not forget to return the control to the original setting!

Compare Data Snapshot with Expected Values

This is often the most difficult step because you may be uncertain about expected values. Expected values for each snapshot parameter can vary with ambient temperature, cooling load and product temperature. You may need some expert help to establish tables of expected values. Fairly detailed plant design data is required to calculate expected values of temperature differences and pressures. Most of the required design data should have been supplied when the plant was installed and during commissioning tests. Appendix 5 provides further information on the calculation of expected values.

Look for Fault Symptoms

The data collected should be assessed using the fault check lists shown in Tables 2 to 7. Each part of the plant can be considered separately.

Probably the most common faults are those linked to **condensers** – and they are usually fairly easy to spot. Inefficiencies linked to condenser problems always result in a compressor discharge pressure that is too high. A high discharge pressure reduces the system efficiency whilst at the same time it also reduces the amount of cooling being done. Most condenser faults are associated with a heat transfer problem that causes the condenser to operate inefficiently. Condenser faults are summarised in Table 2.

Evaporator faults are also common and easy to spot. Inefficiencies linked to evaporator problems always result in a compressor suction pressure that is too low. A low suction pressure reduces the system efficiency whilst at the same time it also reduces the amount of cooling being done (sometimes by a significant amount). Most evaporator faults are associated with a heat transfer problem that causes the evaporator to operate inefficiently. Evaporator faults are summarised in Table 3.

Compressor faults can be more difficult to spot. Compressor problems can relate to mechanical damage inside the compressor or to undesirable pressure drops due to blockage. Compressor faults are summarised in Table 4.

Expansion valve problems can be linked to the valve being open too much (leading to unwanted bypass of high pressure vapour through the valve) or being closed too much (leading to starvation of liquid feed to the evaporator). Expansion valve faults are summarised in Table 5.

There are numerous **controls** on refrigerant plants that could be set incorrectly or that could be operating badly. Control faults are summarised in Table 6.

The **cooling loads** themselves need to be checked to ensure they are not higher than necessary. Cooling load faults are summarised in Table 7.

Table 2: Common Faults Associated with Condensers

General Comments on Condenser Faults. Inefficiencies linked to condenser problems always result in a compressor discharge pressure that is too high. This reduces the system efficiency whilst at the same time it also reduces the amount of cooling being done. Most condenser faults are associated with a heat transfer problem that reduces the condenser's ability to reject heat.		
Symptom	**Possible Fault**	**Required Action**
High P_d, plus visible fouling, corrosion, damage or blockage (for an air cooled condenser), or high water side pressure drop (for a water cooled condenser).	External fouling.	Clean condenser.
High P_d combined with too much liquid sub-cooling (low T_{liq}).	Build up of non-condensable gases in condenser (usually air).	Purge air from condenser.
	Liquid refrigerant back-up inside condenser.	Investigate reason for liquid back-up; e.g. overcharge of refrigerant or incorrect condenser piping configuration.
High P_d combined with high temperature difference between TC_{out} and TC_{in}.	Inadequate air or water flow.	Check for pump or fan failure; check for blockage in flow path.
High P_d combined with TC_{in} being higher than Tamb.	Warm air recirculation.	Modify condenser layout.
High P_d combined with TC_{in} being warmer than expected for the prevailing wet bulb temperature.	Poor cooling tower performance.	Check cooling tower for fan/pump failure, poor water distribution or fouling.
High P_d plus head pressure control mechanism operating at higher Tamb than expected.	Incorrect head pressure control setting.	

Table 3: Common Faults Associated with Evaporators

General Comments on Evaporator Faults. Inefficiencies linked to evaporator problems always result in a compressor suction pressure that is too low. A low suction pressure reduces the system efficiency whilst it also reduces the amount of cooling (sometimes by a significant amount). Most evaporator faults are associated with a heat transfer problem that causes the evaporator to operate badly.		
Symptom	**Possible Fault**	**Required Action**
Low P_s plus visible signs of frost on cooling coil.	Excessive Frost.	Carry out defrost; reduce rate of moisture ingress or improve defrost control.
Low $P_{s.}$	Refrigerant side oil fouling.	Drain oil from evaporator
Low P_s and high $T_{s.}$	Lack of refrigerant.	Adjust expansion valve or add refrigerant.
Low P_s, plus visible fouling or blockage (for an air cooler), or high liquid side pressure drop (for a liquid cooler).	External fouling.	Clean evaporator.
Low P_s combined with high temperature difference between TE_{out} and TE_{in}.	Inadequate air or liquid flow.	Check for pump or fan failure; check for blockage in flow path.

