Toolbox 12
COOLING TOWERS

1. **Mechanical Draft Water Cooling Tower Designs**. These towers usually consist of a vertical shell made of plastic or metal. Water is distributed near the top and falls to the collecting basin. It passes through the air that flows from the bottom to the top by means of forced or induced draft fans. Only cooling towers with induced fans are presented in Fig. 12.1. The inside of a mechanical-draft tower may be filled with a spray of water droplets from nozzles or packed with plastic or ceramic filling down which water cascades from the top to the bottom. In many cases, a combination of spray and plastic or ceramic-filling is used.

Before being discharged into the atmosphere, the water-laden exhaust air passes through a drift (spray) eliminator which removes the water droplets as they are carried along.

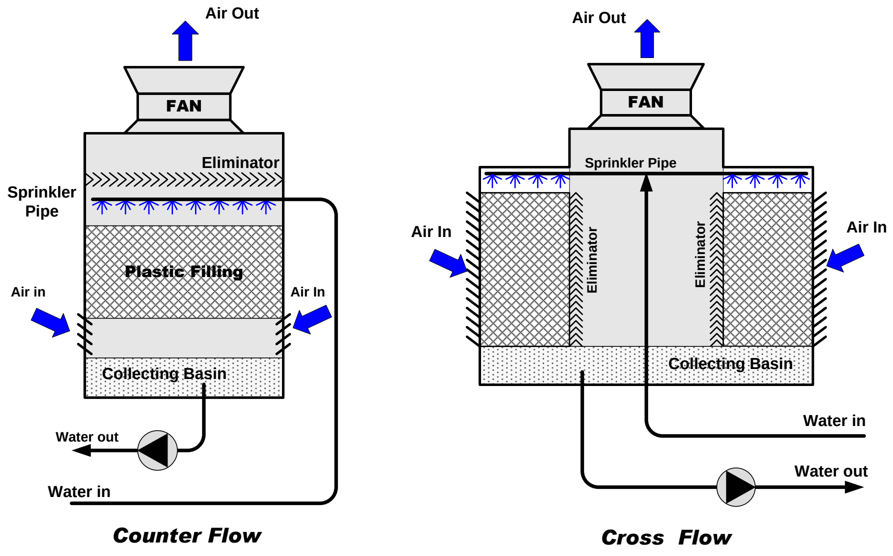

Figure 12.1: Scheme of Counter Flow and Cross Flow Cooling Towers

Counter flow cooling towers have, theoretically, a uniform exit air wet bulb temperature. Cross flow towers exhibit a large variation of exit air wet bulb temperature, which is responsible for further evaporation loss. In addition, cross flow towers require more airflow in order to meet the same cooling capacity and overall evaporation losses are very slightly higher.

Cooling towers used for air-conditioning systems are often located on the top of buildings, and the cross flow cooling tower usually has a lower profile, which lends itself better to architectural treatment.

2. **Principles for Operation Analysis**. A cooling tower cools water by contacting it with the air and evaporating some of the water. One or more axial or centrifugal fans move the air vertically up or horizontally through the tower. Spraying the water through nozzles or splashing the water down the tower from one baffle to another provides a large surface area of water.

The performance of cooling towers is often expressed in terms of *range* and *approach* (Fig. 12.2). ***Cooling range*** is the temperature difference between the hot water coming to the cooling tower and the temperature of the cold water leaving the tower. ***Approach*** is the temperature difference between the temperature of the cold water leaving the tower and the surrounding air wet bulb temperature. Both the range and approach must be greater than zero in normal cooling tower operation. Most common values are approximately 5 ^{o}C for both. Designers use the parameters and geometry of the cooling tower in order to achieve these temperature differences in practice. That these and other parameters are traditionally present in the designer’s practice, does not imply that they are also the best. Their impact is also very important for the size of a cooling tower and its price. Unfortunately, economic parameters change over time. There is no doubt that an optimum exists but it is very difficult to change it as it requires frequent alteration in cooling tower manufacturing technology. This is, however, very expensive.

In the cooling tower, the transfer takes place from water to unsaturated air. There are two driving forces for the transfer:

- the difference in dry bulb temperatures, and
- the difference in vapor pressures between water surface and unsaturated air.

These two driving forces when combined form the ***enthalpy potential***.

