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STRATEGY FOR THE OPERATION OF COOLING TOWERS WITH VARIABLE SPEED FANS

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Abstract

Within the SPS Cooling Water Project at CERN aimed at the reduction of water consumption, this primary open cooling loop will be closed and all the primary cooling circuit components will be upgraded to the new required duty and brought to the necessary safety and operability standards. In particular the tower fans will be fitted with variable frequency drives to replace the existing two speed motors. This paper presents a study to optimize the operation of SPS cooling towers taking into account outdoor conditions (wet and dry bulb temperatures) and the entirety of the primary circuit in which they will operate.

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1 INTRODUCTION

CERN Super Proton Synchroton (SPS) was built in the mid seventies to allow for fixed target physics of energies up to 300 GeV. Although very much modified afterwards to accommodate different new uses, its primary infrastructure has remained much like at the commissioning point up to this day.

The concept of the cooling system of the SPS was simple: A primary pumping station pumped town water into a distribution network. The primary circuit cooled the secondary cooling plants housed in auxiliary buildings called BAs. These plants were composed of a number of circuits, each featuring its own heat exchanger and secondary cooling circuits. The secondary circuits were closed and operated on different thermal and electrical conductivity parameters as required by the users' equipment. After serving the BAs, the primary circuit conveyed the heated water (design capacity of 52 MW) to the SPS cooling towers. The water at a permitted temperature is discharged to the nearby Nant d´Avril river without posing any biological hazard. Already from its conception the SPS primary pumping station featured a number of variable speed motors which guaranteed a constant inlet-outlet differential pressure available for each secondary cooling plant, regardless of the flow required. The cooling tower fans were equipped with two-speed motors.

Within the SPS Cooling Water Project at CERN, aimed at the reduction of water consumption, this primary open cooling loop will be closed and all the primary cooling circuit components will be upgraded to the new required duty and brought to the necessary safety and operability standards. To this end, the mechanical draft counterflow cooling tower sets made of reinforced concrete will see their water distribution system, drift eliminator, fill packing, air inlet louvers, motors, gear reducers and fans replaced. At the same time, the upgraded cooling towers' fans will be equipped with variable speed drives (VFD), which will increase their versatility with respect to the current cooling towers equipped with 2-speed motors. Similarly, the BAs primary heat exchangers, control valves, control systems and the SPS primary pumping station (La Berne) will also undergo serious modifications to adapt to the new working parameters.

This paper presents a study to optimize the operation of SPS cooling towers taking into account outdoor conditions (wet and dry bulb temperatures) and the entirety of the primary circuit in which they will operate. The different simplifying assumptions adopted in the model, their validity and consequences in the range of application and the analytical model itself are presented. Ultimately, the final chapter of the paper contains the discussion of results and the conclusions and gives suggestions for further steps to be taken.

2 OBJECTIVES

The objective of the present work is twofold. First, to establish a one-dimensional analytical model of the behaviour of the SPS primary cooling circuit (primary pumping station-heat exchangers-cooling towers) off-design conditions.

The second objective is to obtain a series of feasible operation points for the reduction of instantaneous power consumption and noise generated (hereafter referred to as operation costs, comprising the pumping and ventilation energy expenses only). The number of points analysed has been kept low to reduce the calculation time and complexity. The methodology employed remains valid, however, and can be applied to more elaborate models without loss of generality.

The study of this first model will help determine the validity of the analysis and it will give orders of magnitude of the operation costs savings as compared to the previous situation. Further, it will hint directions as to the fashion in which the model may be developed in the mid term to permit the prediction of optimal operation work points for different operation conditions.

3 COOLING CIRCUIT COMPONENTS FUNDAMENTALS

The fundamentals of the operation of the different components involved (cooling towers, heat exchangers and hydronic circuits) will be revisited here in a simplified manner, pointing out only the relevant elements. The reader already familiar with the concepts being described is advised to skip this section. Uninitiated persons will find the literature proposed in the References chapter useful.

3.1 Cooling Towers Fundamentals

The physics behind the operation of cooling towers are rather complex as different variables and transfer mechanisms are simultaneously involved.

A cooling tower cools down hot water by the evaporative cooling mechanism and the exchange of sensible heat between the air and the water. Part of the water exposed to the cooling air stream evaporates releasing the latent heat of vaporisation (2514 kJ/kg approx.) into the air. This heat is taken from the water, which thus lowers its temperature at the rate of some 7 K per each 1 % of the total water flow that is evaporated. Sensible heat is responsible also for a portion of the cooling accomplished when the water is warmer than the outside air. For a given cooling tower it may be possible that both sensible cooling and heating are present as the water varies its temperature along the tower. Well-designed cooling towers transfer about 90 % of their total capacity as latent heat transfer when working close to their design conditions. There is a penalty involved in this process: the loss of water discharged into the atmosphere as hot steam. A steam plume shows if, under certain conditions, the air stream goes oversaturated, usually in cold weather.

The hot water circulated into the tower is distributed over its entire cross section in the form of fine droplets. They fall onto a deck fill having as function the provision of very large water surface areas for heat transfer. Each water surface in the tower is surrounded by a film of saturated air at the water temperature. The physical properties of both the water and the air change according to their relative position in the tower.

Most cooling towers manufacturers employ Merkel's performance analysis, developed in the mid twenties, which includes the sensible and latent components of the heat transfer process described above into an overall heat and mass transfer process.

Due to the fact that both the dry bulb (driving the sensible heat transfer) and the wet bulb (driving the latent part of the overall transfer) temperatures are involved, Merkel introduced the idea of the enthalpy potential, employed to this day.

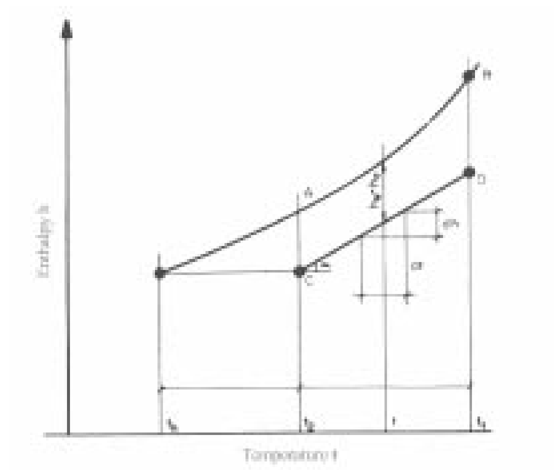
The heat transfer process taking place around the droplets depends on:

- the typical size (available transfer area, *a*) of the droplets (which in turn is a function of the type of fill and the flow rate falling onto it),
- its bulk temperature (t_w),
- the overall heat transfer coefficient (*K*), including contact resistance between the water droplet and the air (which depends primarily on the local air velocity),
- the water (*L*) flow rate
- the saturated air enthalpy for a given temperature (h_w)
- the actual enthalpy of the air at a given point in the cooling tower (h_a)
- the air (*G*) flow rates and inlet and outlet temperatures (t_{w1} and t_{w2})
- the actual temperature of the water at a given point in the cooling tower (t_w)
- the entire volume (*V*) of the cooling tower.

Merkel's idea for the performance analysis hinges on the use of a non dimensional equation that relates all the variables listed above for the definition of the effectiveness of a given cooling tower. This equation has the form:

$$\frac{KaV}{L} = C_{pw} \int_{tw2}^{tw1} \frac{dt_w}{(h_w - h_a)}$$

The notion of enthalpy potential can be observed from the figure shown below and the shape of the equation. It relates to the surface between the lines of enthalpy of saturated air (AB) and the enthalpy of the air flowing in the cooling tower (CD). The considered area covers the temperature range from the hot to the cold water.