Table 4: Common Faults Associated with Compressors

General Comments on Compressor Faults. Compressor problems can relate to poor compressor efficiency due to mechanical damage inside the compressor or to undesirable pressure drops due to blockage.		
Symptom	**Possible Fault**	**Required Action**
Lower Pow_c than expected for measured P_s and P_d, together with high Td	Suction strainer blockage	Check pressure downstream of strainer, e.g. crankcase or shell. Clear blockage in suction strainer
High T_d.	Broken discharge valves on reciprocating compressor	Repair valves
Poor system efficiency on a screw compressor plant and lack of other identified problems.	Compressors running unnecessarily on part load or Damaged tip seals on screw compressor. This is difficult to diagnose directly as oil cooling masks expected high T_d.	Check at full load if possible. Check capacity control sequence. Repair rotors
Compressor running at high load (L%) despite the need for less cooling	Faulty unloading system	Repair unloading system
Large pressure difference between evaporator and suction port or between discharge port and condenser.	Blockage in suction or discharge pipework.	Check suction or discharge pipework for blockage (e.g. partly closed stop valve).

Table 5: Common Faults Associated with Expansion Valves and Liquid Lines

General Comments on Expansion Valve Faults. Expansion valve problems can be linked to the valve being open too much (leading to unwanted bypass of high pressure vapour through the valve) or being closed too much (leading to starvation of liquid feed to the evaporator).		
Symptom	**Possible Fault**	**Required Action**
High P_s. Gas bubbles in liquid line sight glass (if fitted). Lower cooling duty than expected.	Hot gas by-pass between condenser and evaporator.	Identify and fix cause of by-pass (e.g. too little refrigerant in HP liquid receiver; float stuck on HP float valve; too little refrigerant in flooded evaporator).
Low level in evaporator. High suction superheat (T_s high).	Insufficient liquid flow through expansion valve.	Adjust superheat setting (for thermostatic expansion valve); check for blockage in valve or incorrect control action. Check for correct orifice size.
Suction superheat varying rapidly (T_s varying) on a plant with thermostatic expansion valve.	Expansion valve not operating correctly.	Check superheat setting (could be set too low). Check TEV orifice size (could be too large).
Bubbles in liquid line sight glass.	Excess pressure drop in liquid line or heat gain into liquid line.	Check for a blockage e.g. blocked filter drier or partially closed stop valve. For long liquid line, check diameter is large enough to avoid excessive pressure drop. If liquid line passes through warm area (warmer than condensing temperature) insulate line to avoid heat gain.

Table 6 Common Faults Associated with Controls

General Comments on Control Faults. There are numerous controls on refrigerant plants that could be set incorrectly or that could be operating badly.		
Symptom	**Possible Fault**	**Required Action**
Product temperature low. Plant running even though target temperature achieved.	Incorrect temperature controller.	Check setting of main temperature controller; check calibration and location of temperature sensor.
In cool weather P_d is higher than expected.	Incorrect head pressure control setting.	Check head pressure control settings and ensure they are set at lowest practical level.

Table 7 Common Faults Associated with Heat Loads

General Comments on Heat Load Faults. The cooling loads themselves need to be checked to ensure they are not higher than necessary.		
Symptom	**Possible Fault**	**Required Action**
Temperature of the product reaching the refrigeration plant is higher than expected. If product is pre-cooled before reaching the refrigeration system there is an “upstream” problem.	Fouled process heat exchanger.	Check upstream process and temperature control settings.
Product being cooled below required temperature.	Product temperature low.	Check temperature control system.
Heat load in a cold store higher than expected; ice build higher than expected.	Excessive heat and moisture ingress through cold store doors.	Improve door closing “discipline”. Check door seals, airlock, strip curtains.
Evaporator fans or pumps run when product has reached target temperature.	Excessive auxiliary power.	Ensure control systems maximise possibilities to switch auxiliaries off or reduce their speed.
Cold store temperature too low.	Incorrect control.	Adjust thermostat.
Cold store temperature too high.	Inadequate cooling.	Check that load is not too high (e.g. doors left open, excessive warm product load). Then ensure evaporators are defrosted. Then check refrigeration plant performance e.g. control system problem, fouled heat exchangers etc.

5. Strategy 2: Direct Monitoring of Plant Performance

The direct assessment methodology uses "integrated"data measured over long periods of time e.g. kWh consumption of compressors. This data is used to estimate plant efficiency which can be compared to expected values.

The starting point is to understand how we express the efficiency of a refrigeration plant.

