Energy Optimal Control of Cooling Towers. Elforsk PSO 349-033

Elforsk/PSO

Final report

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The expert in **WATER ENVIRONMENTS**

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Energy Optimal Control of Cooling Towers. Elforsk PSO 349-033

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Abstract

This report is a result of the project "Energy Optimal Control of Cooling Towers". The project was initiated by DHI and has involved a number of partners with cooling towers. The project has received funding from the Elforsk/PSO.

The aim of the project is to improve the control of cooling towers in order to save energy. In contrast to many other projects, this project attempts to optimize both cooling towers and the machinery connected to the cooling towers. The logic behind this is that the energy consumption of the machinery is often linked to the temperature of the cooling water from the cooling towers. In other words, if one only looks at the cooling tower and tries to optimize its energy consumption, this may lead to an increased consumption in the connected machinery. So, to obtain the really minimum energy consumption, a more holistic approach is needed, where the energy consumption of both cooling tower and its "clients" is taken into account.

The "clients" are typically compressor refrigeration systems, where the condenser on the refrigeration system is cooled with water from the cooling tower. A low water temperature leads to a low condensing temperature and thereby a low energy consumption on the compressors. To obtain a low water temperature, however, the cooling tower fans must run fast, leading to a high energy consumption on the fans. The goal is thus to find the optimum operating point for the fans, which will give the lowest total energy consumption.

The main content of the project is thus to design and test a control procedure that minimizes the overall energy consumption of cooling towers and "clients".

Conclusions

The conclusions drawn in the project so far can be summarized as following:

- In theory, considerable energy savings can be obtained by optimal control of cooling towers and their clients, especially in cases where the cooling towers and clients are operated at low loads and/or low ambient temperature.
- An optimization algorithm has been designed, based on a simple mathematical model of the cooling towers.
- Many industrial operations run full load 24 hours a day, in order to make the best possible use of the production machinery. In that case, there should still be a possibility for optimization, when the ambient air temperature is lower than the design temperature.
- In the case of compressor refrigeration systems, the optimization process will be limited by the fact that most compressor systems require a certain minimum condensing temperature to ensure a sufficient pressure to supply refrigerant to all consumers. In that case, the cooling tower fan speed will be controlled by the minimum condensing temperature, and no further optimization is possible.
- In the case of evaporative concentrators, it has not been possible to prove any optimization based on reducing the cooling water temperature. Data indicate that capacity actually decreases, as cooling water temperature is lowered. This is not logical in any way, and the reason for this behavior has not been identified.
- A number of project partners already have fan speed controllers attempting to minimize the total energy consumption. Such controllers are available for use on "simple" systems, meaning systems where all cooling towers run in parallel at the same fan speed, effectively acting as 1 big cooling tower.
- The analysis suggested that considerable energy savings were attainable for one site, to be confirmed by on site tests.

Resume

Denne rapport er et resultat af projektet "Energioptimeret styring af køletårne". Projektet blev igangsat af DHI og har involveret en række partnere med køletårne. Projektet har modtaget finansiering fra Elforsk/PSO.

Formålet med projektet er at forbedre kontrollen med køletårne med henblik på at spare energi. I modsætning til mange andre projekter forsøger dette at optimere både køletårne og det udstyr, der er forbundet med køletårnene. Logikken bag dette er, at udstyrets energiforbrug ofte er knyttet til temperaturen på kølevandet fra køletårnene. Hvis man med andre ord kun ser på køletårnene og forsøger at optimere disses energiforbrug, kan det ende med at føre til et øget forbrug i det tilknyttede udstyr. For at opnå det reelt set lavest mulige energiforbrug er der således behov for en mere holistisk tilgang, hvor energiforbruget i såvel køletårnene som disses "kunder" tages i betragtning.

"Kunderne" er typisk kompressorkølesystemer, hvor kondensatoren på kølesystemet afkøles med vand fra køletårnet. En lav vandtemperatur fører til en lav kondenserings-temperatur og dermed et lavt energiforbrug i kompressorerne. For at opnå en lav vandtemperatur må køletårnets ventilatorer dog køre hurtigt, hvilket fører til et højt energiforbrug på ventilatorerne. Målet er således at finde det optimale driftspunkt for ventilatorerne, hvilket vil give det laveste samlede energiforbrug.

Projektets hovedidé er således at designe og afprøve en kontrolprocedure, der minimerer det samlede energiforbrug for køletårne og "kunderne".

Konklusioner

De konklusioner, der indtil videre har kunnet uddrages af projektet, kan opsummeres som følger:

- I teorien kan der opnås betydelige energibesparelser ved optimal styring af køletårne og deres ”kunder”, især i tilfælde hvor køletårne og ”kunderne” kører ved lave belastninger og/eller lav omgivende rumtemperatur.

- Der er blevet designet en optimeringsalgoritme baseret på en simpel matematisk model af køletårnene.

- Mange industrielle operationer kører med fuld belastning 24 timer i døgnet for at udnytte produktionsmaskinerne bedst muligt. I så fald burde der stadig være mulighed for optimering, hvis den omgivende rumtemperatur er lavere end designtemperaturen.

- For kompressorkølesystemer vil optimeringsprocessen være begrænset af, at de fleste kompressorsystemer kræver en vis minimumskondenseringstemperatur for at sikre, at trykket er tilstrækkeligt til, at der kan leveres kølevæske til alle forbrugere. I dette tilfælde styres køletårnets viftehastighed af minimumskondenserings-temperaturen, og det vil derfor ikke være muligt at optimere yderligere.

- I tilfælde af fordampningskoncentrationsanlæg har det ikke været muligt at påvise nogen optimering ved at sænke kølevandstemperaturen. Data tyder på, at kapaciteten faktisk falder i takt med, at kølevandstemperaturen sænkes. Dette er ikke på nogen måde logisk, og det har ikke været muligt at identificere årsagen.

- En række projektpartnere benytter sig allerede af hastighedsregulatorer til ventilatorne i et forsøg på at minimere det samlede energiforbrug. Sådanne regulatorer er tilgængelige til brug på "simple" systemer, hvilket vil sige systemer, hvor alle køletårne kører parallelt med samme ventilatorhastighed og således reelt fungerer som ét stort køletårn.

- Analyserne viser et betydeligt potentiale for energibesparelser hos én af partnerne, som således har muligheden for en efterfølgende afprøvning i praksis.

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FIGURES

1 Introduction

The project seeks to optimize the energy consumption of cooling towers and the machines/processes being cooled by the cooling towers.

The design of cooling towers is usually based on worst-case conditions: warm and humid ambient conditions. When the ambient air is colder/drier than design conditions, or when the actual cooling need is lower than the design value, an overcapacity is available. This can be used to optimize the energy consumption by reducing the pump- and fan speeds in the cooling tower, or even better by lowering the water temperature in the cooling tower, to improve the function of the machine cooled by the tower. Compressor cooling systems get a better energy efficiency with lower cooling water temperatures, and since the electricity consumption of the compressor system is far bigger than that of the cooling tower, a change in the water temperature will have the largest effect. When cooling needs or ambient conditions change, finding the optimal operating point is not a simple matter. Often, the controllers use simple fixed temperature set points instead of using the opportunities to reduce water temperature. Likewise, compressor systems and cooling towers are often supplied by different subsuppliers, each with their own idea of how to control the individual units. Very few have a well-founded advice on controlling the combined units in an energy optimal way. The goal of the present project is accordingly to define and test energy optimal control algorithms for various combinations of cooling towers and their “clients”, for example for compressor-refrigeration systems, evaporators and thermal power plants.

2 Cooling towers and evaporative condensers

In this project, both cooling towers and evaporative condensers are being used, and the following is a brief explanation of the differences and similarities.

2.1 Cooling towers

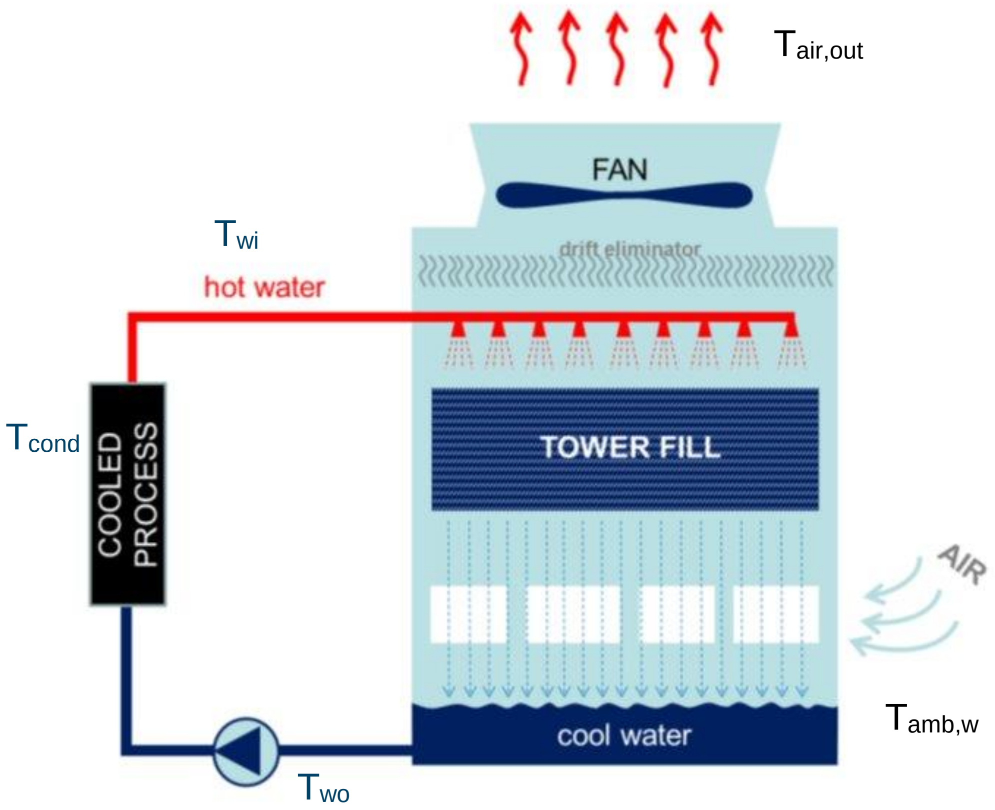

Figure 2-1 Typical cooling tower system

The sketch above shows a typical cooling tower system, where the cooling tower cools a condenser of a refrigeration system or another cooled process in a heat exchanger. The condensation temperature inside the refrigerant system is T_{cond}, and this temperature is obviously higher than T_{wo} and T_{wi}, the water temperatures out of and into the cooling tower. The air coming into the cooling tower is designated by its wet bulb temperature $T_{amb,w}$, since this is the most important temperature for defining the cooling tower performance. The air coming out of the cooling tower is assumed to be fully saturated with moisture at a temperature somewhat lower than that of the incoming water (T_{wi}).

The cooling capacity can be described as the airflow multiplied by the enthalpy difference between air coming in and air going out. Likewise, it can be described as the water flow multiplied by the thermal capacity of water and by the temperature difference $T_{wi} – T_{wo}$. The water circulated in the cooling tower is cooled by the incoming ambient air, partly by thermal conduction to the air, but mostly by evaporation of a fraction of the water (typically around 85% of the actual cooling capacity).

The power consumption of the fan will typically be proportional to the airflow raised to the power of 3. The consumption of the water pump will not be considered, since this is typically far lower than the fan power in the systems studied in this project.

2.2 Evaporative condensers

Evaporative condensers have some similarities to cooling towers. In an evaporative condenser, water is evaporated on the surface of pipes, in which the refrigerant is cooled and thereby condensed.

