THE INSTITUTE OF REFRIGERATION

Comparison of evaporative and air cooled condensers in industrial applications

by

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Evaporative condensers are commonly installed on large industrial refrigeration plant – this paper will consider the range of condenser options and how they are used. It will focus on the economic and efficiency justification for selecting different types of condensers based on a comparison of the total cost of ownership. This will include an evaluation of plant size and type (air cooled or evaporative) including design considerations such as relative first cost, energy cost, water supply, treatment and disposal costs, maintenance requirements and heat exchanger life expectancy. Weather data and part load operation characteristics will also be included. The impact of the condenser selection on refrigeration plant costs, both direct and indirect will be fully considered.

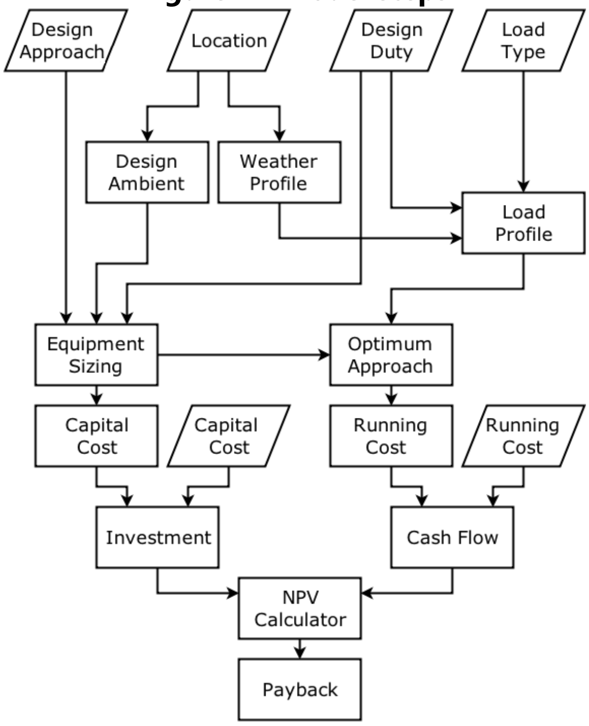
Introduction

Purpose
This paper compares ammonia refrigeration systems using flat bed air cooled condensers with high efficiency variable speed EC (Electronically Commutated) fans with epoxy coated fins against combined coil/fill evaporative condensers utilising axial variable speed fans. The objective was to determine the effect of condenser choice on the economic performance of the system.

Overview of Methodology
Many factors influence the life cycle cost of refrigeration plant for example: location, load type, equipment sizing and fan control. A computer model was developed so that multiple factors and their interaction could be evaluated.

The model assumed that the equipment can be perfectly sized to design requirements. This allows for a fair comparison by avoiding undersized or oversized equipment. Figure 1 shows the flow of data in the model. The model was run for each type of refrigeration system so that the net present value (NPV) calculator could evaluate the difference in capital costs and running costs.

Figure 1 - Model steps



Model Inputs

Design Approach
The design approach is the temperature difference between ambient temperature and the condensing temperature when at full load and maximum ambient temperature. Wet bulb was used as the ambient temperature for evaporative condensers and dry bulb was used for air cooled condensers. When comparing condenser types it is very important to select an approach for each condenser that allows for a fair comparison.

A comparison could be to look at an evaporative and an air cooled plant of the same capital cost. This would be unfair however toward evaporative condensers because they can be expected to have a lower capital cost than an air cooled condenser of the same economic efficiency. This scenario would have the evaporative condenser only receiving a small operating cost saving for a much larger capital cost. When compared against an air cooled system of the same capital cost it could be found that the air cooled offers savings for no difference in investment.

The size of each condenser (air cooled and evaporative) was chosen to deliver the same payback on the cost of improving the approach by 1K (Kelvin).

For a 2000kW cold store the design approach was calculated at 10.2K for an air cooled condenser and 7.3K for an evaporative condenser. In this scenario the end user would achieve a 3 year payback on either condenser by lowering the approach by 1 Kelvin.