It shall be clear at this point for the non-specialist reader that the prediction of the behaviour of a cooling tower is a most difficult task. It relies heavily on manufacturers proprietary test data, mainly focused towards the design operation conditions only, leave alone off-design operation predictions. For these reasons the existing literature is extensive and sketchy, sometimes even contradictory.

3.2 Plate Heat Exchangers Fundamentals

The main advantage of the plate heat exchanger is its compactness. To compensate for the reduced area of this type of exchanger higher efficiency (higher heat transfer coefficients and almost full counter-flow pattern) is necessary. This is achieved by relentless search in the field of plate corrugations. Corrugations increase heat transfer by increasing the plate effective area and generating turbulence by continuous change in the fluid velocity.

The two most important parameters defining a plate heat exchangers are the maximum flow capacity and the Number of Transfer Units (*NTU*) range of the plates. The *NTU* is defined as the temperature change in one fluid divided by the Logarithmic Mean Temperature Difference (*LMTD*). In terms of other variables involved (overall heat transfer coefficient U, an individual's plate exchange surface A_T, and mass flow rate and specific heat of the fluid of lower thermal capacity), it can be written as:

$$NTU = \frac{2UA_T}{\dot{M}\,C_p}$$

It must be stressed that the performance data available in the literature can only be considered as approximate in that it reflects the order of magnitude of the transfer capabilities, the performance of the many different corrugation forms varying greatly with design. The heat transfer capabilities of the plates (U) can be calculated from the usual non-dimensional heat transfer correlations, which remain generally applicable since the functional dependency on non-dimensional parameters remains obviously unchanged. Accurate values for a particular set of exchangers can only be derived from the adaptation of the non-dimensional correlations to the case under study via the constants and coefficients.

3.3 The SPS Hydronic Circuit

There are a few peculiarities that need mentioning at this stage. One is the fact that the SPS primary circuit is fitted with VFD for the pumps motors, the control being driven, in a simplified manner, by the level in an open regulation reservoir (in the highest altimetric point of the primary loop) and the differential pressure available for each SPS BA.

A second peculiarity is the flow limitation on the whole circuit at a flow rate only some 10 % above the design value of the circuit. This limitation comes from the internal pressure drops of the different components of the SPS primary circuit after the regulation reservoir and has played an important role in the re-engineering of the system, mainly the cooling towers.

4 ASSUMPTIONS

This section deals with the main assumptions which could affect the generality of the methodology adopted. Most of these assumptions are intimately related to the mathematical formulation of the problem. Therefore a thorough knowledge of the theory is required to fully understand their implications.

The first assumption made is that the Lewis number (Le, non-dimensional parameter that relates the importance of the heat transfer to the mass transfer effects) remains close to unity and that its value has no effect on the thermodynamic evaluation. This assumption is in fact the starting point for the derivation of Merkel's equation and is widely accepted in the literature to have little effect on the solution (only second order terms are involved). Recent studies confirm that large variations (± 25 %) have very little effect on the tower performance (of the order of the hundredth of the Kelvin for the cold water temperature at the design conditions). Another assumption, implicit in Merkel's formulation, and which will be only quoted briefly here, is the fact that the water droplets are considered to be at the water bulk temperature. The layer of saturated air around the water droplets is considered to be also at the water bulk temperature, thus neglecting in both cases the internal conduction and the existence of film (interface) resistances.

Although Merkel's theory indicates that the entering wet bulb temperature (and the inlet warm water temperature) should not affect the determination of the efficiency of the tower, some researchers do indicate such a dependence. This dependency has been analysed by the authors on the basis of the data available (performance and cooling guarantee curves). The results indicate important performance differences, hinting one of the many directions to be investigated further. The apparent reason for this departure from the design values seems to lie in the fact that Merkel's theory neglects both the film resistances, as mentioned above, and the evaporation of a fraction of the water flow rate. The first of these two effects obviously reduces the real enthalpy potential available making the one-dimensional Merkel model to give over-estimated (non-conservative) results. A more extensive study on the influence of these and other parameters can be found in the References chapter.

Further, only the stationary case has been taken into account in this study. It has been considered that load variations in the SPS circuit, are of secondary importance as most SPS accelerators's components show, when seen from the primary cooling side, a very stable behaviour in time and, in any case, have very little effect on the primary circuit in view of the magnitudes involved.

It has been considered that constant enthalpy lines coincide with constant wet bulb temperature lines in the psychrometric diagram. In the particular case of the design operation point of the SPS cooling towers, the error thus made in the value of the enthalpy due to this assumption is below 0.6 %, reason for which they have been neglected in this analysis.

For computational purposes the numerical formulation of the saturation line proposed by Hyland and Wexler have been preferred over others like Tezuka & Fujita, which although presenting much lower errors with respect to the tabulated values calculated by NASA[1] or JSME[2] (± 0.05 %) show the disadvantage of having a much shorter range of temperature application over the freezing point.

[1] USA "National Aeronautics and Space Administration"
[2] Japan Society of Mechanical Engineers

For the numerical integration of Merkel's equation, a very accurate integration (around 100 points) has been preferred to the more wide spread Tchebycheff's method, recommended by the Cooling Tower Institute codes.

Regarding the calculation of the heat transfer coefficients on the exchangers plates both the laminar and turbulent regimes are considered in the simulation. Nevertheless, it is known that the results obtained for Re numbers much below 1000 (critical Re) lack accuracy although this condition has not been checked here (no data were available at those Re numbers for the existing or the new set of exchangers). The transition regime (between the laminar and turbulent regions) has been interpolated from the previous regions where the flow is well characterised. For the practical purpose of this study the laminar region lacks interest as the flow rate range being investigated falls well above the critical Re. The primary flow rates investigated are those for which manufacturer´s data are available. Needless to say, the limitation in the primary cooling flow rate does not affect the generality of the methodology applied.

Finally, in what regards the calculation of the "operation costs" at partial loads, typical values for the efficiency of the VFD have been obtained from the literature in absence of manufacturer´s data.

5 METHODOLOGY

For an arbitrary value of the outdoor wet bulb temperature, an educated guess is made on the likely cold primary water temperature (t_{wo} or t_{w2}). No temperature increase has been allowed for due to the thermal effect on the primary water of the pumping power (over 1 MW installed), as the temperature rise involved at the design flow rate would be less than 0.2 K over the entire SPS loop (the rise is of similar value for partial loads). Consequently, the same cold water temperature has been considered for all the SPS cooling plants.

This cold water temperature is used to calculate each of the different (circuits) heat exchangers, resolving the necessary primary flow rate iteratively for each of them until the required duty is obtained. This is done by assuming an initial overall heat transfer coefficient (U) and imposing the highest possible primary water outlet temperature (below "crossing" conditions).

There exists at least one (trivial) solution to the problem, that is the design flow rates and temperature difference happening at lower primary temperature values (although not exactly the same values as the thermodynamic properties of the fluid will have changed). Reasonably, other solutions (lower primary flow rate and higher temperature difference) may exist which show the same or higher capacity, as in fact they do.

Since the duty of each particular heat exchanger is known (both power and secondary side parameters – flow rate and temperatures - remain constant) the primary flow rate can be calculated from the primary-secondary side power balance. This primary flow rate is used to calculate the film coefficient on the primary side of the exchanger on the basis of the working temperatures and the geometrical parameters of the plates, which allows to obtain a new U coefficient. This new coefficient is compared to the previous one and an iterative process repeated until convergence is found.

The exchanger´s capacity resolved in this way still is checked for the required duty. If the duty is not met (the flow rate is too low to reach the minimum necessary U value) the primary water outlet temperature is reduced, the flow rate increased and the process repeated until the lowest flow rate solution for that particular cold water inlet temperature is found.

The total flow rate necessary for the SPS power is added up and the return (warm) primary water temperature (t_{w1} or t_{wi}) averaged. Once the necessary primary flow rate is known the pumping operation cost can be calculated assuming efficiency values for the pumps motors and the VFD at partial loads.