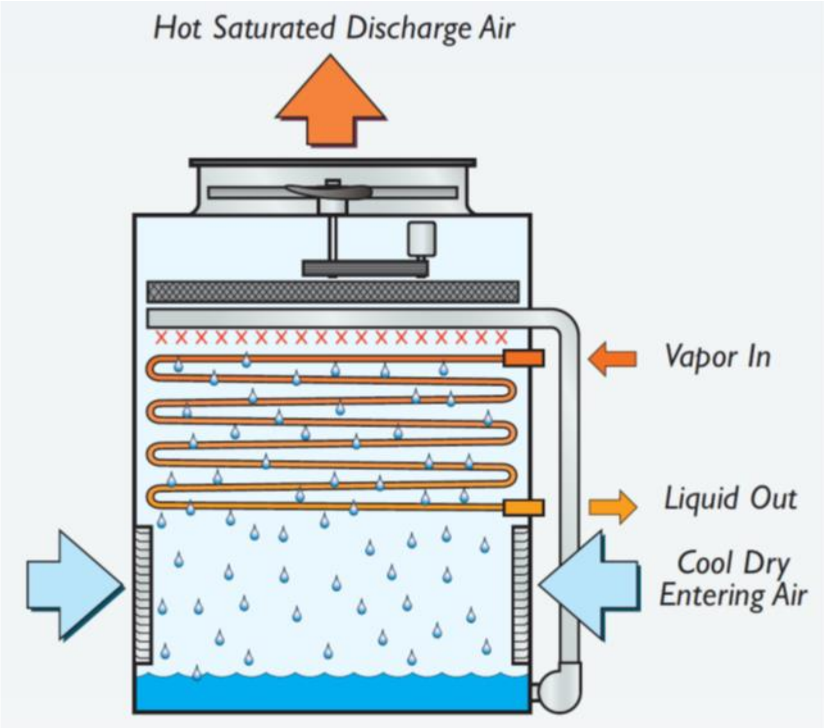

Figure 2-2 Evaporative condenser, where water is sprayed onto the surface of piping. Refrigerant vapor enters at top, and condensed liquid exits at bottom

Since the condensation of the refrigerant takes place directly in the unit, no additional water (or brine) circuit is needed. Condensation temperature can be considered practically constant in all the piping. This makes the performance calculations easier than for cooling towers, since you only need to relate the condensation temperature to the ambient wet bulb temperature (and the air flow, obviously). The water only serves as an intermediate medium, whose temperature will be controlled by the actual design and operating parameters, but is otherwise not relevant for the calculation, especially not for control issues.

Just like for cooling towers, the power consumption of the fan will typically be proportional to the airflow raised to the power of 3, and the consumption of the water pump will not be considered, since this is typically far lower than the fan power. Just like for cooling towers, most of the cooling takes place in the form of water evaporation, and only a small fraction of the cooling is due to thermal conduction between air and water.

3 Project partners

A number of partners have been involved in this project, of which most have cooling towers and evaporative condensers for cooling the condenser of industrial refrigeration systems. One partner has an evaporative concentrator cooled by a cooling tower

3.1 DAKA

DAKA processes animal byproducts from slaughterhouses. In the process, a quantity of stick water is generated, containing valuable proteins. The stick water is concentrated by evaporation, and the evaporator is cooled by water from a cooling tower.

3.2 Arla DP, Nr. Vium.

Arla DP (Dansk Protein) produces protein powders, based on milk. In the process, large quantities of ice water is needed, and Arla DP has several compressor refrigeration systems, which are cooled by water from cooling towers.

3.3 Arla Brabrand.

Arla Brabrand produces yoghurt and other semi-liquid milk products. In the process, large quantities of ice water is needed, and Arla Brabrand has several refrigeration compressor systems, which are cooled by water from 2 cooling towers.

3.4 Arla Rødkærsbro

Arla Rødkærsbro produces Mozzarella cheese. Refrigeration is needed for the production of ice water, storage cooling and a spiral freezer (0°C, -10°C and -20°C). All refrigeration units are coupled to a common condensation circuit, cooled by 6 evaporative condensers.

3.5 Odense University Hospital (OUH)

OUH uses cooling water for air conditioning and cooling of various hospital scanners. Typical cooling water temperature is 12°C. The consumption of cooling water is very variable, depending on the number of scanners in use at any time. In the winter time, cooling is done by a number of free-cooling units. In the summer time, compressor refrigeration systems are set into operation, cooled by a cooling tower.

3.6 Vestas Industrial Cooling

Vestas Industrial Cooling manufactures cooling towers and heat exchangers.

3.7 DONG Energy

DONG Energy is an energy (electricity, district heating) supplier and as such has an obligation to find energy savings at industrial customers.

4 Characteristics of evaporative condensers

The graph below shows brochure data for an Evapco cAT 17-812 evaporative condenser. The graph shows the heat rejection capacity as function of the difference between condensing temperature and ambient air wet bulb temperature. This temperature difference is the driving force behind the heat transfer. Data are all for full fan speed.

More correctly, the real driving force is the enthalpy difference between the saturated air at the water surface surrounding the condenser pipes and the enthalpy of the ambient air. These enthalpies are closely linked to the temperatures, but in a slightly non-linear fashion, which makes it possible to model this as a power function with a power slightly higher than 1.

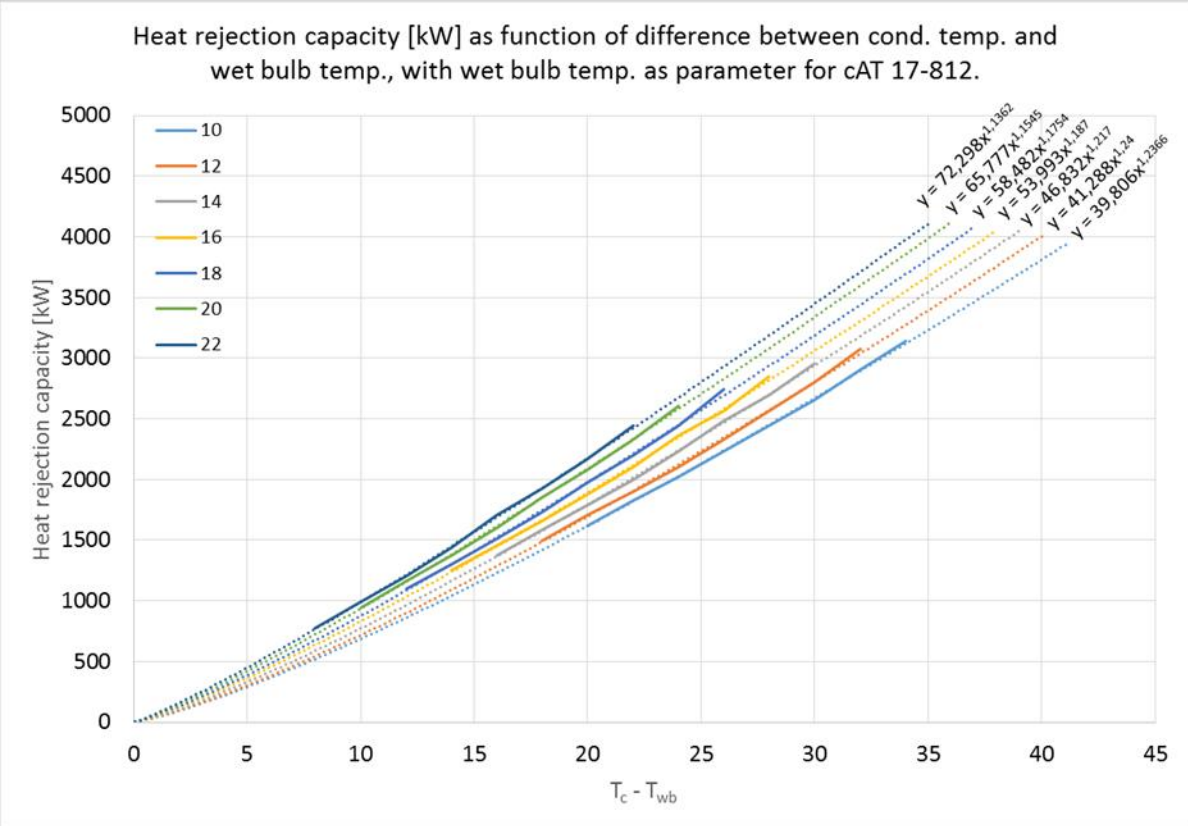

Figure 4.1 Characteristic of a typical evaporative condenser (Evapco cAT 17-812)

The condenser capacity can be described as proportional to $f(T_{wetbulb}) * (T_{cond} – T_{wetbulb})^D$, where D in this case is in the range 1,136 → 1,237.

The function $f(T_{wetbulb})$ can be illustrated by taking values for varying wet bulb temperatures, with a constant difference between condensing and wet bulb temperature. In other words, taking values along a **vertical** gridline in the diagram above.

The graph below shows the capacity at $T_{diff} = (T_{cond} – T_{wetbulb}) = 20$ K, as function of T_{wet}, as an example of the above mentioned $f(T_{wetbulb})$. This indicates that $f(T_{wetbulb})$ is close to being a linear function.

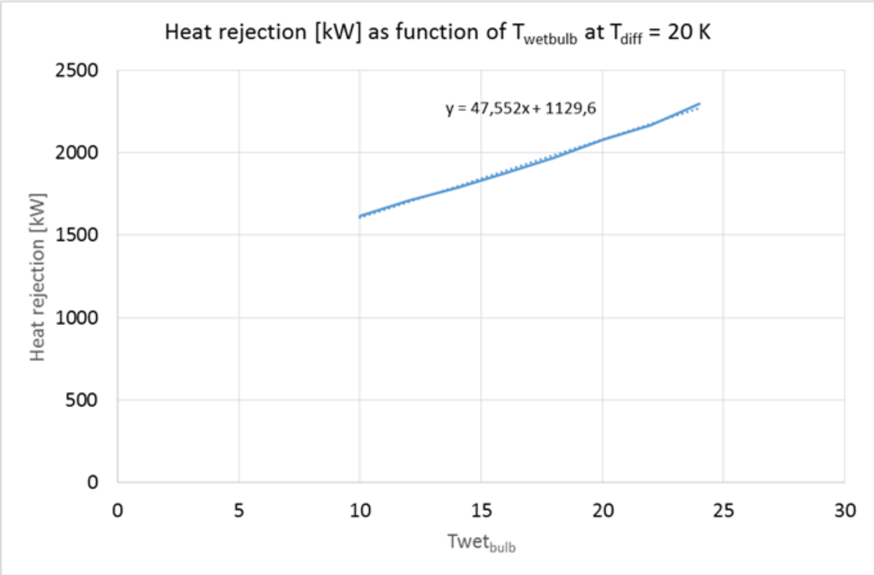

Figure 4.2 Heat rejection as function of wet bulb temperature for T_{diff} = 20 K

In this case, the capacity could be written as:

$$Cap = (A * T_{wetbulb} + B) * \left(\frac{T_{cond} - T_{wetbulb}}{C}\right)^{D},$$

where A = 47,552 and B = 1129,6, C is 20 and D as before in the range 1,136 → 1,237.

For a given wet bulb temperature and a given water inlet temperature , the capacity can be calculated at 100% fan speed, based on brochure data, corrected for the 2 temperatures.

If fan speed is variable, the capacity can in most cases be taken as proportional to $Fanspeed^{0,8}$. The reasoning behind this is, that the turbulent heat and mass transfer rate is known to be proportional to air $speed^{0,8}$. The equation will then be:

$$Cap = (A * T_{wetbulb} + B) * \left(\frac{T_{cond} - T_{wetbulb}}{C}\right)^{D} * Fanspeed^{E}$$

The equation contains 5 parameters, A, B, C, D & E but only A and B are really unknown. C is a temperature chosen for obtaining the dataset giving A and B. D represents the nonlinearity of moist air enthalpy as function of temperature and will only change slightly from case to case. E represents the variation of air and mass heat transfer rate with air speed, and in most cases, a value between 0,8 and 0,9 can be used.

5 Characteristics of cooling towers

Cooling towers have in the past been calculated in different ways. One of the more popular calculation procedures is the so-called Merkel-model. This mathematical model is derived from the physics of what goes on in the cooling tower, taking into account the evaporative heat and mass transfer, and is normally viewed as being a rather accurate model. The calculation is iterative in nature and requires values for the enthalpy of moist air at various points in the process. In addition, it can be shown that the iteration process can lead to more than 1 solution. The details of the Merkel model will be explained in a later paragraph. Using the Merkel model on a real life example, it can be shown that a cooling tower can be calculated/modelled in a manner similar to the evaporative condenser.