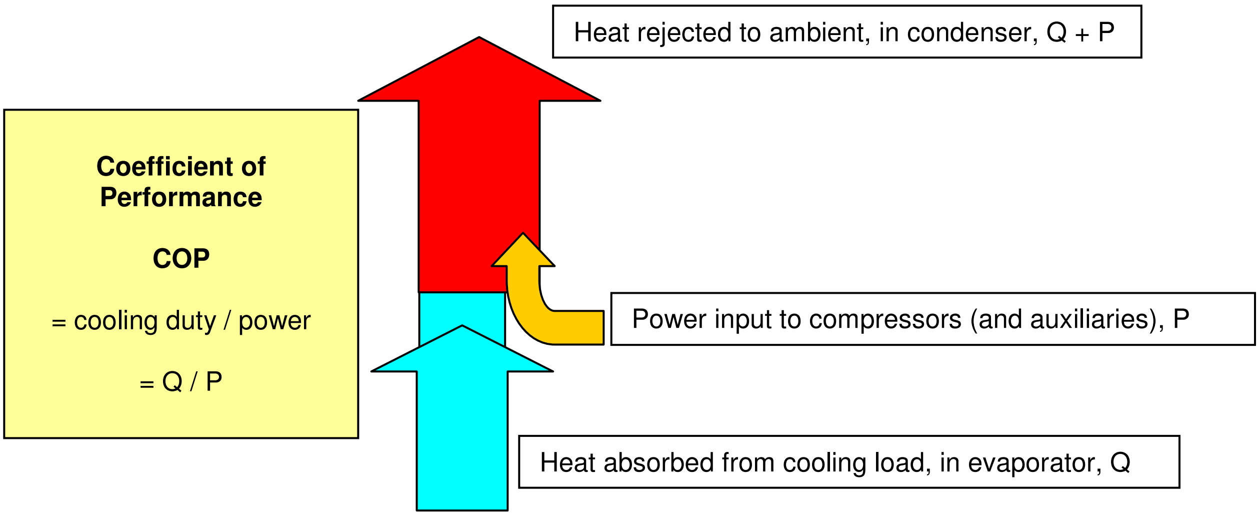
How We Measure Refrigeration Plant Performance

In simple terms, to measure the performance of a refrigeration plant we need to know:

- The amount of cooling being carried out, Q, measured in kW.
- The amount of power consumed, P, also measured in kW.

Then we can calculate the plant efficiency in terms of the ratio of cooling carried out to power consumed (Q/P). This is known as the COP or Coefficient of Performance. This is illustrated in Figure 3.

Figure 3: Heat Flows and Performance



Many refrigeration plant users struggle to understand the concept of COP – the very name "Coefficient of Performance" puts you off before you start! Probably the biggest problem is that the COP is usually greater than 1 (e.g. for a water chiller the COP might be equal to 4). We are used to measuring efficiency on a scale of 0 to 100%, so a COP of 4 is not always an intuitive concept. To explain this, compare a refrigeration plant to a boiler:

- In a boiler we are **converting energy from one form to another**. We start with energy in a fuel and we finish with energy in a flow of steam. It is logical that we will lose a bit of energy in the conversion process. If we express the boiler efficiency as the ratio of the useful energy output (in the steam) to the energy input (in chemical energy in the fuel) then the efficiency will always be less than 1 (or less than 100%).

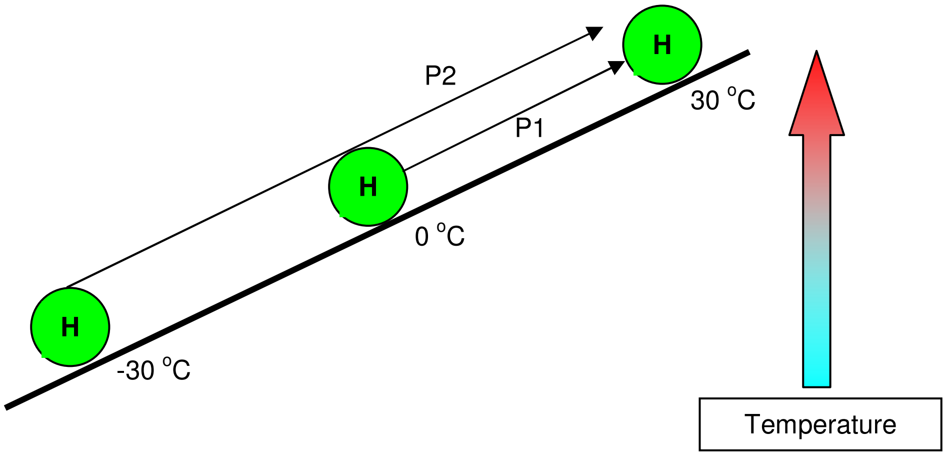
- In a refrigeration plant we are using power to **move energy from a low temperature to a higher temperature.** We are not "generating cold".

We start by absorbing some energy at a low temperature (e.g. removing heat from a warm product placed in a cold store) and we then move that energy to a higher temperature so we can "throw it away" into the ambient.

To move energy "uphill" from a low temperature to a higher temperature requires and input of extra energy. The amount of extra energy input depends mainly on the temperature difference between the cold end and the hot end of the process. This is illustrated in Figure 4 in the "Thermodynamic Mountain".

If the temperature difference is around 30 deg C (typical for a chill store) then 1 kW of input power can "move" about 4 kW of heat up the mountain – this is a COP of 4. If the temperature difference is around 60 deg C (typical for a cold store) then 1 kW of input power can only "move" about 2 kW of heat up the mountain – this is a COP of 2.

Figure 4: COP and the Thermodynamic Mountain



Is Your COP Good or Bad?