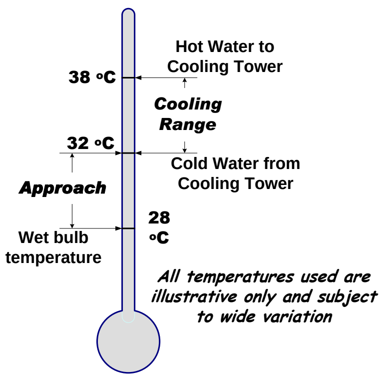

Figure 12.2: Schematic Definition Explains the Terms ***Cooling Range*** *and* ***Approach***

Above the wetted surface (Fig. 12.3), there is a film of air through which temperature and vapor pressure gradients exist. In the immediate vicinity of the wetted surface, the air is saturated at $t_{a,s}$, $x_{a,s}$ and $h_{a,s}$. The rate of diffusion of water vapor through the air film will be equal to the rate of condensation or evaporation of water on the wetted surface. Because of the condensation or evaporation process which occurs over the wetted surface, there is a difference in enthalpy between unsaturated and saturated air ($h - h_{a,s}$). This quantity is the *Enthalpy Potential*, and it is the driving force for the total energy transfer process between water and unsaturated air.

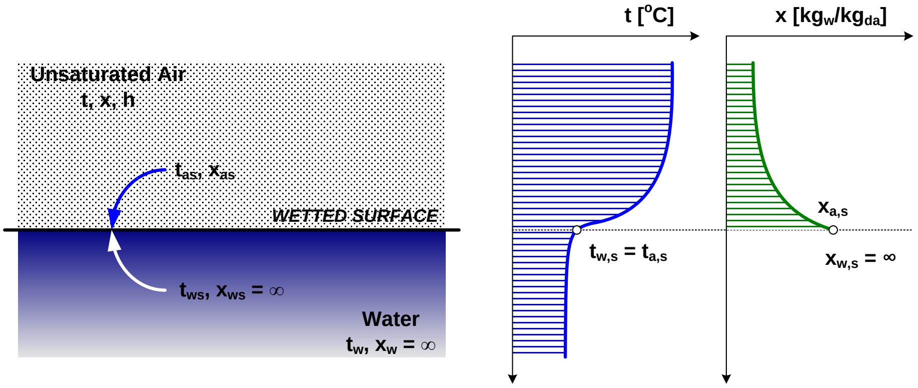

Figure 12.3: One of Possible Temperature and Concentration Profiles with Unsaturated Air over Wetted Surface

The cooling tower calculation is based on the most generally accepted theory of the cooling tower heat-transfer process developed by Merkel. This analysis is based upon ***enthalpy potential difference*** as the driving force.

It is assumed that a film of air surrounds each particle of water, and the enthalpy difference between the film and surrounding air provides the driving force for the cooling process. In its integrated form the Merkel equation is:

$$4.19 \cdot L \int_{t_{out}}^{t_{in}} \frac{dt}{h_i - h_a} = \int_0^A \frac{h_c \cdot dA}{c_{pm}} = \frac{h_c \cdot A}{c_{pm}} \tag{12.1}$$

Where:

t_{in}	=	water temperature entering the cooling tower, °C;
t_{out}	=	water temperature leaving the cooling tower, °C;
h_c	=	convection coefficient, kW/(m^2 K);
h_i	=	enthalpy of saturated air at water temperature, kJ/(kg of dry air);
h_a	=	enthalpy of air, kJ/(kg of dry air);
c_{pm}	=	mean specific heat of moist air, kJ/(kg K);
A	=	total area of wetted surface includes the surface area of water drops as well as wetted slats or other fill material, m^2;
L	=	water mass flow rate, kg/s.
4.19	=	mean ratio between specific heats of water and air

The airflow rate (G) is not shown explicitly in the equation above, but implicitly it is in the convection coefficient.

This method assumes that the temperature of the surface of the water droplets prevails throughout the droplet. Actually, the interior of the droplet has a higher temperature than that of the surface, and heat flows by conduction to the surface where the heat- and mass-transfer processes occurs.

Cooling tower designer and manufacturers often use the Number of Transfer Units (NTU) to refer to the term $h_c \cdot A / c_{pm}$. The higher the value of NTU, the closer the temperature of the water leaving the cooling tower will come to the wet bulb temperature of the entering air. The NTU is one of the performance indicators for cooling towers.