The diagrams below were all calculated using the Merkel model on a real life example of a cooling tower at Odense University Hospital. Calculations were all done at nominal air- and water flow with an EES-program “Cooling Tower NTU-Effectiveness.EES”, produced by DHI for this project.

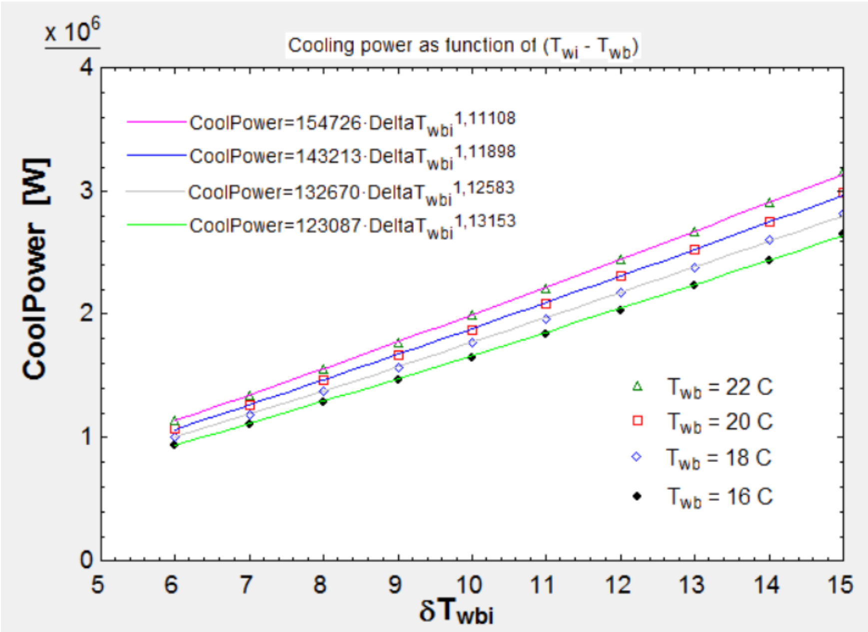

Figure 5-1 Heat rejection as function of the difference between water inlet and wet bulb temperature, with wet bulb temperature as parameter.

The diagram above shows, that the cooling power (or heat rejection) can be calculated as a weak power function of the “driving force”. In the diagram, heat rejection is taken as function of the difference between water inlet and wet bulb temperature. The real “driving force“ is the enthalpy difference between the 2 states, which is a slightly non-linear function of temperature, hence the power function.

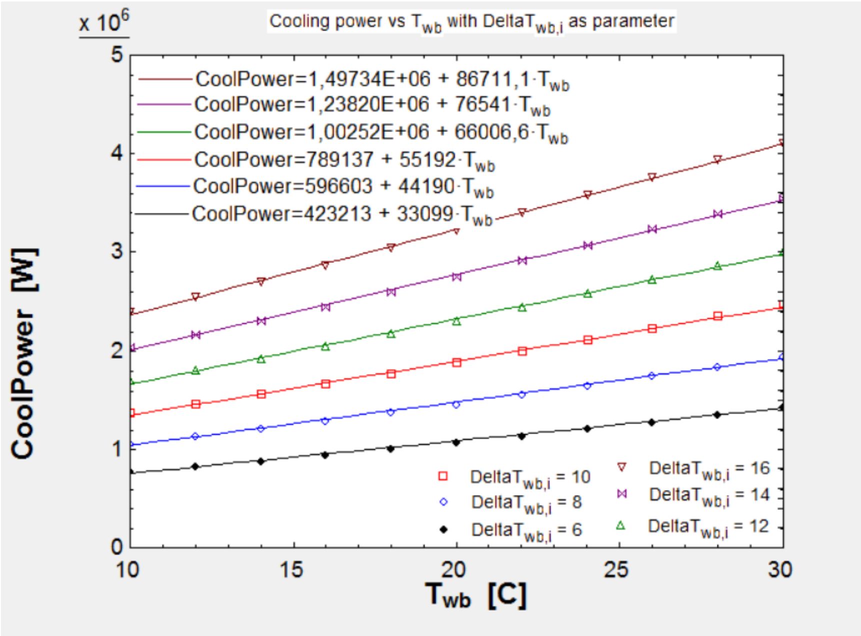

Figure 5-2 Heat rejection as function of wet bulb temperature, with the difference between water inlet and wet bulb temperature as parameter.

In the diagram above, cooling power (or heat rejection) is shown as function of wet bulb temperature, with the difference between water inlet and wet bulb temperature as parameter. Just like for evaporating condensers, this seems to be very close to a linear function. So, just like for evaporative condensers, with a fairly good exactness, one could say that the cooling power at full fan speed is proportional to $\Delta T_{wbi}{}^{D}$, with a small, linear correction factor for hot water temperature.

$$Cap_{Tower} = (A * T_{wetbulb} + B) * \left(\frac{T_{w,i} - T_{wetbulb}}{C}\right)^{D}$$

Taking, for example, ΔT_{wbi} to be 10 (red line on fig. 5.2), A = 55192; B = 789137; C = 10 and D can be set somewhere between 1,11 and 1,13 depending on how accurate one needs the model to be.

Just like for evaporative condensers, it would be reasonable to say that capacity is proportional to $FanSpeed^{E}$, where E would be in the range of 0,8 to 0,9.

So, all in all, the equation would be exactly the same as for evaporative condensers:

$$Cap_{Tower} = (A * T_{wetbulb} + B) * \left(\frac{T_{w,i} - T_{wetbulb}}{C}\right)^{D} * FanSpeed^{E}$$

Obviously, the parameters A, B, C, D and E would be unique for each condenser/tower.

For varying water flows, the situation gets a bit more complicated. Generally, when reducing the water flow in a cooling tower, the capacity also drops. This is partly because a reduced water flow may not spread evenly over the cooling tower fill material, thereby reducing the active heat and mass transfer area. And secondly, a reduced water flow would get cooled more quickly on the way through the cooling tower, thereby reducing the average temperature difference between the water and the air.

The latter can be handled mathematically, if desired, but the first problem cannot, since it will by nature be unique for each cooling tower.

If it is desired to run the cooling tower on reduced water flow, the capacity can be adjusted by a correction factor, dependent on water flow. The correction factor could be given by the tower supplier or be taken from real life measurements with full and reduced water flows.

5.1 The Merkel model.

F. Merkel developed a cooling tower model in 1925, which is still very popular.

In the model, some simplifications are made, to simplify calculations:

- Thermal capacity of air stream mixture is assumed to be the same as for dry air
- Water loss due to evaporation is neglected (1,8% per 10°K cooling of the water)
- Temperature difference across the water film is neglected
- Lewis-number = 1 (characteristic length for thermal diffusion / characteristic length for mass diffusion)

These simplifications make it possible to produce a useful and practical model without sacrificing too much on exactness.

The model starts out with a look at a small differential volume of a cooling tower:

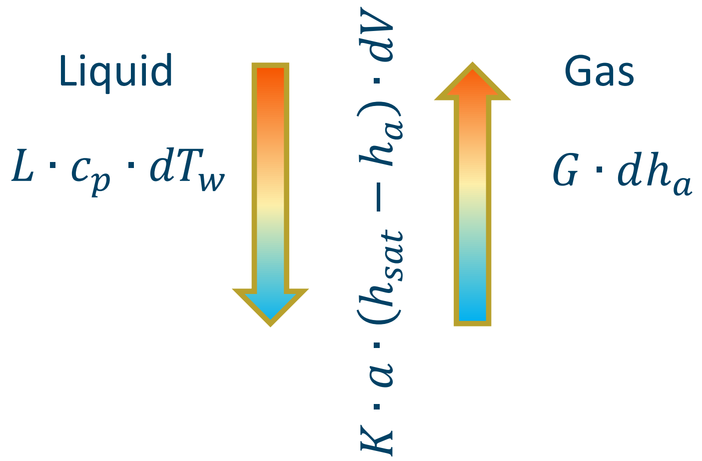

Figure 5-3 The Merkel cooling tower model

In the differential volume, water is falling down through the tower element, and gas flows upwards, in counter flow with the water.

Energy is removed from the water at a rate of L * c_p * dT_w, where L is the mass flow of water (**L**iquid), c_p is the thermal capacity of water and dT_w is the temperature difference of water coming in and out of the element.

The same energy is transferred to the air (**G**as) at a rate of G * dh_a, where G is the mass flow of air and dh_a is the enthalpy difference between air coming in and going out of the element.

And at the interface between water and air, the same energy is transferred in a rate that can be expressed as: K * a * (h_{sat} – h_a) * dV, where K is a (enthalpy) transfer coefficient, a is the surface area per volume unit, h_{sat} and h_a are the enthalpies of the saturated air film at the water surface and the enthalpy of the air. dV is the differential volume.

Integrating over the entire volume of the cooling tower, and rearranging the equations, the following equation can be obtained:

$$Me = \frac{K \cdot a \cdot V}{L} = c_p \cdot \int_{T_{wi}}^{T_{wo}} \frac{dT_w}{(h_{sat} - h_a)}$$

Here, Me is the so-called Merkel number. The Merkel number tells, for a given desired temperature of water in and out, and air inlet conditions, what is required from the cooling tower in terms of transfer coefficient, specific area and volume, all related to the water flow (the K*a*V/L of the tower), and is calculated as an integral (the right side of the above equation.

The integral cannot immediately be calculated, but can be approximated by a so-called Chebyschew polynomial. The equation then reduces to:

$$Me = c_p \cdot \frac{(T_{wi} - T_{wo})\left\{\frac{1}{dh1} + \frac{1}{dh2} + \frac{1}{dh3} + \frac{1}{dh4}\right\}}{4}$$

where:

$dh1=(h_w-h_a)$ calculated at a water temperature of T_{wo} + 0,1 (T_{wi}-T_{wo}),

$dh2=(h_w-h_a)$ calculated at a water temperature of T_{wo} + 0,4 (T_{wi}-T_{wo}),

$dh3=(h_w-h_a)$ calculated at a water temperature of T_{wo} + 0,6 (T_{wi}-T_{wo}),

$dh4=(h_w-h_a)$ calculated at a water temperature of T_{wo} + 0,9 (T_{wi}-T_{wo}),

Using this procedure, it is possible to calculate the Me-number (the tower requirement for the given process) as function of the water to air ratio. An example is shown below. Here, the water inlet and outlet temperatures have been set to 40°C and 30°C, and the calculation has been performed for 4 different ambient air wet bulb temperatures. And as can be seen, the higher the ambient temperature, the higher is the requirement on the tower performance, because the driving temperature difference falls.

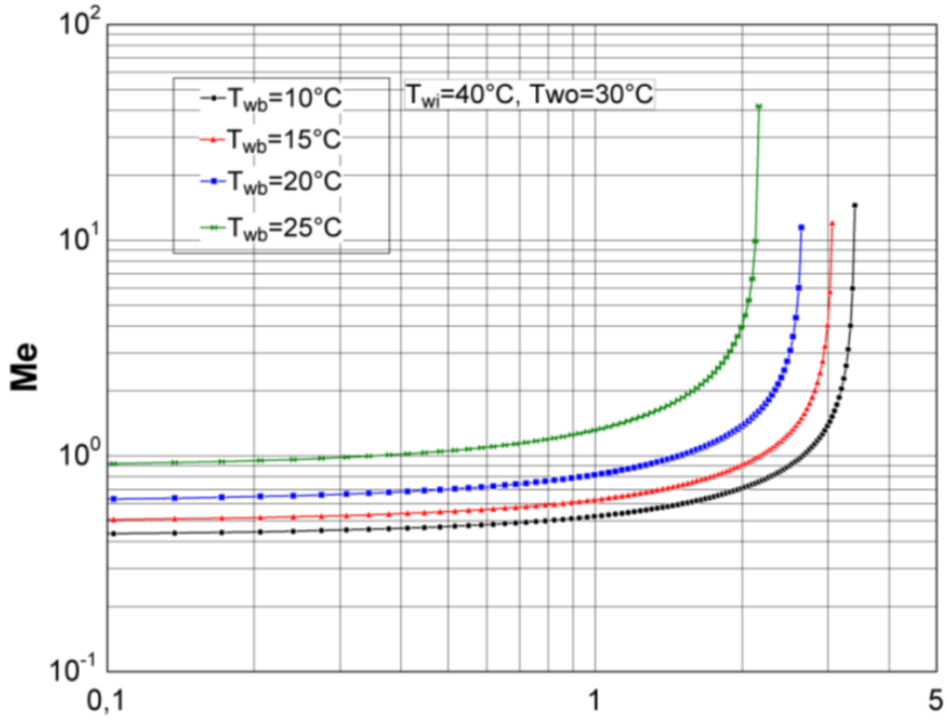

Figure 5-4 Water to air mass ratio [L/G - ratio]

It is also clear from the figure that for each set of conditions, there is a maximum possible water to air ratio (or a minimum air to water ratio). This can be understood as follows:

If you increase the water flow for a given air flow, and still request the same T_{wi} and T_{wo}, then more energy must be transferred to the air. The air, however, cannot be heated above T_{wi}. So, when approaching T_{wi}, the temperature difference between water and air will be very small in the top of the tower, setting enormous requirements on the tower capability in terms of area, volume and/or transfer coefficient.