The most economic size of condenser is affected by the expected load profile. When the condenser size is based on economic payback, a refrigeration plant with an air conditioning load that is only present in warmer ambient temperatures and in working hours will have a smaller condenser than a heavily loaded process plant that runs continuously. The calculation of economic approach typically shows the optimum air cooled condenser approach to be 1.4 times that of the optimum evaporative condenser approach at the design ambient temperatures.

Location, Design Ambient and Weather Data

Five years of weather data (2009 to 2014) was chosen for the following locations to be representative of the United Kingdom: Aberdeen, Belfast, Cardiff, Glasgow, London, Manchester and Newcastle [1].

Each weather station hourly record was rounded to the nearest whole degree. The records were then collated in a table showing the percentage of the year each location experienced each dry bulb temperature.

The average humidity in the dry bulb temperature range 0.5K either side of the whole degree temperature was then used to calculate the average wet bulb temperature for the dry bulb condition. These wet bulb temperatures were used with the dry bulb incidence data and associated duty to calculate the energy and water consumption for each case. The change of wet bulb against dry bulb is shown in Figure 2.

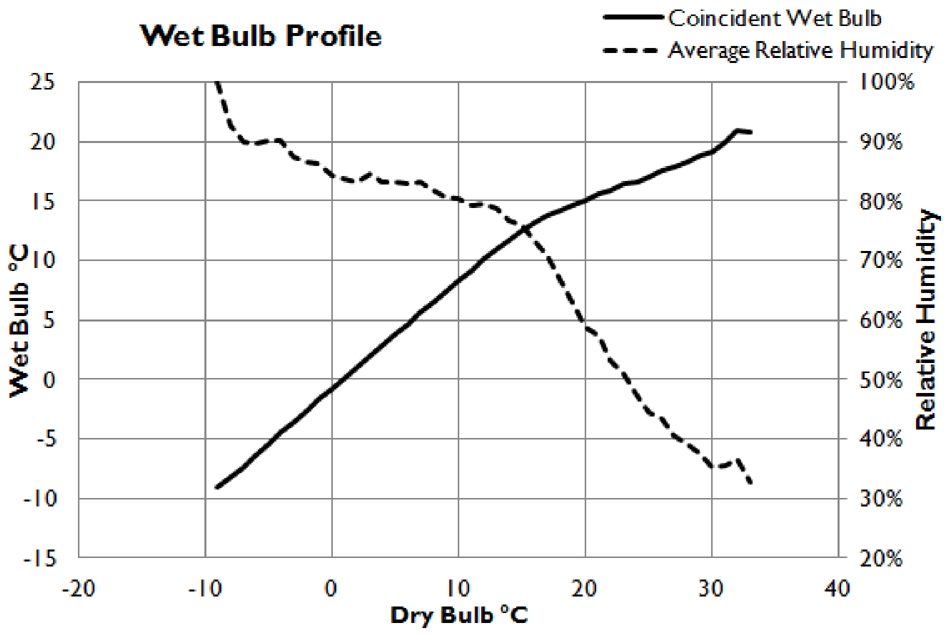


Figure 2 - Change of wet bulb and relative humidity

The design ambient was set at 32°C Dry, 22°C Wet bulb for London and Cardiff, 28°C Dry, 20°C Wet for Aberdeen and 30°C Dry, 21°C Wet for all other locations. This does not have an effect on condenser sizing as the economic design approach for air cooled and evaporative condensers were calculated before solving each condition.

Design Duty
Equipment sizing and costs were obtained from suppliers for a given range of sizes. Linear approximations or polynomial curves were fitted to the costs. For this reason the duty was limited from 250 kW to 2000 kW so that the approximation was not extrapolated outside the range of equipment sizes.

Load Type
5 load types were considered to approximate end user requirements shown in Table 1. The duty starts at the minimum and linearly scales to the maximum based on dry bulb (Min/Max Ambient).