Figure 4 illustrates the water and air relationships and the driving potential that exists in the cooling tower, where air flows in a parallel but opposite direction to the water flow. The water operating line is shown by the line AB and is fixed to the inlet and outlet tower water temperature. The air operating line begin at C, vertically below B (in h - t diagram, Fig. 4(a)) and at the point having enthalpy corresponding to that of the entering wet-bulb temperature.

The slope of air–operating lines is 4.19 · L/G.

Heat capacity is the amount of heat thrown away by the cooling tower. It is equal to the mass flow rate of water circulated times the specific heat of water times the cooling range.

It is very important to stress that:

– The change in wet bulb temperature (due to atmospheric conditions) will not change the cooling tower performance indicator NTU.
– The change in the cooling range will not change the cooling tower indicator NTU.
– Only the change of water and air flow will change the performance indicator NTU.

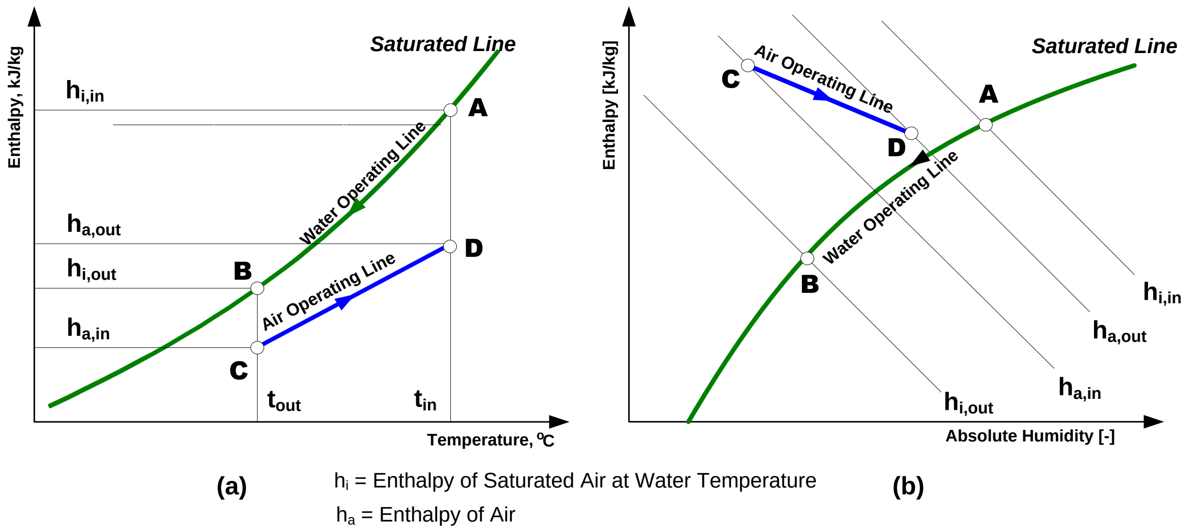

Figure 12.4: Water and Air Operating Lines in Enthalpy-Temperature and Enthalpy-Absolute Humidity Diagrams

3. For the practical analysis of cooling tower operation, the following losses influencing the operation have to be defined:

- Blow-Down
- Evaporating Loss
- Drift Loss
- Make-Up

Blow-Down is the continuous or intermittent wasting of a small fraction of circulating water in order to prevent concentration of chemicals in water. The purpose of blow-down in water cooling apparatus is to reduce soluble solids or hardness. This reduces the scale-forming tendencies of water. The blow down loss is as follows:

$$\text{Blow-Down Loss} = \text{Evaporation Loss} \cdot \frac{TDS_{\text{MakeUp Water}}}{TDS_{\text{Circulating Water}} - TDS_{\text{MakeUp Water}}} \qquad (12.2)$$

where:

$TDS_{\text{MakeUp Water}}$ = Total Dissolved Solids in Make-Up Water (mg/l or ppm)
$TDS_{\text{Circulating Water}}$ = Total Dissolved Solids in Circulating Water (mg/l or ppm)

Solids may be present in suspension and/or in solution and they may be divided into organic and inorganic matters. Total dissolved solids (TDS) are caused by soluble materials whereas suspended solids (SS) are discrete particles which can be measured by filtering the sample through fine paper.