Having calculated the required tower performance, it is possible to add a plot of the performance of an actual tower. An example is shown below.

Typical cooling tower characteristics are straight lines in the double-logarithmic diagram, following an equation of the type:

$Me = C * (L/G)^M$, where C is typically in the range 1 to 3 and M is between -0,6 and -0,8.

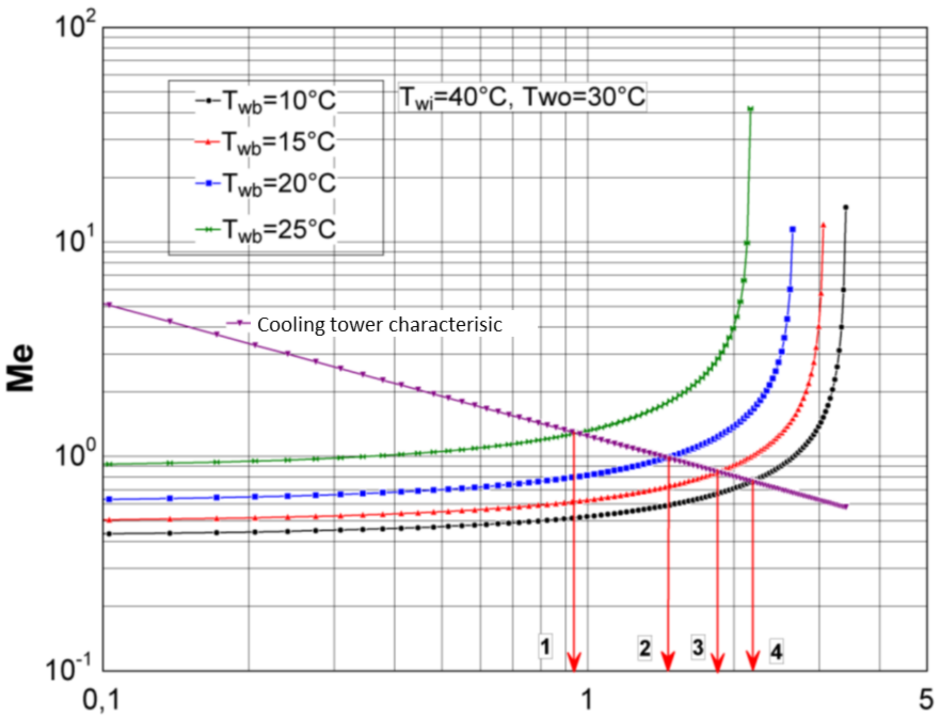

Figure 5-5 Water to air mass ratio [L/G-ratio]

So, for a given operating condition, the actual operating point is found where the cooling tower characteristic crosses the tower requirement curve.

Although the Merkel model is very good for manual calculation, where it is often checked against a diagram, one should be very careful if the equations are used in automatic calculations for control purposes. If, for example, L/G is varied over a large range, the equation system seems to have alternative solutions on the "right side" of the requirement curves. For an iterative solution of the equations, this could be critical. An example is shown below, calculated in EES (Engineering Equation Solver).

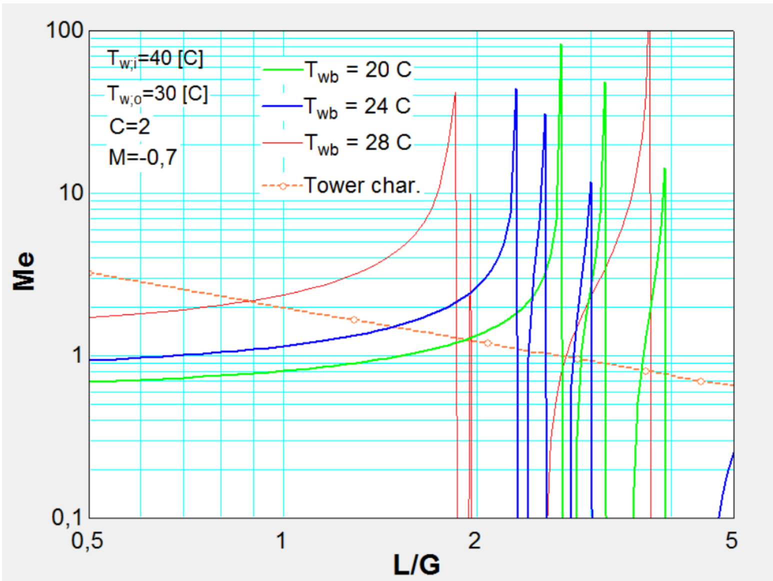

Figure 5-6 Multiple solutions resulting from applying EES

A solution on the “right side” of the requirement curves is off course not physically possible, but may work mathematically. It has not been attempted to find reasons or solutions for this problem in this project.

6 Control strategy considerations

The optimum control strategy should minimize the total operating costs of cooling tower and its “client”. These costs can be divided into subgroups:

- Cost of operating the “client”, for example the electricity cost for operating a compressor refrigeration system.
- Electricity cost for operating the cooling tower fans
- Electricity cost for operating the cooling tower water pumps
- Water related cost, such as cost of water supply, cost of water treatment, cost of dumping/cleaning bleed water.

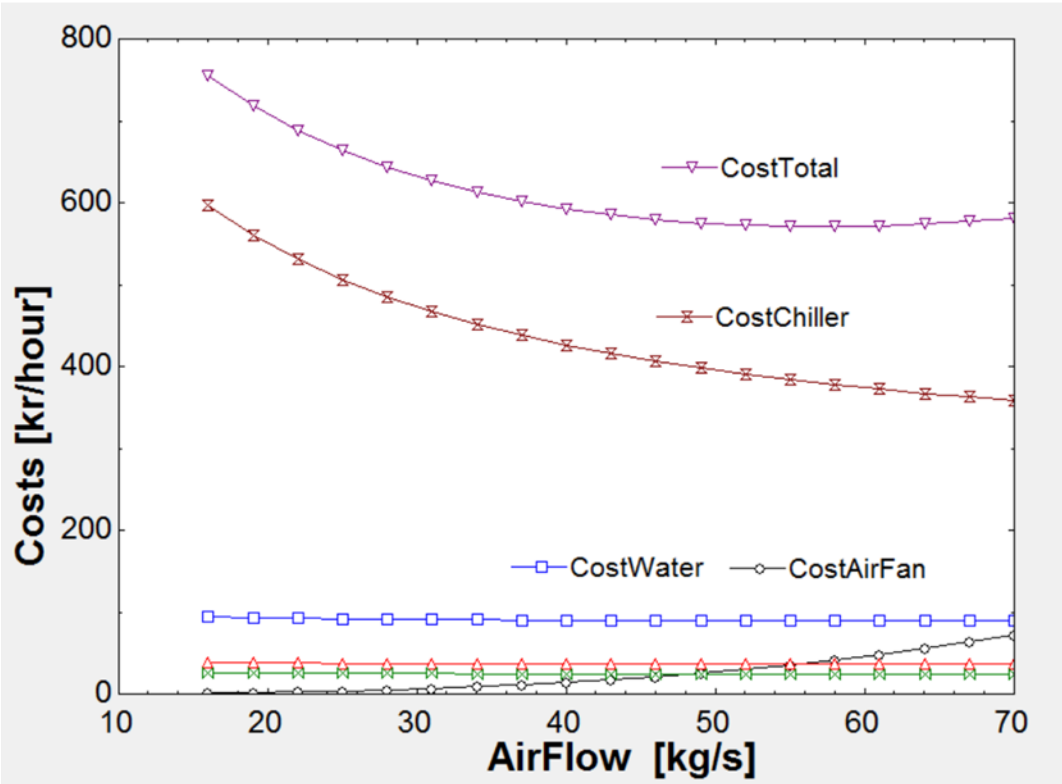

Figure 6.1 Cost breakdown for a cooling tower connected to a compressor refrigeration system. Example from Odense University Hospital.

In the graph above, hourly operating costs have been summarized for an actual cooling tower connected to a compressor refrigeration system at Odense University Hospital, running at a fixed capacity. It is evident that:

- Water related costs are practically constant, and independent on air flow over a very large range. So, water related costs are not relevant for optimization.
- The biggest variation in operating costs are found in fan power consumption and compressor power consumption.
- The sum of all operating costs has an optimum, which seems to be relatively “soft”, meaning that even a relatively large deviation from the optimum point causes a small increase in total cost.

In the case of variable capacity, the situation is a bit different. Off course it makes very little sense to run the cooling tower fans at full speed, if the cooling capacity is very low. The gain from low condensing temperature would then not outweigh the fan consumption.

An example, based on data from Odense University Hospital, is shown in the diagram below. It shows clearly, that total operating costs are very dependent on cooling capacity, and that the optimum air flow is very different at different capacities.

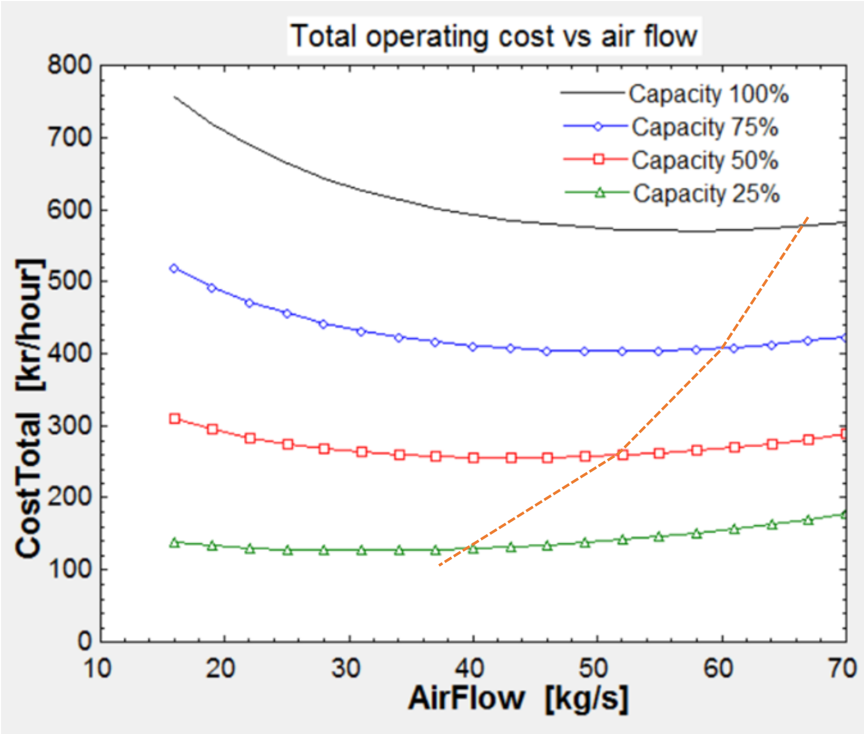

Figure 6.2 Total cost vs airflow for different cooling capacities. Optimum points indicated by orange dotted curve. Example from Odense University Hospital

Similar calculations have been undertaken to evaluate the optimum water pump flow. Also here, the calculations have been done based on a real life example at Odense University Hospital. The indications of these calculations are, that any reduction of the water flow leads to an increase in water temperature and thereby an increase in condensing temperature. Since the water pump consumption is relatively small, the power savings from a reduced flow does not outweigh the increased compressor consumption, unless the cooling capacity is very low.