The warm water temperature known, the cooling tower can be calculated. Several possibilities are available at this stage regarding the distribution of the total water flow amongst the cells available. Taking the dry-out condition (the point where not enough liquid is fed into the tower to ensure that all the distribution channels are filled) as the lower limit for the flow rate per cell, discrete flow rate increments are applied and investigated in order to determine the optimal layout.

On the basis of the water temperature regime and air conditions, Merkel's integral is evaluated and the (required) thermodynamic characteristic obtained. This value is compared to the actual characteristic of the tower for a chosen water to air flow rates ratio.

If the required value is higher than the actual characteristic of the tower, obtained from the manufacturer's data, the tower cannot meet the required duty and the cold temperature assumed is not a valid start point. The tower's capacity is being exceeded (the cold water is too close to the wet bulb temperature or put otherwise the approach is too low) and a solution does not exist for those values. The way to proceed in that case is to modify the air flow rate to increase the actual tower's characteristic until a solution be found for the assumed cold water temperature. If no solution is found within the available water and air flow rate ranges, the assumption on the cold water temperature must be modified and the cycle of calculations repeated for the new input values.

If, however, the characteristic required is lower than the actual manufacturer's guaranteed value, then the tower is up to meet the required duty although it could be that the actual thermal characteristic may largely exceed the value calculated. If this were the case, the cold water temperature assumed would be too high, and also in this case some corrections are needed. This is done by reducing the air flow rate into the tower until the required and actual values of the tower characteristics coincide.

Once the minimum air flow rate configuration is solved the entire set of thermodynamic properties of the air and water are calculated and the fan power obtained. This is done by discretizing the total water temperature range into equal steps (called "stories") and applying separate heat and mass balances to the counterflow water and air streams. This provides the air enthalpy at any point from which the outlet temperature, enthalpy and humidity can be obtained.

This scheme provides a series of possible operation points of the whole SPS primary loop, which can be plotted in terms of the (arbitrary) outdoor conditions to show the best (lowest "operation costs") point of operation and the order of magnitude of the expectable operation savings.

6 RESULTS

The results obtained prove the hypothesis, advanced by the authors when the study begun, that the optimum point of operation of a complex system like the SPS primary is not that of lowest possible cold water temperature. This latter tendency, found in the literature, is obviously the case when only refrigerating machines are being cooled, as their COP depends strongly on the condensation temperature. This actually implies a reduction of the cooling load as the cold water temperature decreases. The SPS however has a much larger proportion of 'fixed" cooling loads (above 80 % of the load) than on "variable" loads.

In the case of the SPS it can be seen that, depending on the wet bulb temperature outside, the minimum power consumption point is obtained for temperatures halfway between the lowest attainable and the design cold water temperature. In terms of the rotational speed of the fans, the optimum power consumption is found (for the case of t_{wb}= 7 °C for instance) some 250 rpm fan speed below the coldest water temperature condition. The power consumption reduction obtained by working at this optimum consumption point, when compared to the coldest water temperature point is around 100 kW. According to the meteorological data available for the Geneva region, the SPS primary runs some 2,000 hours with the wet bulb temperature below or equal to 7°C (which make some 200,000 kWh of energy). Moreover, if compared to the maximum water temperature operation point, one obtains a very similar result, with even somewhat higher gains. The complete family of curves for a range of outside wet bulb temperatures can be seen in Appendix 2, along with the frequency distribution of wet bulb temperatures in the Geneva region, as published by the Cointrin Airport Authorities.

In parallel to the energy implications, one can refer to the reduction in noise generated by the fans as these are run at lower rotational speeds. From a purely theoretical point of view, the reduction in noise generated by a rotating piece of machinery when the speed is reduced from N to N' is given by Ponsonnet's equation, which reads:

$$\Delta Lw = 50 \log \frac{N}{N'}$$

When applied to the case of 7°C, this makes a difference of 5.7 dB.

A possible different way to use the results could be to switch to the minimum air flow rate operation points during the night hours. These points, although not optimal from the whole system energy consumption stand point, present the advantage of reducing greatly the rotational speed of the fans, from 1300 rpm at the point of lowest cold water temperature to 825 rpm for the highest water temperature point. This represents a noise reduction of 9.9 dB.

7 CONCLUSIONS

This article has examined the different valid working points in the SPS primary circuit for given ambient conditions that the use of VFDs for the regulation of the BA6 cooling tower's fans and pumps permit. To do so, a one-dimensional model of the SPS primary circuit has been done. The model has taken account of the hydraulic and thermal characteristics of the different components in the SPS primary circuit including the BA6 cooling tower. The cooling processes in the BA6 cooling tower have been considered by calculating the KaV/L for different wet bulb temperatures, inlet and outlet water temperatures, water flow rate and air flow rate. Furthermore, the efficiency of the motors and VFDs has also been included in the model. The results obtained prove the importance of the VFDs in optimizing the power consumption and noise emission and will serve as input parameters for the control of the cooling towers. Unwittingly the authors have also found a way to foresee the most quiet operation points, which could be used during the night hours.

Further steps to be taken, to refine the model once the commissioning of the all new SPS primary circuit will be complete and measurements will be available for its calibration. In addition to contrasting the assumptions on which the model is based, a line of development worth investigating, in the authors' view, is the possibility of allowing the model to simulate the operation of the four cells at different working points.

REFERENCES

[1] El-Dessouky, H.T.A., Al-Haddad, A., Al-Juwayhel, F., 1997, "A Modified Analysis of Counter Flow Wet Cooling Towers," ASME Journal of Heat Transfer, Vol. 119.

[2] Goshayshi, H.R., Missenden, J.F., Tozer, R., 1999, "Cooling Tower – An Energy Conservation Resource," Applied Thermal Engineering, Vol. 19.

[3] Goshayshi, H.R., Missenden, J.F., 2000, "The Investigation of Cooling Tower Packing in Various Arrangements," Applied Thermal Engineering, Vol. 20.

[4] Bernier, M.A., Bourret, B., 1999, "Pumping Energy and Variable Frequency Drives," ASHRAE Journal, No. 12.

[5] Bernier, M.A., 1995, "Thermal Performance of Cooling Towers,", ASHRAE Journal, Vol.4.

[6] Fujita. T., Tezuka, S., 1980, "Calculations on Thermal Performance of Mechanical Draft Cooling Towers," Heat Transfer Japanese Research.

[7] Kelly, N.W., Swenson, L.K., 1956, "Comparative Performance of Cooling Tower Packing Arrangements", Chemical Engineering Progress, Vol. 52.

[8] Mohiuddin, A.K.M., Kant, K., 1995, "Knowledge Base for the Systematic Design of Wet Cooling Towers. Part I: Selection and Tower Characteristics. Part II: Fill and Other Design Parameters," Intl. Journal of Refrigeration, Vol. 19.

[9] Ibrahim, G.A., Nabban, M.B.W., Anabtawi, M.Z., 1995, "An Investigation into a Falling Film Type Cooling Tower," Intl. Journal of Refrigeration, Vol. 18.

[10] Kintener-Meyer, M., Emery, A.F., 1995, "Cost-Optimal Design for Cooling Towers," ASHRAE Journal, No. 4.

[11] Taborek, J., 1998, "Process Heat Exchangers," Hemisphere, Washington D.C.

[12] “Psychrometrics,” 1997 ASHRAE Fundamentals Handbook.

[13] “Cooling Towers,” 1997 ASHRAE Fundamentals Handbook.

[14] “Performance Curves for BA6 Cooling Towers,” 1999, SPIG Intl. SPA.

[15] “Technical Data Sheet of the SPS PHE”, 1999, APV Heat Exchanger A/S.

[16] DeWitt, D.P., Incropera, F.P., 1981, “Fundamentals of Heat and Mass Transfer,” John Wiley & Sons.