Once you have measured your refrigeration plant COP, how do you know whether the performance good or bad? Ideally we want the COP to be as high as possible, because that means we are getting more cooling for each kW of power input. So, what is a good COP? For a refrigeration plant this is not a simple question.

Minimise "Temperature Lift" to Maximise Performance

The thermodynamic mountain shown in Figure 4 helps begin to answer this question: a plant with a low "temperature lift" will be more efficient than a plant with a high temperature lift. This is a fundamental aspect of efficient refrigeration plant operation. To minimise the temperature lift we want:

- The evaporating temperature of the refrigerant to be **as high as possible**
- The condensing temperature of the refrigerant to be **as low as possible**

The evaporating temperature is related to the product temperature – if you want to cool water to 5 °C then the evaporating temperature must be below 5 °C. However, it is also dependant on plant design and maintenance. If a water chiller producing 5 °C water is evaporating at, say, -10 °C then there is probably a design or maintenance related fault because the evaporating temperature is unnecessarily low.

Similarly, the condensing temperature is related to ambient temperature, but is strongly dependant on design and maintenance. There are many types of plant fault that lead to an unnecessarily high condensing temperature.

The COP of a plant depends on 4 main parameters:

a) **The temperature of the product being cooled**. The COP of a plant cooling a product to +5 °C should be much higher than a plant cooling a product to -40 °C.

b) **The ambient temperature**. The COP of a plant running on a cool winter evening should be much higher than on a hot summer day.

c) **The plant design**. For a given product temperature and ambient temperature there can be a significant variation in COP depending on the way the plant is designed.

d) **The plant load**. Most cooling loads vary considerably on an hour by hour basis and can also vary seasonally. The COP of a plant can vary considerably between 100% load and a low load condition such as 20%.

A Small Amount of Extra Temperature Lift Costs £££s!

Your plant's performance is very sensitive to temperature lift. For example, a plant was condensing at 30 °C, but it should have been condensing at 28 °C. The operator thought that this 2 deg C difference was trivial and ignored it. This was not a wise decision! The extra 2 deg C of temperature lift made the plant use 5% more energy than necessary.

A useful rule of thumb is that each unnecessary 1 deg C of temperature lift will waste between 2% and 4% in terms of energy consumption. You need to make every effort to reduce minimise the condensing temperature and maximise the evaporating temperature of your plant.

It is Normal for COP to Vary Quite Considerably

For one particular refrigerant plant many of the parameters that influence COP are fixed. In most situations the plant design is "fixed" and the product temperature is always approximately the same.

However, if you measure a "snapshot" of the plant COP on two occasions it is normal that the COP could be significantly different. This is because the COP is very sensitive to:

- **Ambient temperature** – as described above, a few degrees difference in condensing temperature makes a big difference to COP. The condensing temperature is directly linked to the ambient temperature, which can vary by at least 10 °C across a day and by 30 °C seasonally.

- **Plant load** – under low load conditions the COP might be much lower than expected because of the influence of compressor part load efficiency and high auxiliary loads.

You Need to Establish a Range of "Expected Values for COP"

To interpret any COPs that you measure you need to have an idea of the "expected best COP" for a range of operating conditions. This involves a few difficult calculations so you may need to ask an expert for help. For a new plant you should ask the supplier to prepare a table such as the one illustrated in Table 8 – hopefully they will not charge for this if you build it into your specified requirements! For an existing plant a table like this can be calculated using design and commissioning data if available. Your maintenance contractor may be able to help with the necessary calculations.

Table 8 Expected Compressor COPs (for a plant chilling a room to +2 °C)

Plant Cooling Load	Ambient Temperature °C	Expected Best COP
100%	25	4.2
	15	5.9
	5	7.0
50%	25	4.0
	15	5.5
	5	6.5
25%	25	3.5
	15	4.8
	5	5.5

In the example used in Table 8 we have assumed a very important design feature called "floating head pressure". This allows the condensing temperature to fall significantly as the ambient temperature falls. You can see that with low ambient temperature the COP is considerably higher than in warmer conditions.

Performance Variation is Common

This type of variability in performance is common for many energy using systems. Think of the car analogy used in Section 1. The design is clearly of importance – a small diesel car might use half the fuel of a large petrol car. But, for a "fixed" design there are many other factors that affect m.p.g. A long motorway journey at modest speeds will usually be much more efficient than town driving. For refrigeration, it is important to realise that factors such as ambient temperature and plant load have a big impact on performance.

What Power is Included in COP?

A potential cause for confusion is what power inputs to include in the COP calculation. Compressor power is obviously included, but what about the main auxiliary loads such as pumps and fans used to run the evaporators and condensers? Many people refer to the "compressor COP" which only includes the compressor power. If we include evaporator and condenser auxiliaries then the COP is referred to as the "system COP" or "COSP". In all cases the COSP of an industrial refrigeration plant will be lower than the compressor COP.

The impact of auxiliaries is especially important at part load, because the pump and fan power is often fixed at the "full load" value. This makes a lot of refrigeration plants very inefficient at low loads.