The electrical conductivity of a solution depends on the quantity of dissolved salts present and for diluted solutions it is approximately proportional to the TDS content:

$$K = \frac{\text{Conductivity}\,[S/m]}{\text{TDS}\,[mg/l]} \tag{12.3}$$

Knowing the approximate value of K for particular water, the measurement of conductivity provides a rapid indication of the TDS content.

Blow-down systems can be instantaneous or continuous processes and may be manual, semi-automatic or fully automatic.

If there are no manufacturer's recommendations on cooling water quality, the following table can be used for the definition of water quality. The potential problems that can appear in the cooling system if the prescribed water quality is not achieved are also given.

Table 12.1: Cooling Water Quality

Item		**Unit**	**Make Up Water**	**System Water**	**PROBLEM**	
					Corrosion	**Scale**
pH (at 25 °C)		-	6.0–8.0	6.0–8.0	♦	♦
Conductivity (at 25 °C)		µS/cm	< 200	< 500	♦	♦
Hardness	$CaCO_3$	ppm	< 50	< 200		♦
M-Alkalinity	$CaCO_3$	ppm	< 50	< 100		♦
Chlorine Ion	Cl^-	ppm	< 50	< 200	♦	
Sulfuric Acid Ion	SO_4^{2-}	ppm	< 50	< 200	♦	
Ferrous Substance	Fe	ppm	< 0.3	< 1.0	♦	♦
Silica	SiO_2	ppm	< 30	< 50		♦
Sulfur Ion	S^{2-}	ppm	0	0	♦	
Ammonium Ion	NH_4^+	ppm	0	0	♦	

Evaporating Loss. The quantity of evaporating water (named **evaporating loss**) is very small in comparison with the water flow rate (L), and as the air flow rate (G) is normally equal or slightly less than the water flow rate, this means that both L and G can be assumed to be constant.

The evaporation loss can be estimated as follows:

$$\frac{\text{Evaporation Loss [kg/s]}}{\text{Flow Rate of Water at Tower Inlet [kg/s]}} = 0.00153 \cdot \text{Cooling Range } [^\circ C] \tag{12.4}$$

Drift Loss is the small amount of un-evaporated water lost from an atmospheric water-cooling apparatus in the form of a mist of fine droplets. It is water entrained by circulating air. Drift is water loss independent of water lost by evaporation. The drift loss, unlike evaporation loss, can be reduced by good design.

Drift losses can be estimated to be between 0.1 % and 0.2 % of **make up water** supply.

Make Up is the water required to replace water that is lost by evaporation, drift, blow-down and small leaks.

4. **Ways to Improve Cooling Tower Performance**. The main purpose of this text is to explain the procedure that has to be performed in order to define the performance indicators of an energy system containing one or more cooling towers. However, the first step in any technical and, later, economical analysis has to be the estimation of the technical performance of present equipment and an analysis of the opportunities to improve their efficiency. This assumes the analysis of the demand and utility side. In the case of a cooling system, the demand side defines the load, but it has to be known whether the necessary energy resources are used in an optimal way. For this purpose, the boundaries of the energy process analyzed will be defined and all input values (energy flows) will be treated as given data, but it has to be analyzed by some other analysis using the same or similar energy efficiency criteria.

As technology is changing daily, it will be very useful to estimate if it is possible to increase the heat capacity of the existing cooling system with or without some reasonable investment, which will ultimately affect energy consumption. Actually, the first energy efficiency measure is to increase the efficiency of the existing plant by improving the maintenance procedures and by implementing new proven technology in the current energy system. Unfortunately, in many factories technical staff hesitate to change to new technologies. Although they do not provide any technical parameters for claiming that the existing technologies are successful, they fear the introduction of new technologies because of lack of knowledge.

When the cooling tower load is exceeded, it is possible to install a second unit, install a larger unit, or increase capacity of the existing cooling tower. The third option may give new life and efficiency without requiring new, costly equipment.

If there is a need for more cooling tower capacity, but there is no room for a larger unit, the first question is how to get more capacity out of the present cooling tower.

Cooling tower owners are often in trouble since many units serving power, process plants, or air-conditioning systems are used for serving increased cooling loads as the plants have expanded. However, many towers have not been able to keep up with actual plant operating conditions since startup. This is not surprising when you realize that this type of equipment is often selected and installed solely on a low first cost basis. As might be expected, a unit thus selected is very likely not to be the most efficient or is even sometimes impractical for the job.