The "minimum duty" dry bulb is the higher of the limit value in the table or the minimum ambient. The "maximum duty" dry bulb is the lower of the limit value in the table or the maximum ambient. An example of a Cold Store Load profile and the count of dry bulb hours for Newcastle is shown in Figure 3.

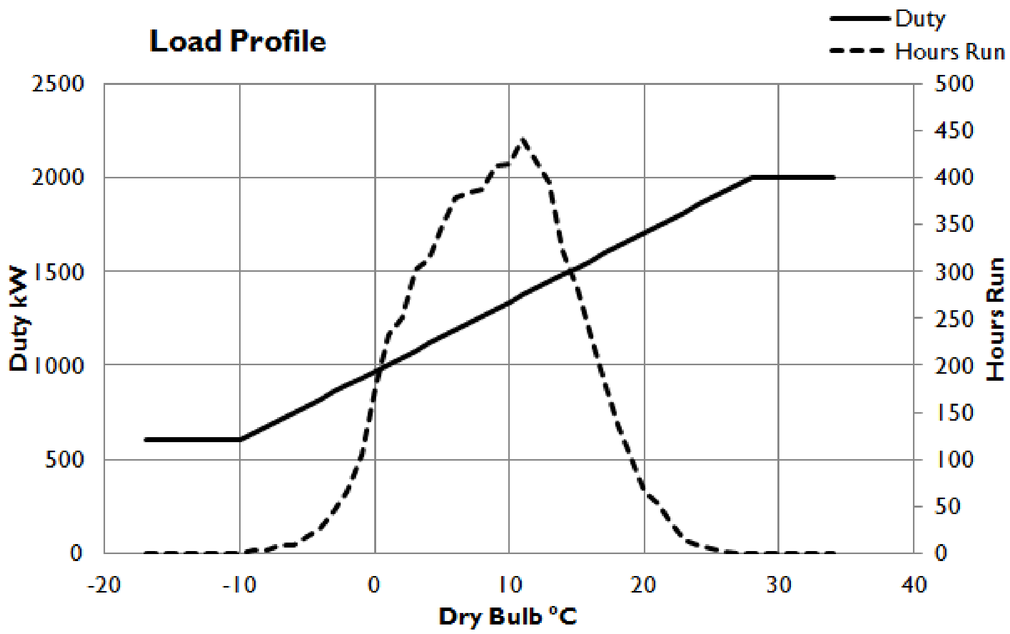


Figure 3 - Load profile and hour count

The evaporation temperature is assumed to be fixed throughout the year. This simplifies the model and allows for a reliable prediction of compressor performance. It is assumed that the difference between evaporating pressure for an evaporative condenser system against an air cooled condenser system is negligible.

Load Type	Utili-sation	Min Duty %	Max Duty %	Evap Temp °C	Min Duty Ambient °C	Max Duty Ambient °C
Office	30%	0%	100%	+3	+10	+35
Chill Store	84%	0%	100%	-7	-10	+35
Cold Store	75%	30%	100%	-31	-10	+35
Blast Freezer	64%	90%	100%	-38	-10	+35
Process	100%	80%	90%	-25	-10	+35

Table 1 - Limit values and coincident duty for each load type

Air cooled condensers have a higher capital cost but lower running costs (as discussed later). If the assumed utilisations were to increase this would increase the yearly savings and so decrease the payback time of an air cooled condenser.

Model Processes

Compressor Sizing
An economised screw compressor was modelled using data from a manufacturer's selection program. As the evaporation pressure was fixed by the load type it was convenient to populate a table against varying condensing pressures. This table contained coefficient of performance

(COP), oil cooling duty and economiser pressure. A curve was fitted through these values so that the rate of change of gradient was smoothed out.

It was observed that the compressor swept volume capacity requirement barely changed when changing from an evaporative to air cooled condenser. This was due to the fact that the compressor selection was based on economised operation so may favour air cooled condensers relative to other compressor arrangements.