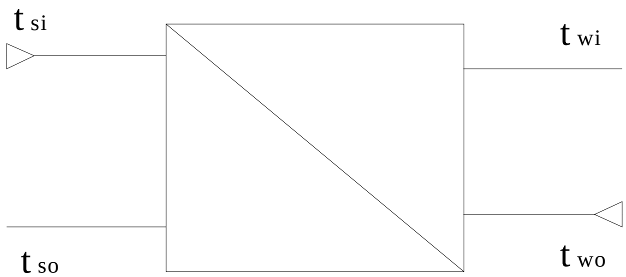
[17] Miranda Barreras, A.L., Rufes Martínez, P., 1997, “Torres de Refrigeración,” Ediciones CEAC.

[18] Kumar, H. et al., “Heat Exchanger Design Handbook”, 1983, Hemisphere Publishing Corporation.

[19] “Torres de Refrigeración,”, Centro de Estudios de la Energia, Madrid.

[20] Palenzuela, D., Moutte, V,. 1995, “Le Bruit des Tours de Refroidissement et des Aéroréfrigérants,” Promoclim n°2/95.

APPENDIX 1.



For each circuit to be solved, the heat exchanger capacity and the secondary side temperatures are fixed by the process. Therefore, once an assumption has been made on the cold water temperature, t*wo*, one can impose an outlet primary temperature somewhat lower than the secondary inlet one, let us assume that 1 K below.

$$t_{wi} = t_{si} - 1$$

$$\dot{m}_p = \frac{\dot{P}}{C_{pw} \cdot (t_{wi} - t_{wo})}$$

$$LMTD = \frac{(t_{si} - t_{wi}) - (t_{so} - t_{wo})}{\ln \frac{(t_{si} - t_{wi})}{(t_{so} - t_{wo})}}$$

$$U_{req} = \frac{\dot{P}}{(A \cdot LMTD)}$$

The equation above provides the primary channel flow rate, defined as the total primary flow rate over the number of primary channels. Once this channel flow rate known, one can calculate the channel mass velocity (kg/s m^2) from the channel cross section (S_c = W x b, width of the plates times plate spacing). The hydraulic diameter appearing in the Reynolds number being twice the plate spacing over a factor ϕ, which takes into account the ratio of developed and projected areas.

$$\dot{m}_{pc} = \frac{\dot{m}}{N_{pc}}$$

$$\dot{m}_{pc}'' = \frac{\dot{m}_{pc}}{S_c}$$

$$d_e = \frac{2 \cdot b}{\phi}$$

$$\mathrm{Re} = \frac{\dot{m}_{pc}'' \cdot d_e}{\eta}$$

For chevron-trough plates, a series of correlations due to Taborek have been employed, after adjusting (linearly) to the performance of the new set of equations in two points (Re = 1,000 and Re = 40,000). These correlations, which have the form of the equation below, allow to obtain the film coefficient.

$$\frac{Nu}{\mathrm{Pr}^{0.33}(\frac{\eta}{\eta_w})^{0.17}} = \exp[k1 + k2 \cdot \log(\mathrm{Re})]$$

$$h = \frac{Nu \cdot \lambda}{d_e}$$

$$U_{calc} = [\frac{1}{h_p} + \frac{1}{h_s} + \frac{th}{\lambda_{plate}}]^{-1}$$

Once the calculation of the heat exchangers completed, one proceeds to the addition of the total flow rate in the SPS primary loop, the calculation of the return primary water temperature (averaged with each circuit's flow rate) and the calculation of the primary loop pumping power. The latter is evaluated on the basis of a calculated pressure drop, composed of three terms:

- The pressure drop long the pipeline, depending on the primary flow rate,
- The pressure drop inside the cooling stations, which is kept constant at 4 bar,
- The level differences between elements, also constant.

The final step in the process is the calculation of the cooling towers. To do that the starting point is the basic water-to-air heat transfer equation from a saturated surface to an air stream. This equation reads:

$$\dot{Q} = h_{\mathrm{int}} A \frac{(h_w - h_a)}{C_{p-air}}$$

The assumption that the moist air at the interface with the water droplets is at the same temperature as the bulk water has been introduced in the equation above, allowing the equation to be written in terms of h_w instead of $h_{a.}(t_{int})$ This assumption simplifies the problem greatly as it eliminates the need to solve the conduction problem from the water droplet core (at bulk temperature) to its surface and then into the water-air interface. One should observe that the h_{int} value used above refers to a film coefficient (dimensions of kJ/m^2 K) and must not be confused with the saturated and moist air enthalpies, h_w and h_a respectively.

Posing the heat balance in the cooling tower and rearranging the equations (subscripts *i* and *o* in the equations below referring to inlet and outlet conditions respectively), one obtains:

$$LC_{pw}(t_{wi} - t_{wo}) = G(h_{ao} - h_{ai})$$

$$d\dot{Q} = Gdh_a = -LC_{pw}dt_w$$

$$d\dot{Q} = \frac{h_{\text{int}}}{C_{p-air}} dA(h_w - h_a)$$

$$Le = \frac{h_{\text{int}}}{KC_{p-air}} = 1$$

$$d\dot{Q} = KadV(h_w - h_a)$$

$$Gdh_a = KadV(h_w - h_a)$$

$$\frac{KaV}{L} = C_{pw} \int_{two}^{twi} \frac{dt_w}{(h_w - h_a)}$$

Once the cooling tower solved, the thermodynamic characteristic and the operation liquid to air area are known. Using the latter ratio one can come back to the water-air balance and obtain the enthalpy of the air leaving the tower.

$$h_{ao} = h_{ai} + \frac{L}{G} C_{pw}(t_{wi} - t_{wo})$$

One can write now the total heat exchange equations between air and liquid at bulk temperature (on the assumption there is no interface resistance, as mentioned above), and obtain:

$$d\dot{Q} = Gdh_a$$

$$d\dot{Q} = K(h_w - h_a)dA$$

Dividing both equations,

$$\frac{G dh_a}{K(h_w - h_a)dA} = 1$$

A similar scheme for the sensible heat exchange yields:

$$d\dot{Q}_s = GC_{p-air}dt_a$$

$$d\dot{Q}_s = h_{\text{int}}(t_w - t_a)dA$$

Dividing again one obtains:

$$\frac{GC_{p-air}dt_a}{h_{\text{int}}(t_w - t_a)dA} = 1$$

When both divisions are equated one gets:

$$\frac{G dh_a}{K(h_w - h_a)dA} = \frac{GC_{p-air}dt_a}{h_{\text{int}}(t_w - t_a)dA}$$

And applying now the assumption that the Lewis number is equal to 1,

$$K = \frac{h_{\text{int}}}{C_{p-air}}$$

$$\frac{dh_a}{dt_a} = \frac{h_w - h_a}{t_w - t_a}$$

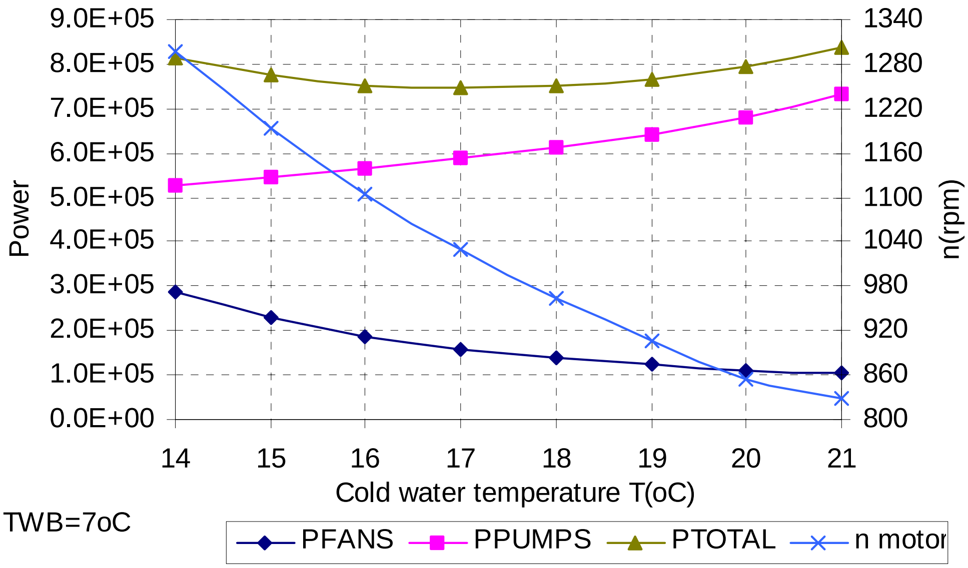
The last equation allows the calculation of the temperature of the air leaving the tower. This suffices to fully determine the condition of the air, as the enthalpy is already known. It is possible to derive, in the same fashion that it has been done for the sensible heat equation, a similar relationship for the humidity.