A special problem arises when reducing the water flow: At reduced flows, the spray nozzles may not distribute the water evenly over the cooling tower fill material. This means that the actual operating surface area of the cooling tower is reduced. Mostly, no data are available to cover this phenomenon, so a reduction of water flow is an uncertain venture.

The conclusion of the arguments above must be, that an optimum control strategy should only be concerned with the fan power consumption and the “client” power consumption, attempting to find the optimum fan speed to minimize total power consumption.

6.1 Control strategy mathematics

Generally, in cooling towers and evaporating condensers, water temperature falls with increasing fan speed, leading to a reduction in “client” consumption, but also an increase in fan power consumption.

From a mathematical viewpoint, it is difficult to relate the “client” consumption to fan speed; it is mostly easier to relate “client” consumption to water (or condensation) temperature. So, in the following math, we will relate “client” consumption and fan power consumption to water (or condensation) temperature. A typical example is shown in the graph below.

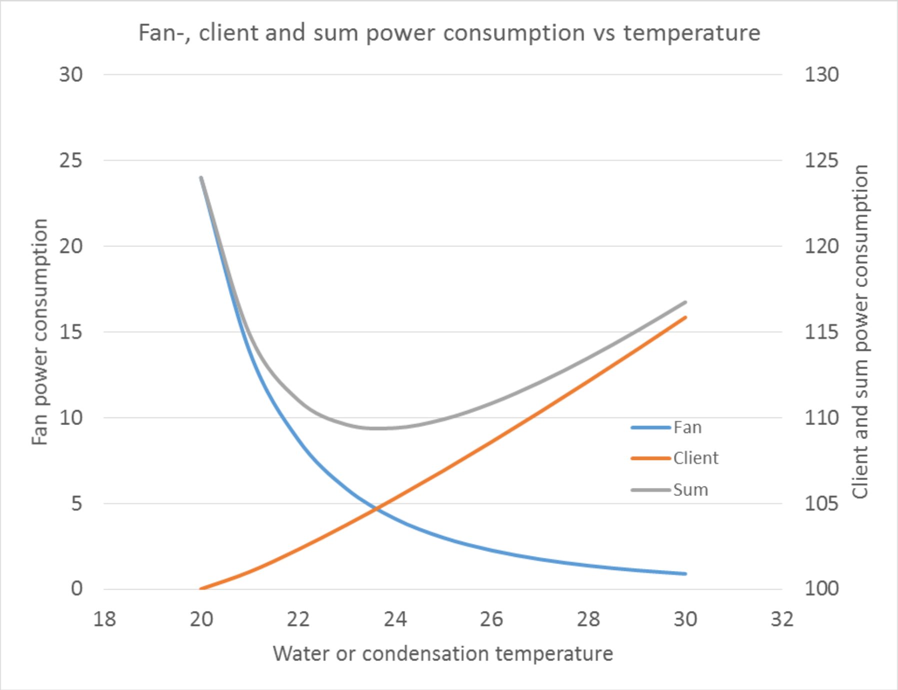

Figure 6.3 Power consumption (= cost) for fan and client and the sum of the two.

Mathematically, the optimum point is found where the slope of the fan consumption equals minus the slope of the client consumption. In other words:

$$\frac{dPower_{fan}}{dT} = -\frac{dPower_{client}}{dT}$$

Mathematically, it is off course very important to ensure that the derivatives are calculated at the same capacity, otherwise the calculation would not give the optimum point. To ensure this, we can modify the equation to (subscript cond meaning condenser/tower):

$$\frac{1}{Cap_{cond}} * \frac{dPower_{client}}{dT_{cond.}} = -\frac{1}{Cap_{cond}} * \frac{dPower_{fan}}{dT_{cond.}}$$

The left side expresses how much the client consumption would increase per degree increase in water temperature related to the capacity.
The right side expresses how much the condenser/tower fan consumption would decrease per degree increase in water temperature, related to the same capacity.

6.2 Cooling towers / evaporative condensers

Using the equation for capacity previously found for cooling towers and evaporative condensers, and assuming capacity to be proportional to Fanspeed^E, the equation can be written as:

$$Cap_{cond} = (A * T_{wetbulb} + B) * \left(\frac{T_{cond} - T_{wetbulb}}{C}\right)^D * FanSpeed^E$$

The equation is here written in the nomenclature of evaporating condensers (Cap_{cond} and T_{cond}). For cooling towers, these are replaced by Cap_{Tower} and $T_{w,i}$.

The actual fan power can be written as $P_{fan} = P_{nom} * Fanspeed^3$, where P_{nom} is the fan power consumption at full speed (Fanspeed is here taken as going from 0 to 1).

Rewriting the above capacity equation yields:

$$P_{fan} = P_{nom} * \left(\frac{Cap_{cond}}{A * T_{wet} + B}\right)^{\frac{3}{E}} * \left(\frac{1}{C}\right)^{\frac{-3*D}{E}} * (T_{cond} - T_{wet})^{\frac{-3*D}{E}}$$

For a given operating condition, the term $P_{nom} * \left(\frac{Cap_{cond}}{A*T_{wet}+B}\right)^{\frac{3}{E}} * \left(\frac{1}{C}\right)^{\frac{-3*D}{E}}$ is all constants, so it can be named K for simplicity.

So, $\frac{1}{Cap_{cond}} * \frac{dP_{fan}}{dT_{cond.}} = \frac{1}{Cap_{cond}} * K * \frac{-3*D}{E} * (T_{cond} - T_{wet})^{\left(\frac{-3*D}{E}-1\right)}$

For a given operating condition, given in terms of T_{cond}, $T_{wetbulb}$ and FanSpeed it is possible to calculate the actual condensing capacity. Based on this, $\frac{1}{Cap_{cond}} * \frac{dPower_{fan}}{dT_{cond.}}$ can be calculated. The desired value for this is $-\frac{1}{Cap_{cond}} * \frac{dPower_{client}}{dT_{cond.}}$, which for now can be called the SetPoint.

The optimum condensing temperature can now be calculated as:

$$T_{cond,new} = \left(Setpoint * Cap_{cond} * \frac{-E}{3 * D} * \frac{1}{K}\right)^{\frac{1}{\frac{-3*D}{E}-1}} + T_{wetbulb}$$

From the condensing capacity equation, this leads to a new fan speed:

$$FanSpeed_{new} = \left(\frac{Cap_{cond}}{A * T_{wetbulb} + B}\right)^{\frac{1}{E}} * \left(\frac{1}{C}\right)^{\frac{-D}{E}} * \left(T_{cond,new} - T_{wetbulb}\right)^{\frac{-D}{E}}$$

Waiting an appropriate amount of time, a new capacity calculation can be made, and the whole sequence can be repeated.

6.3 Compressor refrigeration systems

A very typical ”client” for a cooling tower or an evaporative condenser would be a compressor refrigeration system. The compressor refrigeration system removes heat/energy from a product and needs to discard this heat + the power consumption spent on compression to somewhere, typically the ambient.

In order to optimize the common operation of cooling tower and refrigeration system, we need to define the term

$$\frac{1}{Cap_{cond}} * \frac{dPower_{client}}{dT_{cond.}}$$

Instead of doing a lot of complex math, it is possible to use various existing calculation programs to evaluate this term. One possibility is the free program “CoolPack”, available from DTU, to calculate the power consumption of a refrigeration system.

http://www.en.ipu.dk/Indhold/refrigeration-and-energy-technology/coolpack.aspx

An example of such a calculation is shown in the diagram below. Here, an ammonia (R717) refrigeration system with a cooling capacity of 1 MW (chosen arbitrarily) and an isentropic efficiency of 0,7 and 0,8 (the typical realistic range) has been calculated at varying condensing temperatures and evaporation temperatures of 0, -10 and -20 °C.

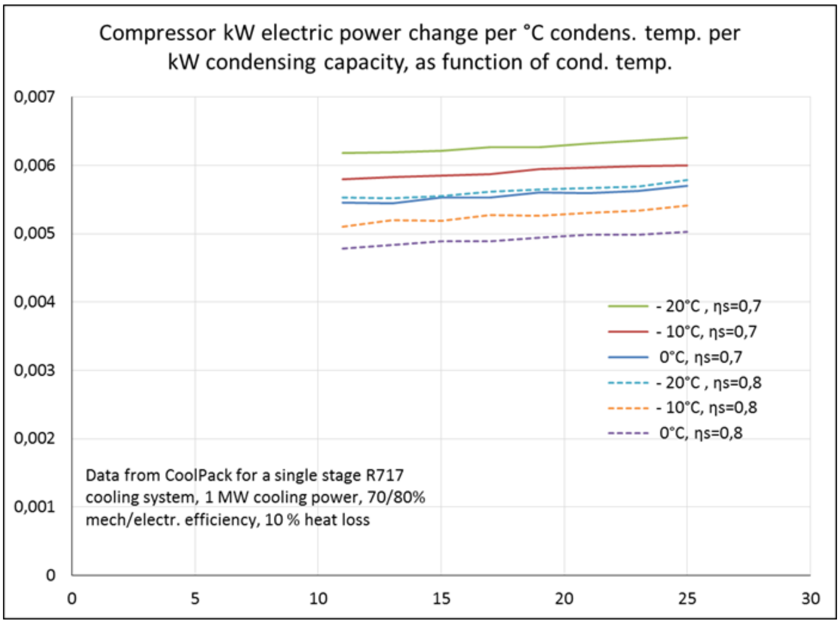

Figure 6.4 Compressor power consumption change per condensing capacity and per degree condensing temperature for a R717 refrigeration system

For a given evaporation temperature (which is usually fixed, since it is dependent on the actual operation mode), the value is rather constant. And even varying evaporation temperatures doesn't seem to produce much variation in the value.

This may seem counterintuitive, since the power consumption for removing heat at 0, -10 and -20°C varies quite a lot. It is important to realize, however, that we are not asking how much power it takes to remove a certain amount of heat; we are asking how much **extra** power it takes, if we increase the condensing temperature. The power consumption for the initial temperature lift is not of any concern here.

So, for most refrigeration systems, running at a rather fixed evaporation temperature, and with known isentropic efficiencies, it is reasonable to choose a fixed value:

$$\frac{1}{Cap_{cond}} * \frac{dPower_{client}}{dT_{cond.}} = Constant$$

where Constant in this case could be somewhere between 0,005 and 0,006. This Constant would then be used as the Setpoint for the condenser/tower calculation.

7 Partner setups

In the following, the cooling tower/condenser and “client” setup for the various partners will be described. In case measurements or experiments have been conducted, these will also be described.

7.1 DAKA

DAKA has a 5 stage evaporative concentrator for concentrating stick water from the processing of slaughterhouse by-products.