The compressor motor was assumed to be operating with a variable speed drive. This allowed the compressor swept volume to exactly match the required condition with no performance decrease for part load operation.

Motor Sizing

The motor size was determined as the compressor shaft power at design condition plus 15%. The motor size was used to calculate the absorbed power at varying load. A 4% factor was added to the motor power usage to represent the variable speed drive and a table of part load efficiencies for the motor (assumed IE3) were used to increase power consumption accordingly.

Condenser Sizing

The condenser was sized from the design conditions: refrigeration duty + motor shaft power - oil cooling. It was assumed that the condenser only rejects heat from ammonia and that oil cooling was handled by separate equipment. This allows the full coil performance to be taken and eliminates an error that may bias the results.

The installed condenser fan power and pump power were linearly approximated using manufacturers selection data. Depending on the manufacturer, there can be large variations in fan and pump power (±30% in some cases). It was assumed that selections should be weighted toward the more efficient models for both condenser types. Pan heaters were excluded from the model as they only operate in low ambient and when the condenser is not operating.

Condenser Performance

The duty of the condenser at a different condensing pressure and approach can be calculated through the use of a factor multiplied by the design duty.

The air cooled condenser approach varied directly with the heat rejection regardless of the ambient temperature.

The evaporative condenser approach for a given duty widens as the wet bulb temperature falls. This is because colder air doesn't have as high a capacity to absorb moisture and is a significant penalty that is not obvious when making a simple comparison based on the design ambient conditions alone.

Three second order polynomials using approach were created to describe the coefficients and constant of a final second order polynomial using wet bulb. This allowed the model to relate evaporative condenser duty to any ambient and wet bulb.

Condenser Fan Performance
Fan speed and fan power are plotted against heat exchanger capacity in Figure 4. Air cooled and evaporative condensers both have the same profiles.

Best fit equations were found and limits added: Fan Power cannot be less than 0%, Fan Speed cannot be less than 20%.

EC fans on air cooled condensers can run at a lower speed than inverter driver induction motors on evaporative condensers. For a fair comparison the same minimum fan speed was applied to both.

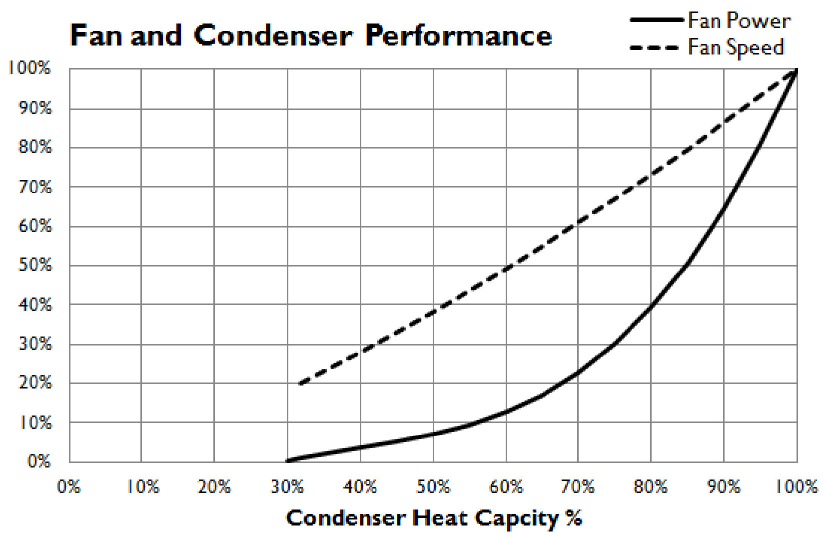


Figure 4 - Fan and condenser performance

Optimum Approach
For each dry bulb and corresponding duty, the operating conditions of the plant were calculated.