The process is concluded by calculating the power absorbed by the fans, taking into account the efficiencies of the motors and their VFD.

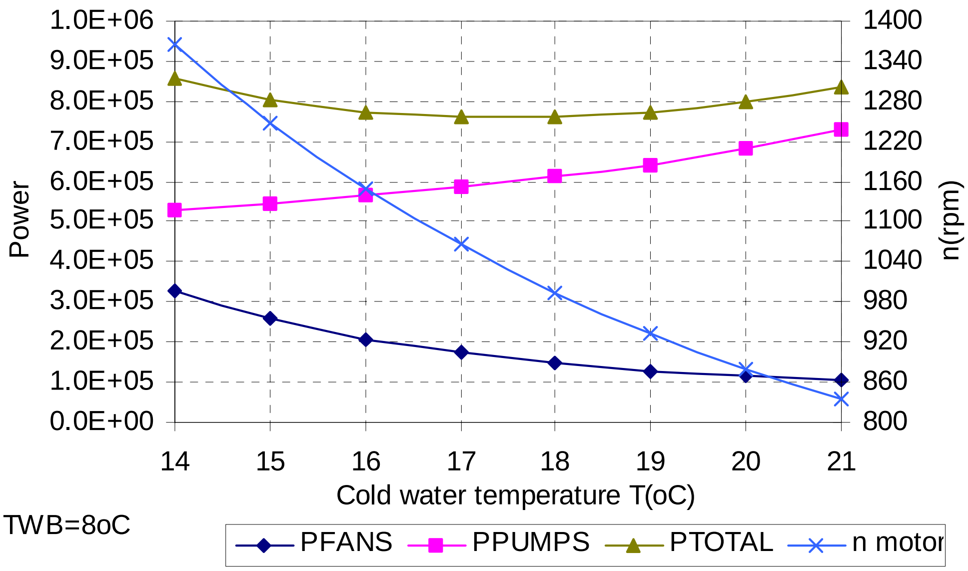
APPENDIX 2.

FOR DIFFERENT WET BULB TEMPERATURES TWB:

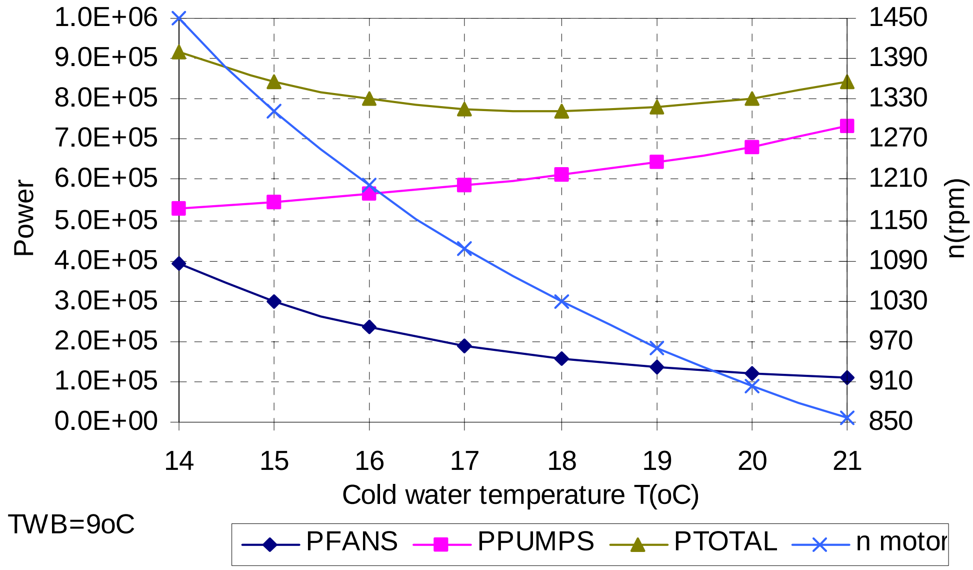
POWER CONSUMPTION FOR FANS, PUMPS AND TOTAL (W), AND SPEED OF THE FANS' MOTORS (rpm) AS A FUNCTION OF THE COLD WATER TEMPERATURE ACHIEVED IN THE COOLING TOWER