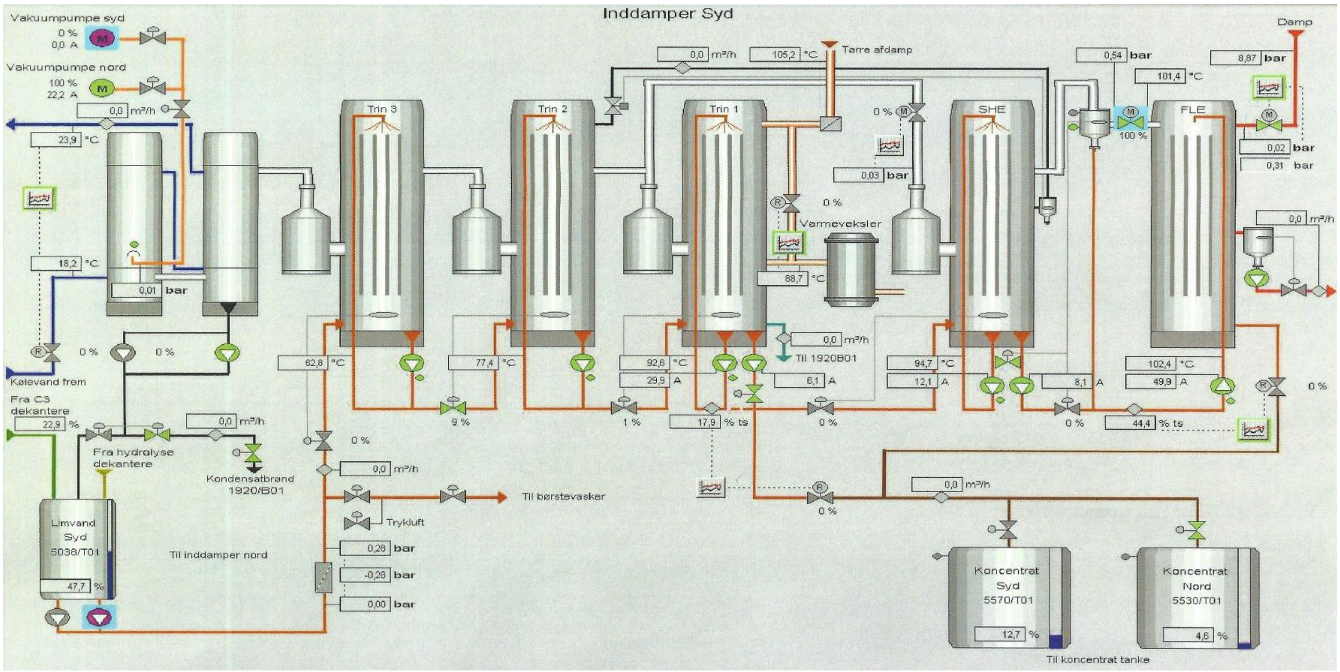

Figure 7.1 5-stage evaporative concentrator for stick water. Stage 1, 2 and 3 are heated with “free” waste heat from dryers. Stage FLE is steam-heated.

The stick water enters the evaporator from the stick water tank (bottom left) and passes through stage 3, stage 2 and finally stage 1. Stage 1 is heated by waste heat from dryers, in the form of hot, very humid air, typically 105°C hot. Stage 2 is heated by the vapors from stage 1, and stage 3 is heated by the vapors from stage 2. Finally, the vapors from stage 3 are condensed in a condenser (2 tanks at top left), cooled by water from an outdoor cooling tower.

If the product coming out of stage 1 is not concentrated sufficiently, steam is added to FLE, to drive out more water from the product. The dry matter percentage of the final product is measured, and this parameter is used to control the product flow. All 5 stages have a level control to make sure that they have a sufficient filling at all times.

In normal operation, the cooling water going out of the condenser and entering the cooling tower is set to a fixed 25°C. From thermodynamical considerations, it is expected that the temperatures decrease almost linearly from the waste heat through stage 1, 2 and 3 and finally down to the condenser. If the cooling water temperature could be further reduced, the temperature difference over each stage would increase, and since this temperature difference is the driving force for evaporating water from the product, the capacity of stage 1, 2 and 3 should increase.

In other words: The colder we can get the cooling water, the less need will there be for steam in stage FLE. To investigate this, an experiment was conducted, where the cooling water was lowered 6°. The experiment was performed in early January, where outdoor temperature made it possible to go far lower than the normal set point.

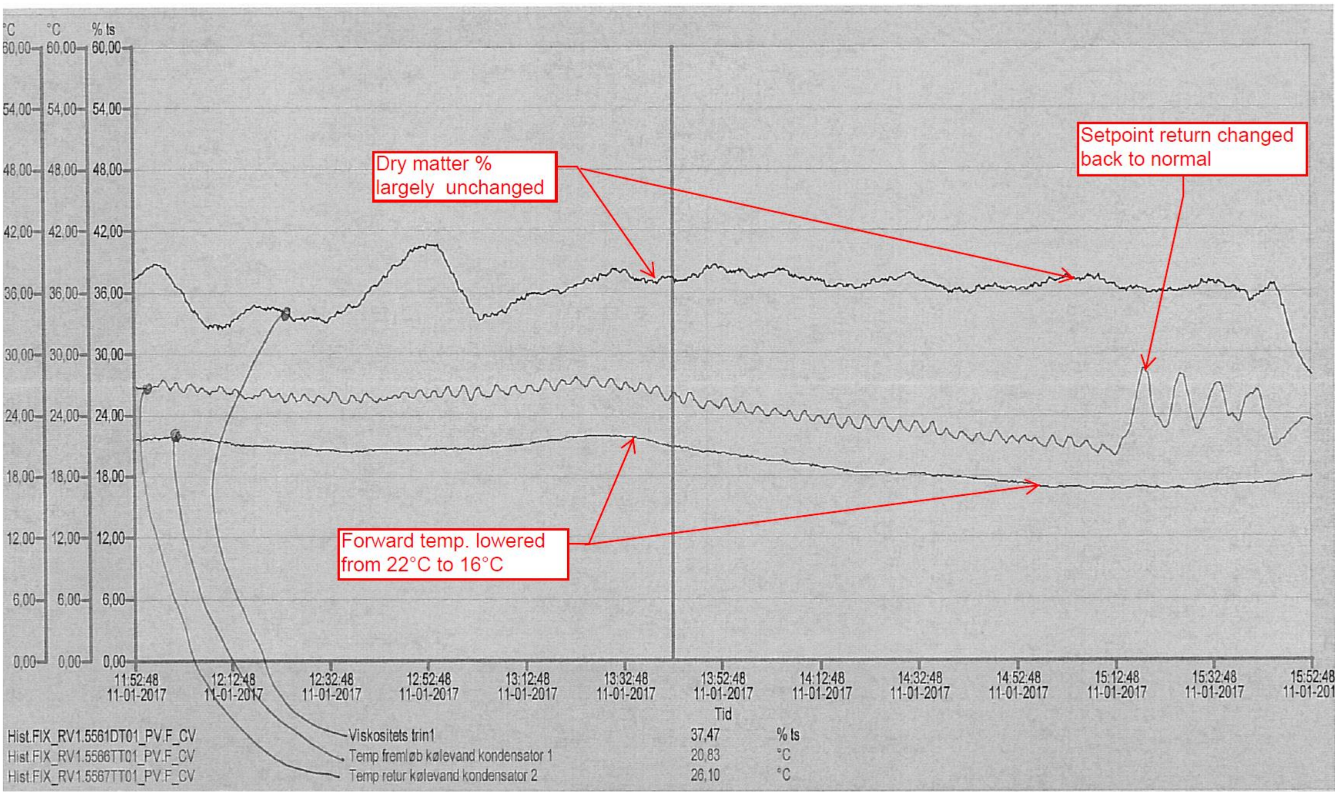

Figure 7.2 Data from first test with reduced cooling water temperature.

The cooling water set point was changed at approx. 13:30. Due to the very large quantity of water in the cooling tower and piping, the temperature changes very slowly. At 15:15, the water set points were returned to their original values.

Surprisingly, the dry matter percentage out of stage 1 did not increase when running at lower water temperature. Changing the set points back gave a relatively quick change in the return temperature from the condenser, but even this did not seem to have an effect on the dry matter percentage.

The same experiment has been repeated 2 times, in both cases without any positive results.

Various explanations have been investigated, why the cooling water temperature reduction did not result in a capacity increase. One theory could be, that the condenser contained large amount of air, either from the product or from leaks in the system. In that case, the air content would block the water vapors from condensing, thereby reducing the effect of the lower cooling water temperature.

This theory was at first partly confirmed by the fact that the condenser pressure was far above the vapor pressure of water at the condenser temperature. In addition, the pressure did not change appreciably during the experiment. In other words, pressure and temperature did not follow each other as expected.

Data for condenser vacuum is shown in the graph below. The vacuum is relative to atmospheric pressure, so -0,9 is equivalent to 0,1 bar absolute pressure.

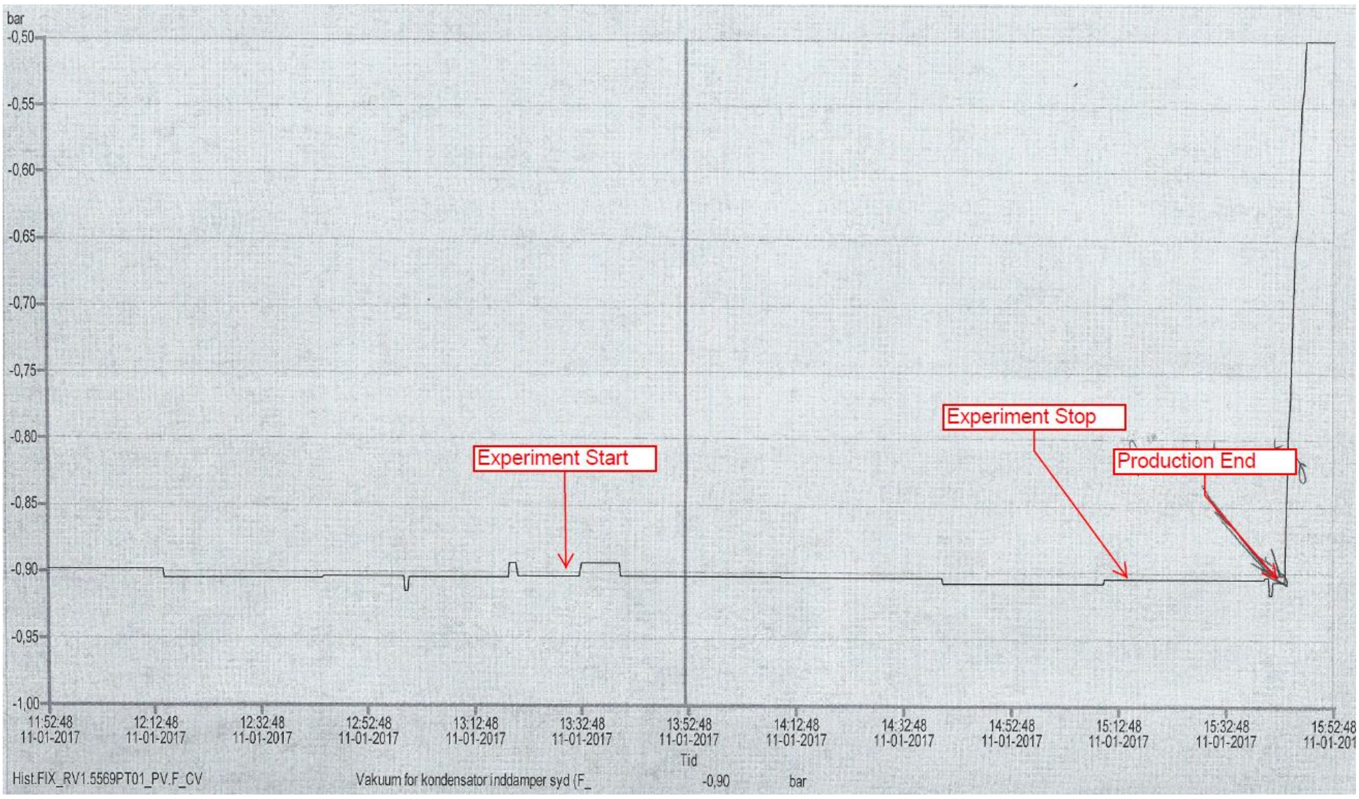

Figure 7.3 Data for condenser vacuum during the experiment.

Later experiments have run at lower pressure, but none have been under 0,05 bar absolute pressure. In comparison, the vapour pressure of water at 25°C is 0,03 bar abs.