An algorithm was written to automate selection of the approach that yields the lowest sum absorbed power (compressor and condenser fan) whilst obeying the following rules:

- Minimum condensing pressure 15°C Sat
- Maximum condenser utilisation 100%
- Minimum condenser approach 2K

An example is given in Figure 5 (1000kW required duty, 2000kW design duty and 10.7K design approach). It can be seen that the minimum sum power for this condition is at 6.62K approach. Whilst the model assumes optimum fan control, the approach can change by ±1K and there was only a 1.3% increase in sum absorbed power.

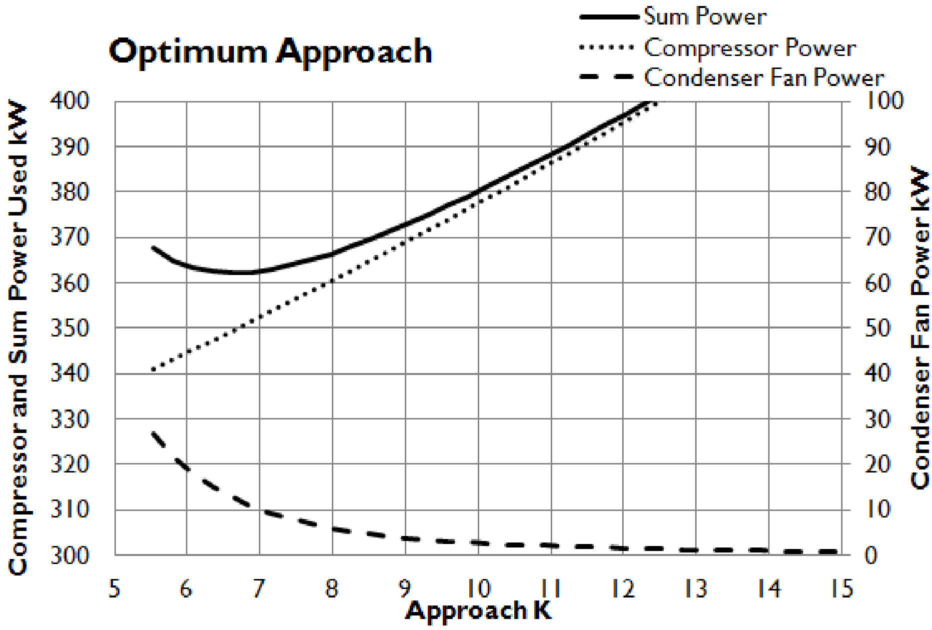


Figure 5 - Optimum approach

Financial Appraisal

Definition of Payback
Throughout this paper the word payback was used as the years taken for the investment in a larger condenser or a different condenser type to be recovered through saving operating costs. The operating costs have been devalued to account for the time value of money, see NPV section below.

Utility Costs
Electricity rates were taken at £0.089 / kWhr[2].

Water costs can vary greatly depending on location. An average of £1.15 / m^3 for supply and £1.28 / m^3 for disposal were used [3][4][5][6][7][8][9][10][11].

The volume that end users are charged for waste water is typically 90 to 95% of the supply volume [5][10]. Exemptions can be applied on a case by case basis however this requires metering of waste water. Rainwater capture may reduce supply and disposal costs. As a balance, it was assumed that the disposal cost is only applied to the bleed water.

Chemical costs were set at £0.39/m^3 based on sites where evaporative condensers are used and we were able to access copies of relevant billing information.

Capital and Running Costs
The compressor cost does not vary greatly as the swept volume difference is negligible.

The compressor motor and inverter cost is typically 25% more for an air cooled condenser compared to evaporative. This includes the increase in inverter cost but does not include the increase due to a larger electricity supply.

Equations describing the cost of the condensers and water treatment were approximated based on quotations.

No allowance was given for any difference in civil works that may be required e.g. steel work or baffling (air management) for air cooled condensers. No allowance was given for the installation of a water supply to the evaporative condenser.

No allowance was made for a difference in equipment life. Air cooled condenser life expectancy varies depending on operating environment and material specification whilst the life of an evaporative condensers is based on the water treatment and maintenance regime. Whilst it has been suggested air cooled condensers offer a longer life than evaporative [12] a check of replacement condenser work at the authors company and of the ASHRAE database was inconclusive [13].