Setting up a Monitoring Programme

The direct assessment of overall plant performance is done using long term measurements of energy use; hence it is best done on a regular basis, with the key measurements being made at regular intervals. For most food and drink sites a weekly monitoring interval is appropriate. For very large systems it may be worthwhile doing daily measurements.

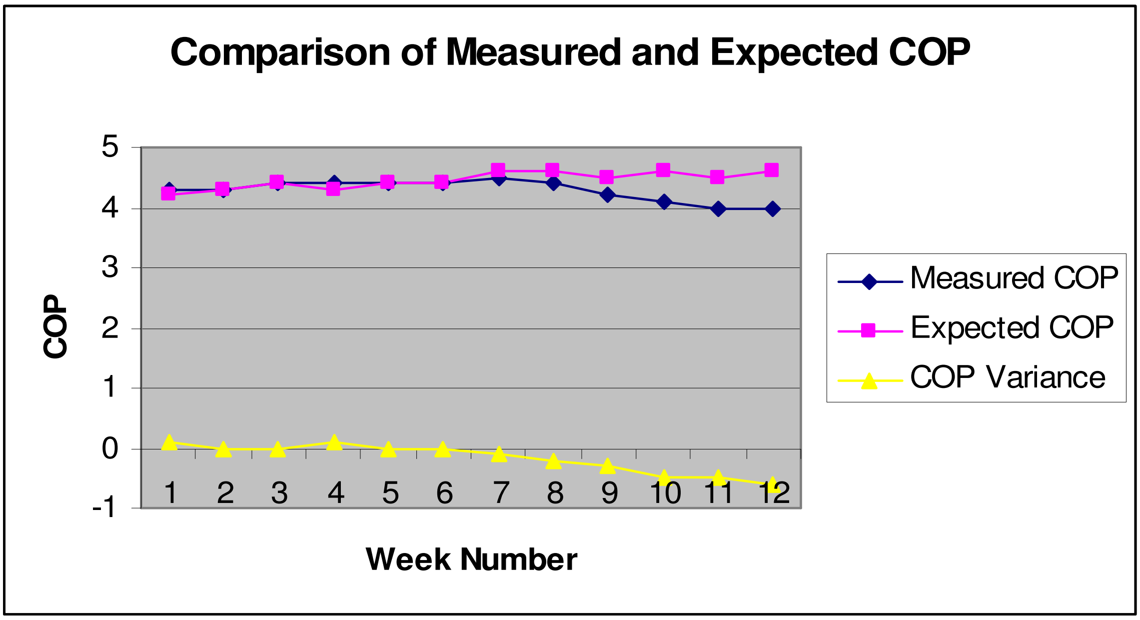
The overall plant COP is calculated using the cooling load data and power input data collected as described in Section 3. The COP calculation is very simple to do:

COP = Cooling carried out during period (kWh) / Electricity used during period (kWh)

The value of COP should then be compared with an expected value based on the plant design, the average plant load and the ambient temperature, using data like that illustrated in Table 8.

The COP data (including both measured and expected values) can then be plotted on a week by week basis to try and spot adverse changes in performance. Figure 5 shows an example plot of actual and expected COP. In Week 7 the actual COP was higher than in Week 6 (which at first inspection seems encouraging!) but it was lower than expected. Week 7 shows the start of a continuing adverse COP variance – a plant fault has started and needs investigation. If the measured COP is lower than the expected value then you should carry out fault assessments (using Strategy 1) to try and identify the cause of the inefficiency.

Figure 5



Appendix 1: GLOSSARY

Auxiliary energy	Energy used by devices in a refrigeration system other than the main compressors – usually this refers to pumps and fans.
Coefficient of Performance (COP)	A way of expressing the efficiency of a refrigeration plant. Defined as Cooling carried out divided by Energy input.
Compressor suction and discharge pressures	The pressure at the inlet and outlet of a refrigeration compressors.
Cooling duty	The total cooling load on a system.
Cooling load	The total amount of cooling carried out by a refrigeration plant – usually made up of several individual heat loads.
Dry bulb temperature	The "normal" temperature of ambient air that does not depend on the relative humidity of the air. (See wet bulb temperature)
Heat rejection temperatures	The temperature that a refrigeration plant rejects heat from the condenser, usually into ambient air or cooling water.
HP liquid receiver	A High Pressure Receiver, a vessel located beneath a condenser, used a reservoir of liquid refrigerant.
L%	Load setting of compressor in percent of full load.
Losses test	A way of measuring boiler efficiency based on measuring the heat lost in the flue gases.
P_d	Discharge pressure at compressor exit, which is approximately equal to the condensing pressure.
P_s	Suction pressure at compressor inlet, which is approximately equal to the evaporating pressure.
Pow_A	Power being absorbed by auxiliaries.
Pow_C	Power being absorbed by compressor.
Reciprocating compressor	A type of refrigeration compressor using a piston to compress vapour trapped in a cylinder..
Screw compressor	A type of refrigeration compressor using a rotating screw to trap a volume of vapour and compress it.
Suction strainer	A strainer at the inlet of a compressor designed to prevent damage caused by small objects entering the compressor.
T_{amb}	Ambient temperature
TC_{in}	Temperature of cooling medium (e.g. ambient air or cooling water) as it enters a condenser.
TC_{out}	Temperature of cooling medium (e.g. ambient air or