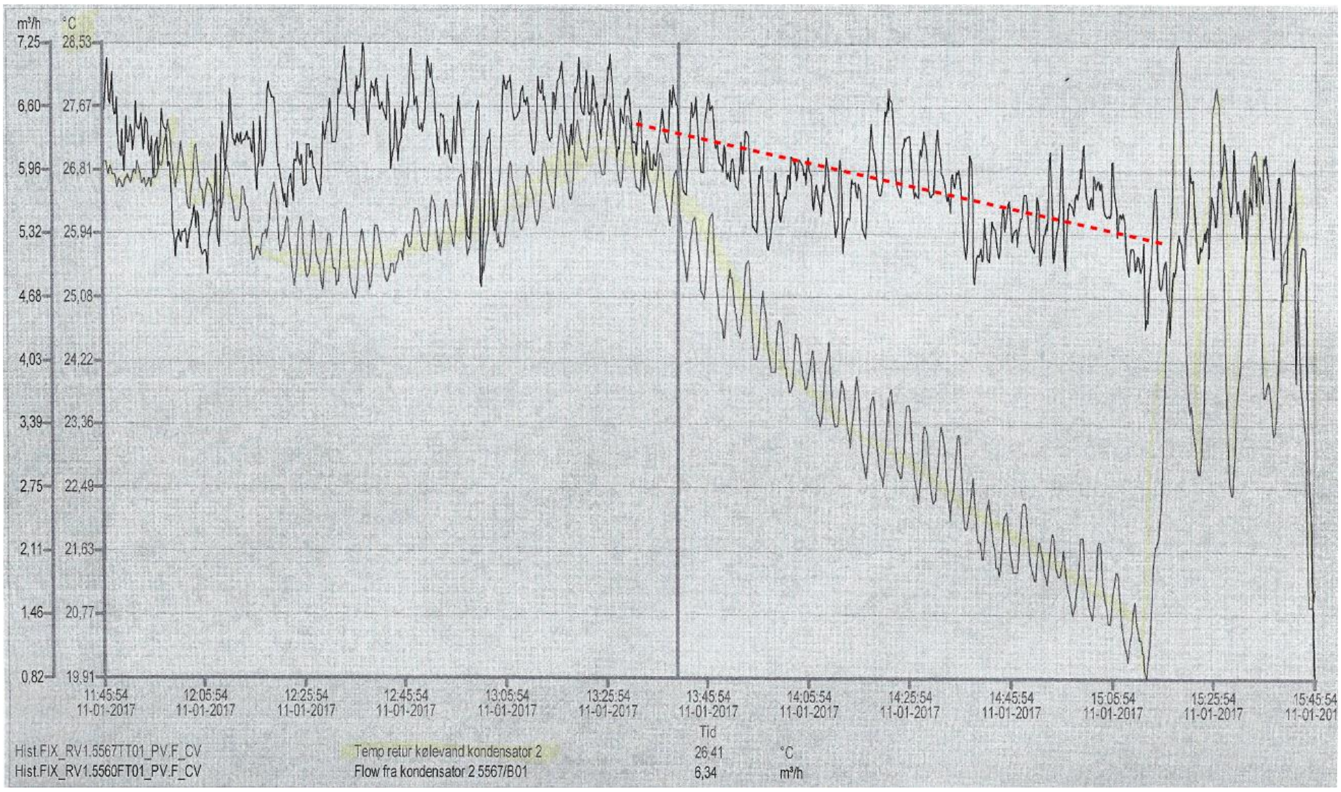

Figure 7.4 Data for cooling water temperature (yellow) and condensate flow

One possible indicator for the capacity of the concentrator system is the flow of condensed water out of the condenser. This flow should be more or less proportional to capacity. As can be seen on the diagram above, the condensate flow varies quite a bit during the experiment, but overall it seems to decrease (dotted red line) with falling cooling water temperature.

Another possibility to explain this seemingly illogical behavior could be that the supply of waste heat varies (decreases) during the experiments. Measurements indicate, though, that the waste heat temperature is very stable during the experiment. Actual waste heat flow can unfortunately not be measured.

7.1.1 Conclusion at DAKA

All in all, no conclusion has been drawn at DAKA, other than that it has not been possible to prove any savings based on reducing the cooling water temperature.

7.2 Arla DP

Arla DP has a large number of ice water chillers, all cooled by cooling towers. During the inspection on-site, it was quickly clear that:

- All units run at full capacity most of the time, because Arla DP has a rising demand for ice water and actually plans to put in more units. So the idea of using extra available capacity to produce savings was not relevant here.
- All units are separate single units, meaning that they each consist of 1 chiller and 1 cooling tower. There is physically no way to distribute capacity between them or connect them in “smarter” ways.

Based on these findings, it was decided to focus on other partners for now.

7.3 Arla Brabrand

Arla Brabrand has a relatively large ice water system, where water is cooled in 2 plate heat exchangers and later is fed through 2 large basins with ice-spirals.

For the plate heat exchangers, the evaporation temperature is around -1°C, and for the ice spirals, the evaporation temperature can go as far down as -10°C, depending on the thickness of the ice layer.

There are 7 compressors, whose suction lines can be connected independently to either plate heat exchangers or to ice spirals. The compressor types are:

- 2 pcs SMC 8-100, 252 kW (-1/+33); 164 kW (-10/+33)
- 2 pcs SMC 16-100, 504 kW (-1/+33); 327 kW (-10/+33)
- 1 pcs CR128, 520 kW (-1/+33); 367 kW (-10/+33)
- 1 pcs WMY 347H, 1276 kW (-1/+33); 895 kW (-10/+33)
- 1 pcs Frick, 1110 kW (-1/+33); 760 kW (-10/+33)

All in all, a total possible power of 4418 kW (-1/33) or 3004 kW (-10/33).

There are 2 evaporative condensers run in parallel. The type is Baltimore VXC 1360; each with 4 pcs 22 kW fans; nominal air flow in one condenser 104 m3/s; each with 2 pcs 4 kW water pumps; water flow in one condenser 93,4 l/s.

Each condenser has a nominal capacity of 1360 evaporator tons (4800 kW) at 96,3°F (35,7°C) condensing temperature, 20°F (-6,7°C) evaporation and 78°F (25,5°C) wet bulb temperature.

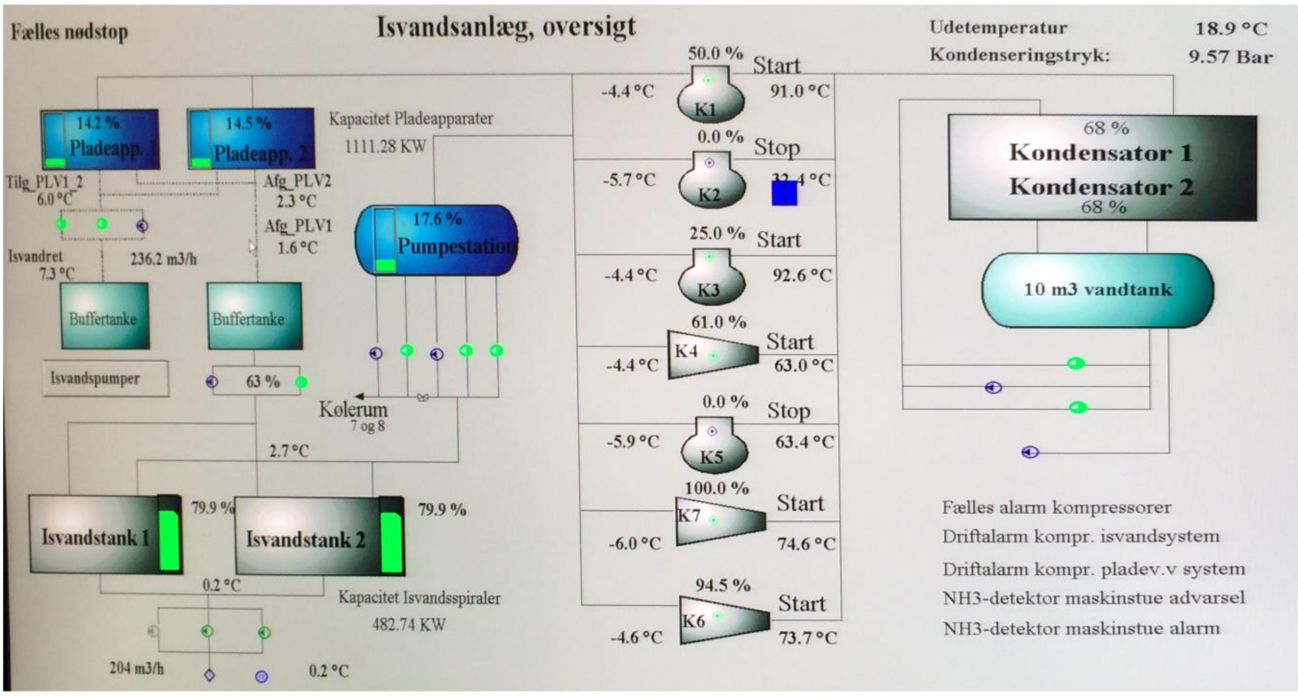

Figure 7.5 Overview of the refrigeration system at Arla Brabrand

7.3.1 Conclusion at Arla Brabrand

Although the total refrigeration system at Arla Brabrand is relatively complex, and there are several possibilities of selecting compressor units depending on the needed capacity, it was found that the setup was irrelevant in the context of this project.

The 2 evaporative condensers always run in parallel with the same fan speed. The fan speed is controlled by a so-called CP-optimizer, attempting to optimize the fan speed to obtain the minimal consumption of condensers and compressors together. In other words: doing practically what we want to do in this project.

Off course, it might be possible to obtain some savings by doing intelligent choices of compressors and by time-shifting some refrigeration operations. But this would mean optimizing the control of everything else than the condensers, which is NOT the main scope of this project.

7.4 Arla Rødkærsbro

Arla Rødkærsbro has a number of refrigeration compressors coupled to a common condensing line. The compressors supply cooling for various needs:

- Ice water production (typical evaporation temperature -1°C)
- Storage cooling (typical evaporation temperature -10°C)
- Spiral freezer (typical evaporation temperature -20°C)

The common condensing line is cooled by a number of evaporative condensers, all connected in parallel:

- 1 Baltimore VXC 1608. 4 fans, of which 2 have variable speed
- 2 Baltimore VXC 804, each with 2 fans with variable speed
- 1 older VXC 650, 2 fixed speed fans
- 1 older VXC 420, 1 fixed speed fan

7.4.1 Present control strategy

In the present control strategy, condensers are set into operation sequentially:

- First, the VXC 1608 is started on the fixed speed fans. If this is not sufficient, the variable speed fans are started and stepwise increased in speed.
- If this is not sufficient, 1 VXC 804 is started, and fan speed is increased stepwise.
- If this is not sufficient, the 2nd VXC 804 is started, fan speed increased stepwise.
- Finally, the VXC 650 and 420 are started. Data for 2016 indicate that this only happens rarely.

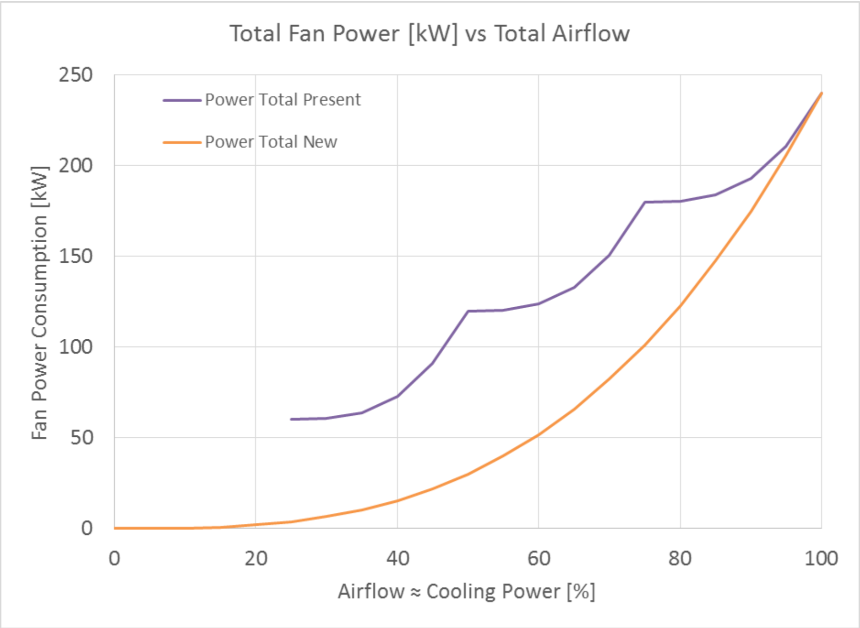

Figure 7.6 Fan power consumption for present and new control strategy.

The purple graph above shows the total fan power consumption of the VXC 1608 and 804 units. In the present control strategy, fans are started sequentially, and fan power increases with speed lifted to 3rd power. Full speed fan consumption is approx. 30 kW each.