The condenser cost was based on typical end user purchase prices for well known brands of each type of condenser.

No costs were included for water softening. Water usage was assumed at cycles of concentration 3.0 for chemical treatment and 6.0 for chemical free treatment. In all cases it was assumed that condenser performance was equal to design.

Equations for maintenance costs were created from quotations and previous bills.

Net Present Value Calculation (NPV)
A simple discounting calculation was used to determine the payback for investing in air cooled rather than evaporative. A real discount rate of 3.5% was used[14].

$$(NPV) = A_{CF_0} + \sum_{1}^{n} \frac{A_{CF_n}}{(1+i)^{(n-0.5)}}$$

Equation 1 - NPV Equation

Where:
- NPV = Net Present Value
- n = Years
- A_{CF} = Net annual cash flow

Payback time was determined from Equation 1 [15] by solving for n when NPV is zero.

The net present value calculation was adjusted by 0.5 years to account for the running costs being equally spread throughout the year.

A number of factors can complicate the payback calculation.

- Real electricity and water rates may increase
- Company savings on running costs could result in higher taxable profits
- Carbon tax may be payable. This may be increased over time to help the government meet emission targets. It may also decrease as the power sector reduces carbon emissions.
- Enhanced capital allowance schemes may reduce paybacks

For the purposes of this paper a simple net present value discounting method was used and assumptions over company tax arrangements were avoided.

Analysis of Results

Model Details

The majority of the hours in the year are between 9°C to 14°C dry bulb, see Figure 3 (this is true for all locations evaluated). Most load profiles show the plant to be part loaded at these conditions. Good fan speed control is therefore important to keep operating costs to a minimum.

The wet bulb closely follows the dry bulb at low temperatures however diverges at higher dry bulbs. This contributes to the decrease in difference between condensing pressure at lower dry bulbs.

An analysis presented earlier this year showed a lower condensing pressure for an air cooled condenser compared to an evaporative condenser at low ambient [12]. The ratio of approaches used was 0.83. However when changing to a ratio of 1.4 as discussed earlier, this effect was not observed.

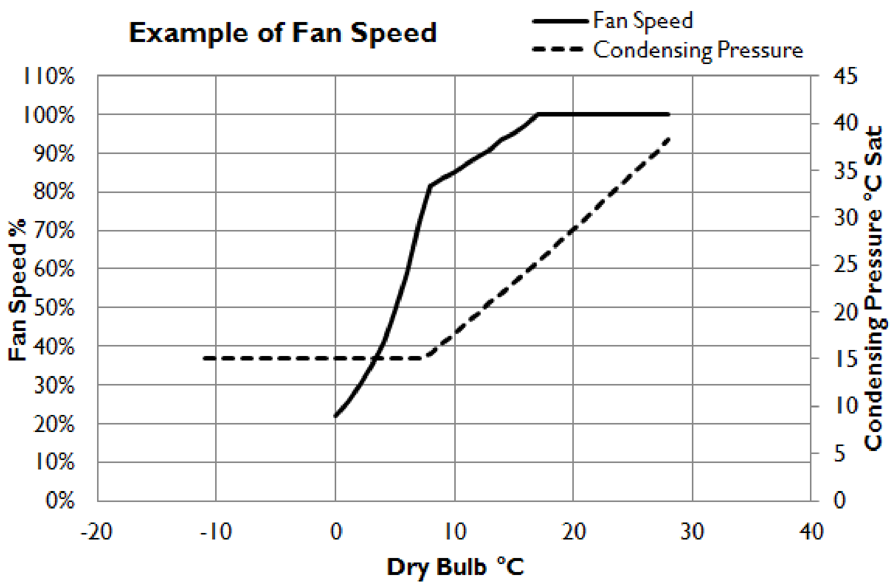


Figure 6 - Fan speed and condensing pressure

In the example of Figure 6, there are three phases of system operation across the range of dry bulbs.