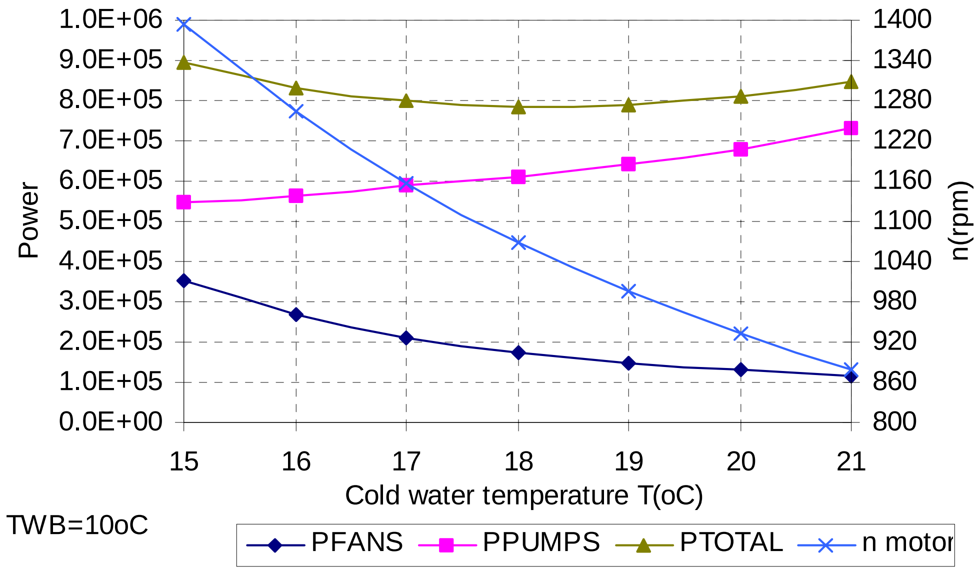


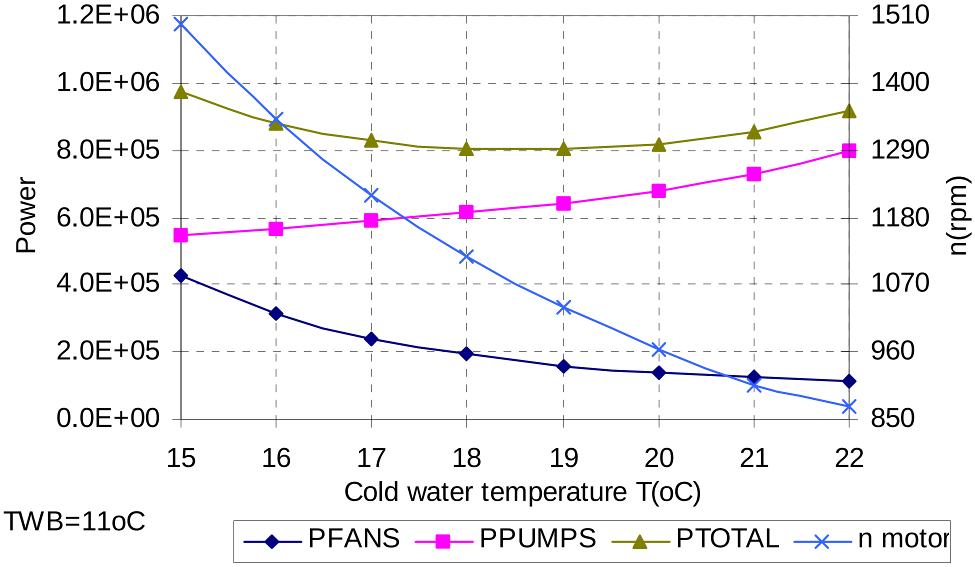
TWB=7oC



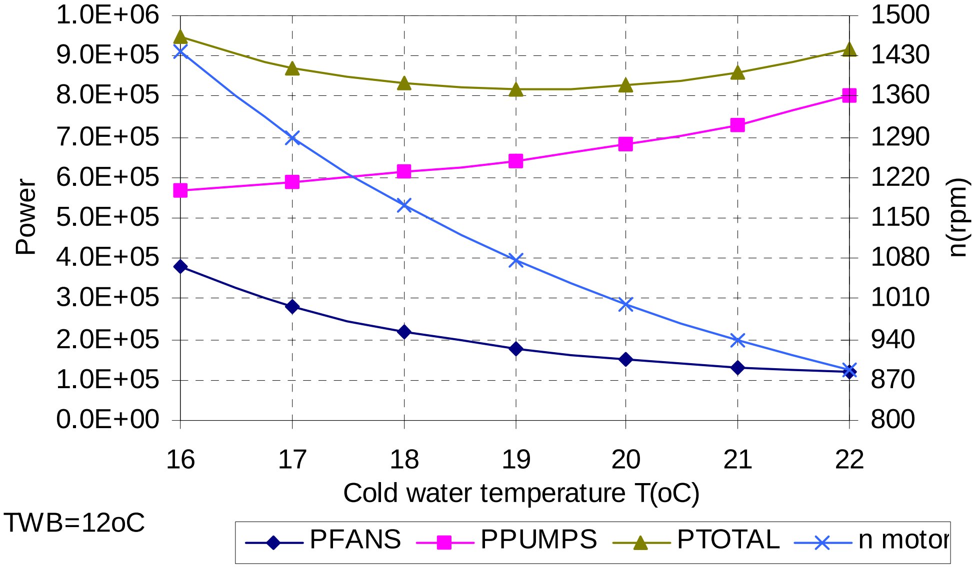
TWB=8oC


Power
n(rpm)
1.0E+06
9.0E+05
8.0E+05
7.0E+05
6.0E+05
5.0E+05
4.0E+05
3.0E+05
2.0E+05
1.0E+05
0.0E+00
1450
1390
1330
1270
1210
1150
1090
1030
970
910
850
14 15 16 17 18 19 20 21
Cold water temperature T(oC)
TWB=9oC
PFANS
PPUMPS
PTOTAL
n motor

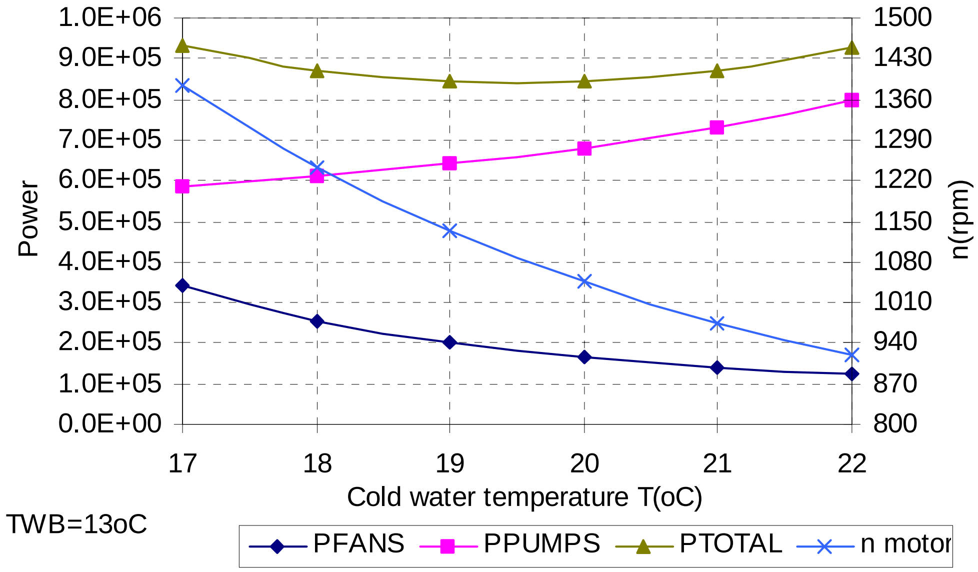

Power
n(rpm)
1.0E+06
9.0E+05
8.0E+05
7.0E+05
6.0E+05
5.0E+05
4.0E+05
3.0E+05
2.0E+05
1.0E+05
0.0E+00
1400
1340
1280
1220
1160
1100
1040
980
920
860
800
15 16 17 18 19 20 21
Cold water temperature T(oC)
TWB=10oC
PFANS
PPUMPS
PTOTAL
n motor