All water pumps in the condensers are always kept running to keep the condensers clean, even if not in function. So, a new and improved control strategy could be to fit all fans with variable speed drives and start up all fans at low speed. Since fan power consumption is proportional to fan speed lifted to the 3rd power, it will always be cheaper to have many fans running at low speed than to have fewer fans running at high speed. And water pump consumption needs not to be taken into consideration, since they will be kept running anyway and have a much smaller consumption than the fans.

The orange graph shows the fan power consumption with the new strategy.

Operation data for 2016 on an hourly basis have been made available by Arla, and these indicate a possible annual saving of **290.000 kWh** by switching to this new control strategy.

In the present control strategy, the set point is a minimum condensing temperature, which is set at approx. 20°C. This minimum set point ensures that condenser pressure is always high enough to supply all compressor units with liquid ammonia. If the condensing temperature goes below minimum, fan speed is reduced to reduce the cooling.

The graph below shows data for 2016 for the actual condensing temperature vs ambient wet bulb temperature. Apparently, it is possible to maintain the minimum condensing temperature up to 8°C wet bulb temperature

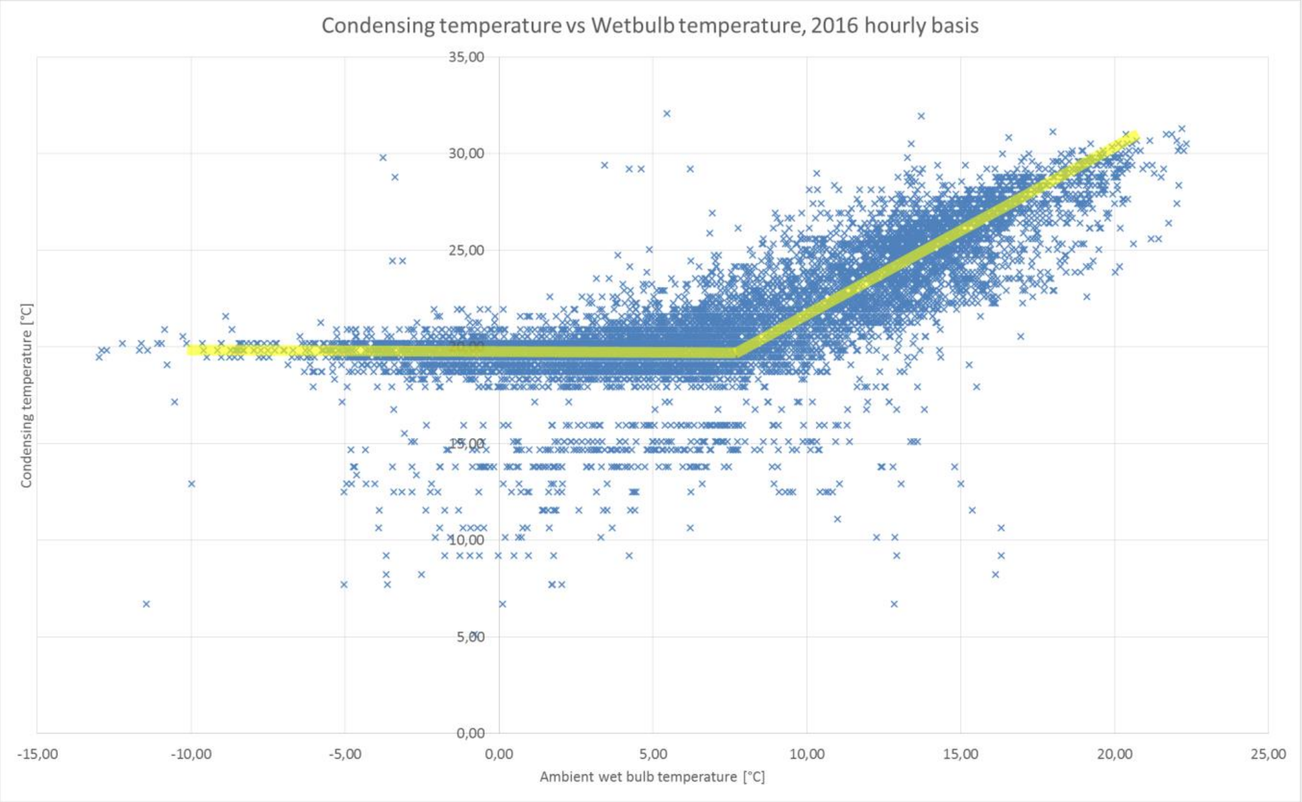

Figure 7.7 Condensing temperature vs Ambient wet bulb temperature for 2016

If it were possible to lower the minimum condensing temperature, the compressor consumption could be reduced. Based on the 2016 data, it has been estimated that the annual power consumption would decrease approx. **120.000 kWh per degree**.

During periods with high ambient temperature, where operation is NOT limited by the minimum condensing temperature, it is possible to optimize the fan speed, so that the combined consumption of fans and compressors is minimized. Assuming that all fans have been changed to operate in parallel, as previously described, it has been estimated that this optimization would lead to an annual saving of approx. **40.000 kWh**.

The various optimization procedures are awaiting test on site.

7.5 Odense University Hospital (OUH)

The University hospital in Odense has several cooling systems. The major system serves to cool scanners and other hospital machinery, and to provide air condition in the summer. The cooling system supplies a flow of cold brine to the users.

The major system is composed of 2 independent systems that do not operate simultaneously. The first system is a free-cooler, mainly used in winter time or other colder periods. During these periods, air conditioning is not needed, and a higher brine temperature can be used, still satisfying the cooling needs of scanners and other machinery.

In warmer periods, a compressor cooling system is used, cooling the same brine system as the free-cooler.

Typically, the switch-over between free-cooling and compressor cooling takes place at an ambient day-temperature of around 11°C. In practice, the set point for brine temperature is set at (for example) 13°C at low ambient temperature, and the set point for brine is changed to 9°C at high ambient temperatures. This forces the compressor system ON at high temperature.

In 2015, compressor cooling was started up in April and ended operation in October. The rest of the year was operated on free-cooling.

Typical evaporation temperature for the compressor system is 6°C, and typical condensation temperature is 30-40°C.

The cooling tower uses de-ionized water, produced by the hospitals own utility service.

The diagram below shows the setup, with the “customer” cooling water entering and leaving on the right side. The 2 compressor systems can run independently

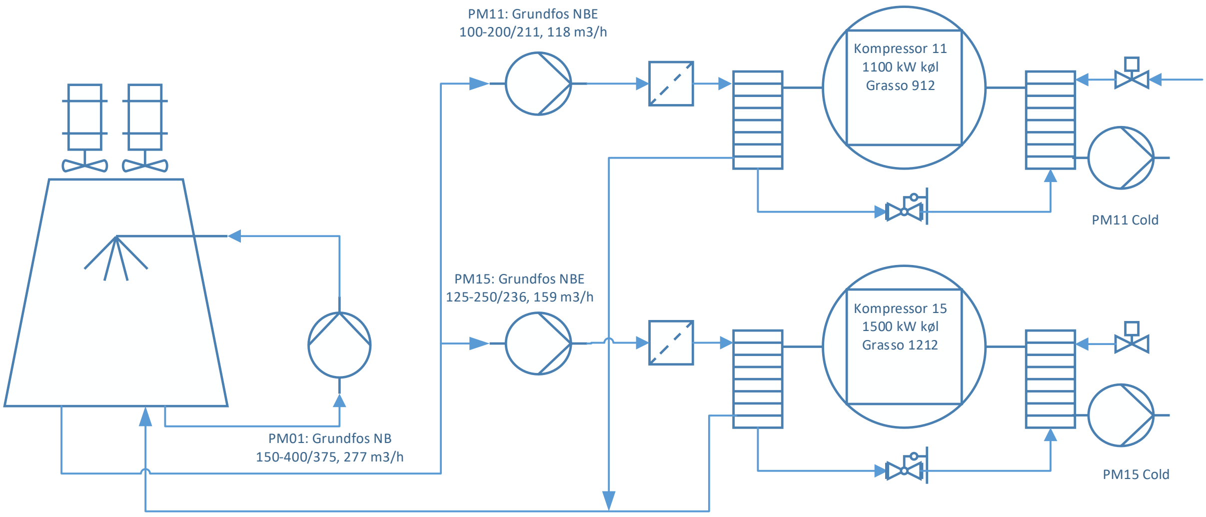

Figure 7.8 Cooling system setup at Odense University Hospital

The 912-compressor is a 9-cylinder machine, that can control capacity by coupling in cylinders in numbers from 3 (lowest capacity) to 9 in 7 steps of each 1 cylinder. Likewise, the 1212 compressor is a 12-cylinder machine, running from 3 to 12 cylinders in 10 steps. In practice, this means that capacity can be controlled almost continuously.

Fans and pumps are all controlled by frequency converters, so speed (and thereby flow) can be controlled continuously. Notice that the circulation of water in the tower is here independent from the circulation of water in the condenser(s). This adds more items to consider when trying to optimize the system.

7.5.1 Control of the system

The compressor system runs almost autonomously, controlled by the water temperature leaving through pumps PM11 Cold and PM15 Cold. The set point for this temperature is typically 8°C, and if the set point is exceeded, more cylinders will be coupled in to increase performance. The hysteresis is typically 1,5°C, so the compressors will reduce the active number of cylinders if the water temperature falls below 6,5°C.

The running speed of pumps PM11 Cold and PM15 Cold has a profound influence on the evaporation temperature. The higher speed, the higher is the temperature, thereby giving a higher cooling power per kilowatt electricity used by the compressors (COP-factor). It can be shown that the compressor consumption is very close to being a linear function of the evaporation temperature.

Likewise, the running speed of pumps PM11 and PM15 on the condenser side has a profound influence on the condensation temperature. The higher speed, the lower temperature and thereby a higher COP-factor. The compressor consumption is very close to being a linear function of the condensation temperature.

More interestingly, it can be shown, that the optimum speed of the pumps on the cold side is practically independent on the optimum speed of the pumps on the hot side. In other words, the saving achieved by increasing the evaporation temperature by 1° at a condensation temperature of X°C is practically the same as the one achieved at a condensation temperature of Y°C, within reasonable limits of X and Y.

This means that the cold side pump speeds can be optimized independently of the rest of the system. In practice, this is done by adjusting the pump speeds until the ratio of cooling power to the sum of pumps and compressor consumption (COP-factor including the pumps consumption) is maximal.

The procedure would be as follows:

- The water outlet temperature is assumed to be at set point (or maybe in the middle of the hysteresis loop). This may not be correct at the start of the system, but will eventually be correct once the system is up and running.
- The compressors have xx cylinders coupled in.
- For each possible water flow, the evaporator (heat exchanger) equation is solved, using a linear relationship between the cooling power and the evaporation temperature. The linear equation will have to be created first, using 1 actual operating condition (maybe brochure value) and extrapolating with a program such as CoolPack. The heat exchanger data must be taken from suppliers data or from real operating conditions.
- The water flow giving the highest COP-value (including pump consumption) will be chosen.

On the condenser side, one might argue that the compressor consumption is also a linear function of condensing temperature, and that the linearity is almost independent on the water temperature coming from the cooling tower, so the pump speed can be trimmed independently. This argument doesn't hold, however. If the pump speed is increased, then the water temperature out of the condenser will be lower. This means that the cooling tower is now getting colder water from the condensers, and will have more difficulty getting rid of the condenser heat, since the temperature difference to the ambient is now smaller. This means that the temperature level of the cooling tower must now increase. In other words: increasing pump speed may give lower condensing temperatures, but not as much lower as expected, based on a heat exchanger calculation alone.

In the OUH case, things are further complicated by the fact that the cooling tower circulation pump does not necessarily circulate the same amount of water as the condenser pumps. If the cooling tower circulation pump has a smaller flow than the condenser pumps, some of the hot water from the condensers will be mixed back to the cold-water side. If the circulation pump has a higher flow than the condenser pumps, some of the water from the cold side will be mixed into the hot water going to the cooling tower.

7.5.2 Conclusion OUH

Since the compressor system is only in operation during the summer time, tests at OUH have been postponed relative to other partners. In practice, this means that no tests have been conducted so far.