In the first phase, the approach that yields the lowest combined fan and compressor power would have had the plant condensing at a pressure lower than 15°C. However the minimum condensing pressure limit forces a wider approach to be used. The only optimisation possible is in reducing fan speed to minimise power usage. Figure 6 shows that as the dry bulb rises from 0°C to 8°C the fan speed increases rapidly. The effect on fan power is even more substantial as can be seen in Figure 4.

The second phase occurs when the condensing temperature is higher than the 15°C minimum. A balance can be found between condenser fan power and compressor power by adjusting fan speed and condensing pressure in order to minimise the total power used. The model is sensitive to the rate of change of power and this is why best fit equations were carefully chosen instead of using linear interpolation of a table.

The final phase shows the fans running at 100% because the rate of change of compressor power with respect to condensing temperature is greater than the rate of change of fan power.

Variation of Plant Type and Location
Appendix A shows the various load profiles applied across the different locations. Using air cooled condensers a payback under two years can be achieved for all conditions. A slightly longer payback is shown for warmer locations which is expected due to better evaporative condenser performance in higher ambient temperatures. The load profile of the plant also has a large impact on the payback period for more expensive air cooled condensers. A significant extension is seen with plant in applications where the winter time load is low or nonexistent.

Variation of Plant Duty
Appendix B shows the various load profiles against plant duty. All configurations show air cooled condensers having either a lower install cost or a payback less than two years. Newcastle was chosen for its average typical UK weather profile.

Plant size and type has a significant effect on the economics of using an evaporative condenser. At 250kW and 500kW it is both cheaper to install and operate an air cooled condenser system. This is principally due to the purchase and installation cost for water treatment systems which are largely independent of the condenser size.

Breakdown of Costs Example
Appendix C shows air cooled condenser running costs relative to a chemically treated and a chemical free evaporative condenser. Both the chemical free and air cooled condenser offer savings however the air cooled condenser has a shorter payback and a higher saving per year.

The increasing payback time for larger duties is mainly due to the decreasing relative cost of maintenance. It is interesting to note the total annual cost of running the evaporative condenser pump is similar to the cost of the condenser fans. This is because the condenser pump must run at full speed to deliver adequate head to wet tubes and prevent scaling.

Chemical Free Condenser
Chemical free evaporative condensers were generally found to offer paybacks below three years on 1000kW duties and above compared to chemically treated evaporative condensers. All 250kW duty load profiles and the office profile for every duty were more expensive to operate as chemical free compared to chemical treatment.

Practical Considerations

The results have shown air cooled condensers to be an attractive alternative to evaporative condensers. Practical considerations need to be taken into account however.

Air cooled condensers are more susceptible to performance degradation because of air recirculation. Additional costs may need to be allowed for air management such as baffles.

Air cooled condensers require a larger footprint on site compared to evaporative which may be prohibitive for a variety of reasons.

Air cooled condensers installed at coastal locations are likely to be considerably more expensive. This is due to protective coatings and/or the fin material changing to copper.

More economic arrangements may be available such as V bank air cooled condensers proved air recirculation is managed.

Conclusions

An accurate cost comparison of both air cooled and evaporative condensers has been demonstrated.

Evaporative condensers are both more expensive to install and operate for all plant types up to approximately 500kW refrigeration duty.

For larger systems evaporative condensers offer a lower first cost and good refrigeration plant efficiency at peak ambient conditions. The cost of water supply, disposal and treatment however meant that the payback period for the more expensive air cooled condensers is less than 2 years for all plant types even when costs such as larger compressor motors and increased compressor power consumption were taken into account. This short payback when choosing an air cooled condenser instead of evaporative should be attractive to most end users.

Some end users might find it economical to add an air cooled condenser to an existing system with an evaporative condenser. However great care must be taken in the design of the liquid outlet from parallel condensers to ensure that problems are avoided.