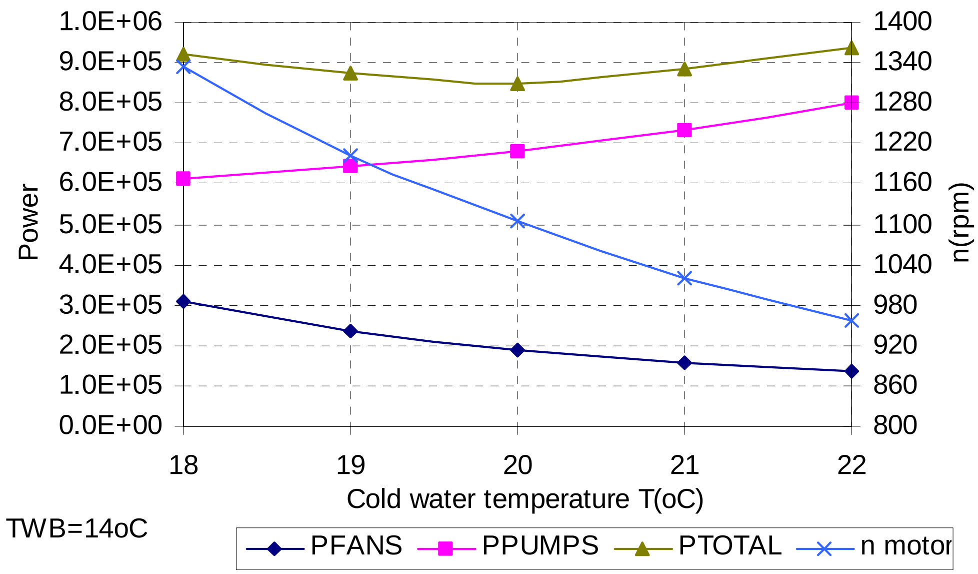


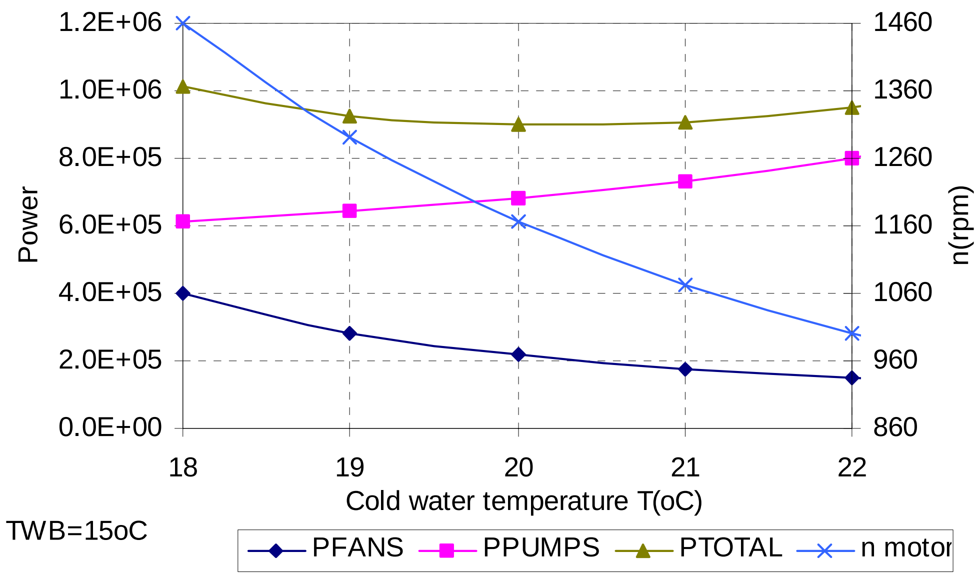
TWB=11oC

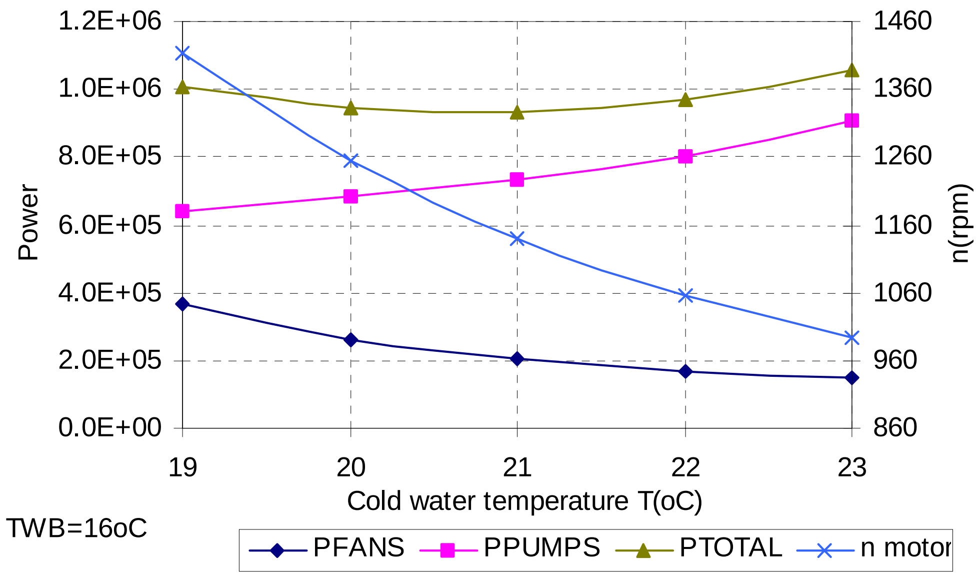


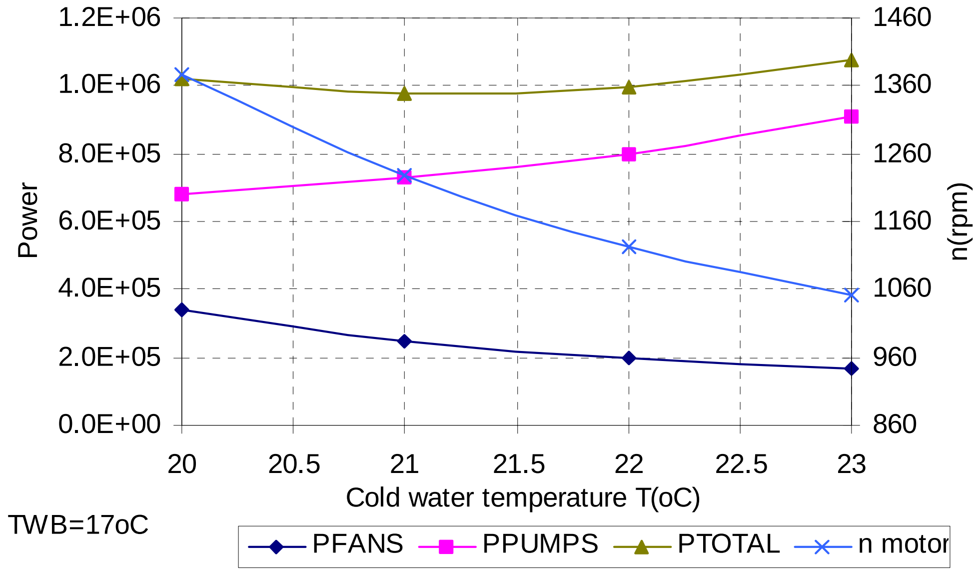
TWB=12oC

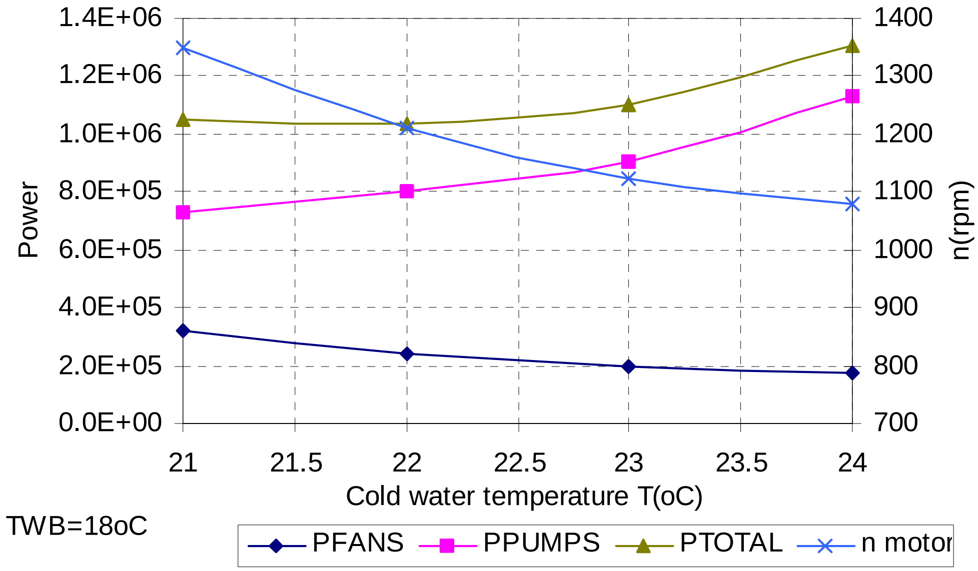

1.0E+06
9.0E+05
8.0E+05
7.0E+05
6.0E+05
5.0E+05
4.0E+05
3.0E+05
2.0E+05
1.0E+05
0.0E+00
Power
1500
1430
1360
1290
1220
1150
1080
1010
940
870
800
n(rpm)
17
18
19
20
21
22
Cold water temperature T(oC)
TWB=13oC
PFANS
PPUMPS
PTOTAL
n motor