	cooling water) as it leaves a condenser.
TE_{in}	Temperature of fluid being cooled as it enters an evaporator.
TE_{out}	Temperature of fluid being cooled as it leaves an evaporator.
Temperature Lift	The temperature difference between the “cold end” of a refrigeration plant (the evaporator) and the “hot end” (the condenser).
TEV	Thermostatic expansion valve, a commonly used expansion valve used on small and medium sized refrigeration systems.
T_{liq}	Refrigerant liquid temperature at condenser exit.
T_s and T_d	Refrigerant vapour temperatures at compressor suction and discharge.
Wet bulb temperature	The temperature of ambient air that accounts for the relative humidity of the air. It can be measured by wetting the bulb of a mercury in glass thermometer – in periods of low humidity the evaporation that takes place cools the thermometer bulb. (See dry bulb temperature)

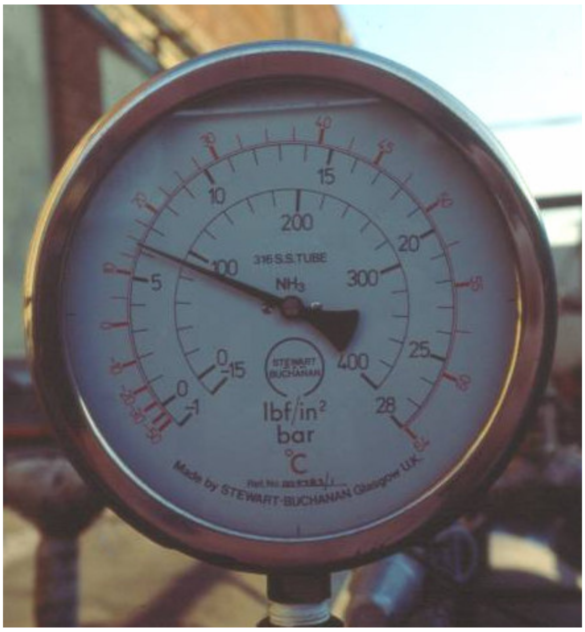
Appendix 2: SOURCES OF FURTHER INFORMATION

Food and Drink Federation	Trade association for food and drink manufacturers.	www.fdf.org.uk
Institute of Refrigeration	Professional body for refrigeration and air conditioning engineers.	www.ior.org.uk
British Beer and Pub Association	Trade association for brewing and pub sector.	www.beerandpub.com
Dairy UK	Trade association for dairy sector	www.dairyuk.org
Cold Storage and Distribution Federation	Trade association for the temperature controlled supply chain.	www.csdf.org.uk
British Refrigeration Association	Trade organisation for companies in the refrigeration and air conditioning industry.	www.feta.co.uk
Carbon Trust	Information and support regarding climate change issues.	www.carbontrust.co.uk
Guide 1	Appointing and managing refrigeration contractors.	www.ior.org.uk
Guide 2	Procurement of new plant.	www.ior.org.uk
Guide 3	Checklist for operational improvements.	www.ior.org.uk
Guide 4	HCFC phase out and F gas regulations.	www.ior.org.uk
Guide 5	Reducing heat loads.	www.ior.org.uk
Guide 6	Avoiding high head pressures.	www.ior.org.uk
Guide 7	Improving part load performance.	www.ior.org.uk
Guide 8	Reducing auxiliary fan and pump power.	www.ior.org.uk
EN378	Refrigerating systems and heat pumps. Safety and environmental requirements.	www.bsi-global.com
Refrigeration and Air Conditioning	Comprehensive text book covering all aspect of refrigeration and air conditioning.	ISBN 0-13-323775-3
GPG 278	Purchasing efficient refrigeration – the value for money option.	www.carbontrust.co.uk www.ior.org.uk
GPG 279	Running refrigeration plant efficiently – a cost saving guide for owners.	www.carbontrust.co.uk www.ior.org.uk
GPG 280	Energy efficient refrigeration technology – the fundamentals.	www.carbontrust.co.uk www.ior.org.uk
GPG 347	Installing and commissioning of refrigeration systems.	www.carbontrust.co.uk www.ior.org.uk
GPG 364	Service and maintenance technicians guide.	www.carbontrust.co.uk www.ior.org.uk
RAC	Monthly subscription trade journal and year book.	www.emap.com

Appendix 3: MEASURING ACTUAL TEMPERATURE AND SATURATION TEMPERATURE

One aspect of refrigeration plant performance measurement that often causes confusion for non-experts is the concept of saturation temperature, which is not usually the same as the actual temperature at the same location!