Due to the investment being funded by the saving in operating cost, the load profile has a large impact on payback time. Plant that is more heavily loaded particularly in winter time will benefit with a shorter payback period.

Optimised fan control is important to help achieve maximum savings. Figure 5 shows that the fan control does not have to be perfect however.

Further work is needed to evaluate how hybrid condensers compare. Hybrid condensers cost around 13% more than air loaded condensers that have the same approach. This increase in capital cost would need to be justified by providing even lower annual operating costs than an air cooled condenser.

Hybrid condensers have additional yearly costs of increased maintenance, pump electricity, water supply and sewerage costs compared to air cooled condensers. They operate using water above 15°C dry bulb which is only 17% of the year in Newcastle (See Figure 3). This will require the hybrid condenser to provide lower running costs over a small period of the year to pay back its increased capital cost against air cooled.

Acknowledgements

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Appendix A - Payback in Years (Evaporative Chemical to Air Cooled - 2000kW)

		Cardiff	London	Manchester	Belfast	Newcastle	Glasgow	Aberdeen
Plant Type	Office	1.82	1.98	1.81	1.81	1.86	1.85	1.67
	Chill Store	0.70	0.83	0.71	0.66	0.68	0.68	0.60
	Cold Store	1.31	1.64	1.25	1.14	1.23	1.17	1.04
	Blast Freezer	1.51	1.85	1.51	1.39	1.44	1.43	1.34
	Process	0.82	0.93	0.79	0.75	0.76	0.76	0.70

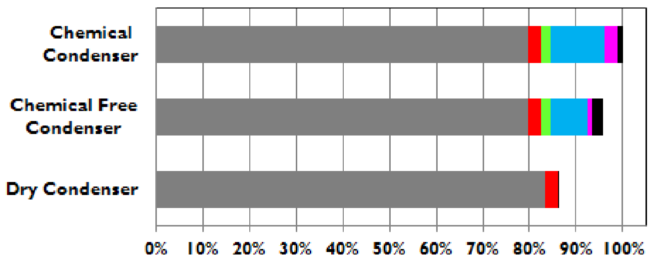
The locations are listed from left to right by latitude. When changing from evaporative condensers to air cooled condensers, London has the longest payback and Aberdeen has the shortest payback. The longer payback for the Office plant type shows evaporative condensers are better suited for a load type that is seasonal and during periods of high ambient compared to a Process plant type that is constant most of the year.

Appendix B - Payback in Years (Evaporative Chemical to Air Cooled - Newcastle)

		Plant Duty (kW)				
		250	500	1000	1500	2000
Plant Type	Office	—*	—*	—*	0.94	1.86
	Chill Store	—*	—*	0.21	0.52	0.68
	Cold Store	—*	0.03	0.79	1.07	1.23
	Blast Freezer	—*	0.45	1.07	1.31	1.44
	Process	—*	0.10	0.53	0.69	0.76

* The capital and yearly cost are lower for the air cooled option.

Appendix C - Yearly Cost Breakdown (Newcastle Cold Store 2000kW)



	Dry Condenser	Chemical Free Condenser	Chemical Condenser
Compressor Electricity	83.3%	79.8%	79.8%
Condenser Fan Electricity	2.8%	2.6%	2.6%
Condenser Pump	0.0%	2.2%	2.2%
Water	0.0%	7.9%	11.4%
Water Treatment	0.0%	0.8%	2.8%
Maintenance	0.2%	2.4%	1.1%
Total	86.4%	95.8%	100.0%

Meeting Venue Details
Room 110 Roberts Building,
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Roberts 110 is on the 1st floor. Turn right, walk through the double doors and walk up stairs to the first floor. Walk through the double doors on your left. Walk straight; walk through the double doors on your right. Walk straight through 2 double doors and Roberts 110 is the second right.

The room is accessible from Torrington Place (marked with the UCL Box on the map below) and a short walk from Euston Square, Warren Street or Goodge Street underground stations.