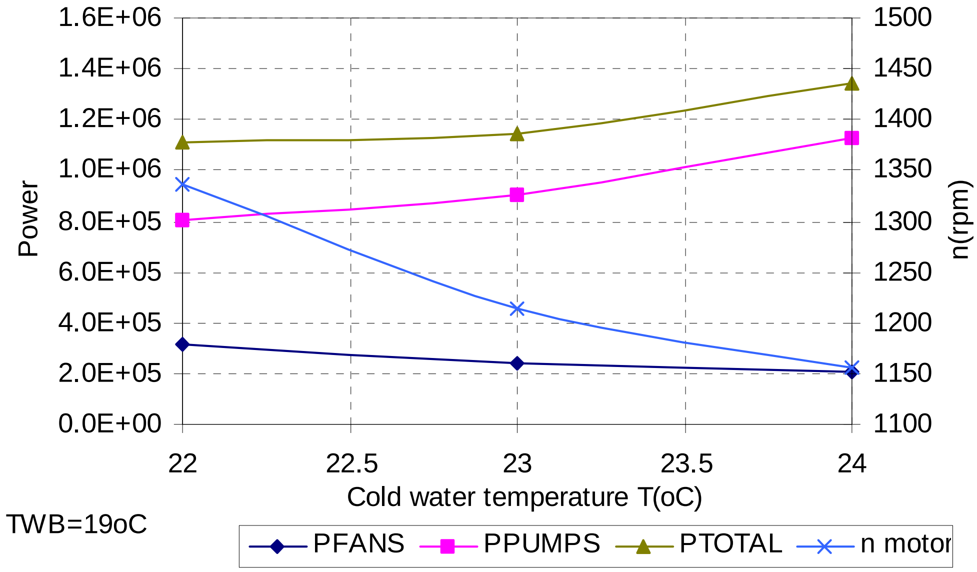

1.0E+06
9.0E+05
8.0E+05
7.0E+05
6.0E+05
5.0E+05
4.0E+05
3.0E+05
2.0E+05
1.0E+05
0.0E+00
Power
1400
1340
1280
1220
1160
1100
1040
980
920
860
800
n(rpm)
18
19
20
21
22
Cold water temperature T(oC)
TWB=14oC
PFANS
PPUMPS
PTOTAL
n motor

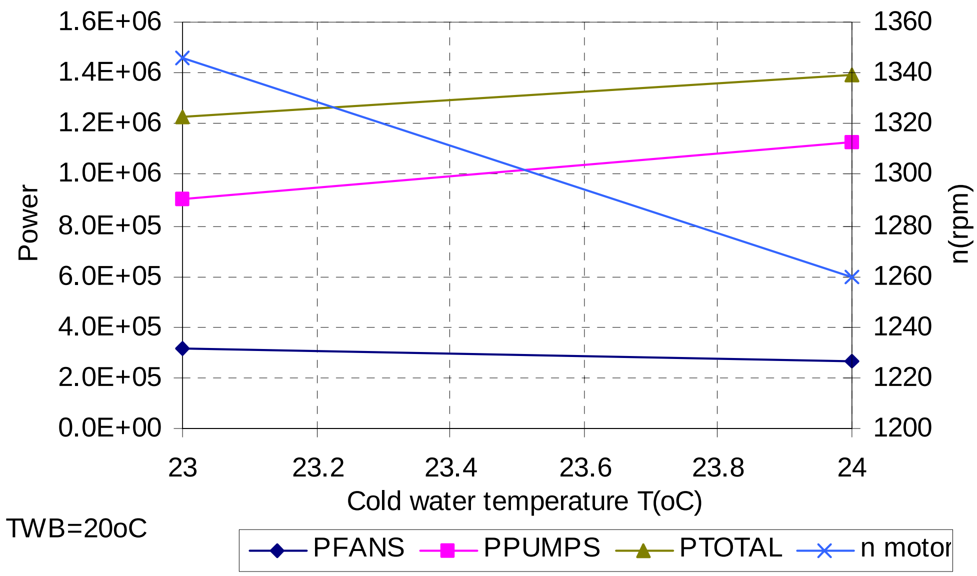

1.2E+06
1.0E+06
8.0E+05
6.0E+05
4.0E+05
2.0E+05
0.0E+00
Power
1460
1360
1260
1160
1060
960
860
n(rpm)
18
19
20
21
22
Cold water temperature T(oC)
TWB=15oC
PFANS
PPUMPS
PTOTAL
n motor

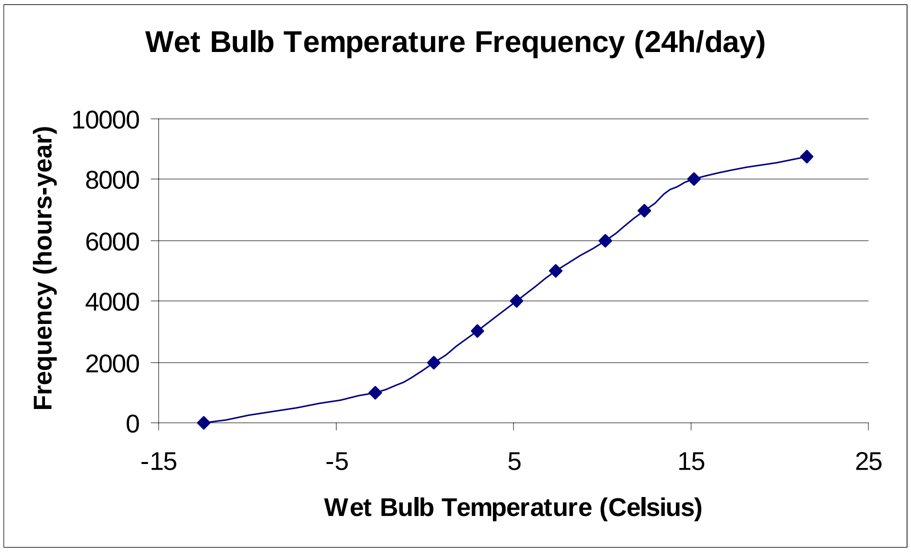

1.2E+06
1.0E+06
8.0E+05
6.0E+05
4.0E+05
2.0E+05
0.0E+00
Power
1460
1360
1260
1160
1060
960
860
n(rpm)
19
20
21
22
23
Cold water temperature T(oC)
TWB=16oC
PFANS
PPUMPS
PTOTAL
n motor


1.2E+06
1.0E+06
8.0E+05
6.0E+05
4.0E+05
2.0E+05
0.0E+00
Power
1460
1360
1260
1160
1060
960
860
n(rpm)
20
20.5
21
21.5
22
22.5
23
Cold water temperature T(oC)
TWB=17oC
PFANS
PPUMPS
PTOTAL
n motor


1.4E+06
1.2E+06
1.0E+06
8.0E+05
6.0E+05
4.0E+05
2.0E+05
0.0E+00
Power
1400
1300
1200
1100
1000
900
800
700
n(rpm)
21
21.5
22
22.5
23
23.5
24
Cold water temperature T(oC)
TWB=18oC
PFANS
PPUMPS
PTOTAL
n motor


1.6E+06
1.4E+06
1.2E+06
1.0E+06
8.0E+05
6.0E+05
4.0E+05
2.0E+05
0.0E+00
Power
1500
1450
1400
1350
1300
1250
1200
1150
1100
n(rpm)
22
22.5
23
23.5
24
Cold water temperature T(oC)
TWB=19oC
PFANS
PPUMPS
PTOTAL
n motor


1.6E+06
1.4E+06
1.2E+06
1.0E+06
8.0E+05
6.0E+05
4.0E+05
2.0E+05
0.0E+00
Power
1360
1340
1320
1300
1280
1260
1240
1220
1200
n(rpm)
23
23.2
23.4
23.6
23.8
24
Cold water temperature T(oC)
TWB=20oC
PFANS
PPUMPS
PTOTAL
n motor


Wet Bulb Temperature Frequency (24h/day)
10000
8000
6000
4000
2000
0
Frequency (hours-year)
-15
-5
5
15
25
Wet Bulb Temperature (Celsius)