Figure A1: Refrigerant Pressure Gauge



Where a refrigerant is actually boiling or condensing – which means liquid and vapour refrigerant are both present in the same part of a pipe or vessel – the refrigerant is defined as "saturated". For most refrigerants, the saturation temperatures and pressures follow a defined curve; this means if you measure the pressure during evaporation you know what the evaporating temperature must be.

Figure A1 is a photograph of a pressure gauge used on a refrigeration plant. Notice how it has 3 scales – 2 in black showing pressure and one in red showing temperature. But the instrument is only a pressure gauge – why does it have a temperature scale? The scale is the saturation temperature for ammonia at each pressure level – just above the centre of the gauge it states: NH_3 which confirms this gauge is for ammonia refrigerant.

At the condition shown the gauge is showing a pressure of 6.2 bar and an equivalent saturation temperature of about 14.5°C. This gauge was mounted on a condenser, and it shows that the ammonia was condensing at 14.5°C – presumably on a cool day.

If we measure pressure when there is only vapour present e.g. at the compressor suction or discharge, then the vapour temperature <u>is not</u> equal to the value shown on the gauge as the vapour is superheated. Referring to Figure 2, the values of temperature, T_s and T_d are actual temperatures that need to be measured with a thermometer. The compressor suction gas temperature, T_s, is usually just a few degrees higher than the saturation temperature that would show on the pressure gauge. However, the compressor discharge gas is highly superheated – hence T_d will be much higher than the saturation temperature at the compressor exit (this saturation temperature is usually referred to as the condensing temperature).

The values of saturation temperature are very useful when we try to assess the performance of a condenser or evaporator. For example, an air-cooled condenser would normally have a saturated condensing temperature that is about 10 to 15 °C above the ambient air temperature. So if the ambient temperature is 20 °C, the condensing temperature might be between 30 and 35 °C (the exact value to be expected depends on the size of the condenser). This is much easier to remember than an expected condensing pressure, because that depends on which refrigerant you are using. If the plant used ammonia the pressure at 30 °C condensing temperature would be 10.7 bar, as shown on the gauge in figure 4. However, if the refrigeration was R134a the pressure would only be about 6.7 bar. That is why refrigeration engineers tend to refer to a condensing (or evaporating) temperature, when they are actually measuring it with a pressure gauge!

Appendix 4: MEASURING COOLING LOADS

Introduction

Using Strategy 2, Direct Measurement of Plant Performance, seems a simple and effective approach. You measure the cooling carried out and the amount of power used and then calculate the COP.

Measuring the power consumption is reasonably easy – although it does require kWh meters on the main electrical loads, especially the compressors.

Unfortunately measuring the cooling load is not always so easy to do – and sometimes is virtually impossible! For example if your main cooling load is a cold store, how do you measure the amount of cooling? You would need to measure the air flow and temperature difference across each cooling coil. This is impossible to do accurately and very expensive – that is why no industrial cold store users try to do it!

In some situations measuring the cooling load is easier. For example, if the cooling load is a flow of process liquid through a heat exchanger (e.g. milk through a pasteuriser) then you know the overall throughput (from production data) and the temperature difference can be measured fairly easily and accurately – so a cooling load can be assessed.

Options to Measure or Estimate the Cooling Load

There are 3 basic ways of estimating the cooling load:

1. **Direct measurement**. In some cases you can measure a flow rate and a temperature difference to directly calculate the heat load. This is most likely if a liquid is being cooled (e.g. chilled water or glycol in a secondary refrigeration circuit). This requires use of a "heat meter", which is a device that combines flow measurement with temperature difference measurement to calculate heat flow. Unfortunately they are fairly expensive if an accurate measurement is to be obtained.

2. **Estimate based on production throughput**. If the load is dominated by process cooling then it is quite easy to relate the load to the tonnes of production. For example if you are freezing peas you can model the heat load in three steps: (a) sensible cooling of the pea from the inlet temperature to freezing point, (b) latent heat of freezing the pea and (c) sensible cooling of the pea from the freezing point to the storage temperature. This can be calculated once using reference books to obtain the specific and latent heat values – for example you can calculate the kWh per tonne of peas being frozen. Then you can use production data for tonnes of peas processed to estimate a cooling load. This can be a very reliable method if done with care. It applies to process cooling in most food and drink factories.

3. **Estimate based on load modelling**. Some heat loads are not related to product throughput. In particular this applies to cold stores and chill stores. In these circumstances you need to model the cooling load by listing the individual elements of the load and modelling these in relation to relevant parameters such as ambient temperature. This is probably the least reliable method, but the only option available for plants with large amounts of "non-process" load.

Example of Load Modelling

To estimate the cooling load through load modelling you need to identify each of the individual elements that make up the overall cooling load. Then you must identify what each element is dependant on in order to calculate the total cooling load in a given period of time.

This methodology is best suited to a simple spreadsheet model. To set up load models of this type you may require expert help – but once the load model is established you can use it for on-going monitoring for many years.

A common example is the cooling load for a cold store. As discussed above this is very difficult to measure accurately – it is more practical to try and model the load. The main load elements for a typical cold store are shown in Table A1. The table also shows the dependant variables that must be taken into account during the load modelling.

Table A1: Modelling a Cold Store Load

Load Element	Dependant on TD*	Other Dependencies
1. Wall and roof heat ingress.	Yes	Insulation type, condition and thickness. Wall and roof areas.
2. Floor heat ingress.	Yes	Floor insulation type and thickness. Heater mat type and temperature. Floor area.
3. Doorway heat ingress.	Yes	Door size. Door usage patterns (number of openings, average time open) Measure door openings.
4. Evaporator fans.	No	Fan input power. Hours of fan operation.
5. Store Lights.	No	Light input power. Hours of light operation.
6. Product load.	Yes	If any product enters the store above the store temperature then it creates a cooling load within the store, dependant on the specific heat and TD.
7. Fork lift trucks.	No	Power consumption of FLT. Hours spent inside cold store.
8. Evaporator defrost.	No	Heat input during defrost. Hours of defrosting.

* Note, TD is the temperature difference between the inside of the store and the relevant "outside" temperature. Different outside temperatures can apply e.g. for a wall it may be average ambient but for a product load it is the average temperature of product entering the store.

Some of these load elements are fairly easy to calculate. For example:

- Element 1, wall and roof insulation load. The "U-value" of the insulation (a function of insulation type and thickness) is a constant that can be identified from manufacturer's data. The area of each wall element is fixed and can be easily measured. The temperature difference across the element will usually vary with ambient weather conditions. Once these parameters have been established the instantaneous heat load is calculated with the simple equation $Q = U * A * TD$ where Q is the heat load (kW), U is the U-value (kW/m^2K), A is the wall area (m^2) and TD is the temperature difference across the wall element (K). By using the average TD over a period of time the kWh of heat load over that period can be calculated.

- Element 4, evaporator fans. You need to identify the fan motor power and the hours of operation. Then the kWh heat load is simply the fan power (kW) multiplied by the hours of use.

Some of the load elements such as door openings are more tricky to calculate and require access to detailed refrigeration literature.

Appendix 5: EXPECTED VALUES

In Section 5 we describe the indirect assessment of plant faults by measuring a snapshot of plant performance and comparing measured values to expected values. This is not a simple process and you may need expert help to set up a table of expected values. However, once this has been done you can use the values during the life of the refrigeration plant. In this section we shall give a short example of the procedures that might be involved.

Expected Value of Condensing Temperature

The condensing temperature is a function of the heat transfer performance of the condenser. This is governed by the simple equation:

$Q = U * A * TD$

Where Q is the heat rejection load (kW), U is the U-value of the heat exchanger (kW/m^{2}K), A is the surface area of the heat exchanger and TD is the log mean temperature difference across the heat exchanger (between the cooling medium and the refrigerant).

U * A is a fixed parameter for a given condenser installation and it is easy to calculate from design data. The manufacturer of your plant will have stated a set of operating conditions for the whole plant, referred to as the design point. For a plant with a water cooled condenser, an example of design point data might be:

Cooling duty	100 kW
Compressor power	30 kW
Evaporating temperature	-10 ^oC
Condensing temperature T_c	32 ^oC
Cooling water on to condenser TC_{in}	24 ^oC
Cooling water off condenser TC_{out}	27 ^oC

From this data we can first calculate the TD at the design point. Although strictly speaking we should use the log mean temperature difference, within the accuracy required it is sufficient to use the arithmetic mean temperature difference, which is easier to calculate! This is defined as the average of (T_c-TC_{out}) and (T_c-TC_{in}), hence:

$TD = ((32-27) + (32-24)) /2 = 6.5\ ^oC$

Then we can calculate U * A at the design point, using:

$U * A = Q / TD = 130 / 6.5 = 20\ kW/^oC$ (note Q = 130 kW which is cooling duty + power)

Having established U * A it is then possible to calculate an expected T_c for any values of cooling duty, compressor power and cooling water on temperature. For example on a cool day, if the cooling water on temperature is 15 ^oC and the cooling duty is 95 kW and the compressor power is 22 kW, then

$Q = 95 + 22 = 117\ kW$

$TC_{out} = TC_{in} + (27-24)*117/130 = 17.7\ ^oC$

(Note: as the heat rejection is reduced to 117 kW, the temperature rise across the cooling water has reduced from 3 ^oC to 2.7 ^oC).

Then we can calculate T_c by rearranging the equation Q = U * A * TD as follows:

$T_c = Q/(U*A) + (TC_{out} + TC_{in})/2 = 22.2\ ^oC$

Hence, the expected value of T_c for operation at the conditions stated is 22.2 ^oC. This can be compared to the measured value, and if the measured value is much higher than this Table 3 can be used to identify a condenser fault.