5. **Increasing Capacity**. Perhaps half of the field-erected and a few of the prefabricated towers installed today can be improved to some degree. This is largely because of new technologies allowing for more efficient components, and new materials that can be built into harder working, more rugged and, therefore, more reliable packages.

Before trying to upgrade capacity, a complete study must be made of the cooling requirements. Then, a careful inspection and analysis of the existing tower components is needed. This can be tied together with respect to the existing tower size, pressure drop, airflow characteristics, drift loss, fan performance curves, gear selection, service factors and installation problems.

The possible improvements of cooling tower capacity are presented in Table 12.2. Generally, savings can be high bearing in mind that capacity can be increased without enlarging the physical size of the currently occupied space, the piping or making any electrical alterations to the plant. In all of the presented cooling tower cases, it is assumed that water flow rate is increased. However, in some cases the in and out water temperatures are unchanged, and in some of them water temperature is decreased.

Table 12.2: Examples of Possible Improvements

	IMPROVEMENTS	**As Designed**					**After Modification**					**Improvements**
		Water Flow Rate	Water Temperature		Wet Bulb	Cooling Capacity	Water Flow Rate	Water Temperature		Wet Bulb	Cooling Capacity	
			In	**Out**				**In**	**Out**			
		m^3/m	°C	°C	°C	kW	m^3/m	°C	°C	°C	kW	
1	New spray distribution system, new 60 degrees drift eliminators and film packing	49.96	48.9	32.2	26.7	3486	75.70	48.9	32.2	26.7	5281	**51.5 %**
2	Larger fan motor and gearbox, larger spray nozzles and film packing	4.73	51.7	23.9	18.4	550	7.00	51.7	23.9	18.4	814	**48.0 %**
3	New spray distribution system, new film packing	694.93	53.8	34.6	23.9	55917	722.94	50.3	32.2	25.0	54640	**-2.3 % colder water**
4	Larger fan motor and gear box, nozzles with larger orifices and film packing	11.73	40.6	29.4	25.0	546	14.69	40.6	29.4	25.0	683	**25.2 %**

5	Fan motors increased from 29.4 t0 44.1 kW and larger gearbox changed from high-pressure spray to low-pressure down spray type and put in additional splash-type fill	17.03	43.3	29.4	25.6	990	24.60	39.4	29.4	25.6	1030	**4.0 % colder water**
6	Increased flume height of tower by 5 m, new spray distribution system, larger motor, more splash-type fill and tip seals placed in fan stack	49.96	48.9	32.2	26.7	3486	84.03	48.9	32.2	26.7	5862	**68.2 %**

A cooling tower specialist should carry out studies and make recommendations. Once the findings are acted upon in order to achieve practical results, substantial savings can be achieved and often save on cost for a new unit.

6. The variables that influence cooling tower capacity and which have to be considered in order to increase capacity are:

- Fill Configuration
- Distribution System
- Drift Elimination
- Mechanical Equipment
- Fan Stack
- Partitioning

Fill Configuration. Cooling tower packing has improved greatly over the years. When older towers were originally installed, tower manufacturers designed the units to perform according to specified conditions by using the type of packing that they considered to be the most economical for their manufacturing facilities to produce. The design was, of course, consistent with the performance and test data available at that time. Since then, improvements in design have come a long way, and more field test data is currently available on different fill configurations.

For example, most old counter flow towers have splash type packing installed. Rows of splash decks are usually spaced about 600 mm apart in the tower. The theory is to break up the water into drops as it cascades through the tower. And each drop surface is increased by the continuous interception of its fall by the splash decks, thus exposing a fresh evaporating surface on every new drop.

Appraisal of the fill arrangement, and its quantity, by an experienced cooling tower specialist will establish if more fill can be added in order to provide increased capacity. In this case, film surface packing, or special packing that combines both film surface as well as additional splash surface is installed. Some fill designs will increase the surface area and also the time that the water surface is exposed to air. As a result, the rate of exposure may be high enough to permit the reduction of air required. Therefore, film packing may be added without increasing fan horsepower requirements.

The percentage of capacity improvement that results from adding film surface-type fill depends on the severity of the duty, the performance level and the height of the tower. For example, by adding substantial quantities of film packing for a very low tower that is subjected to a very severe duty, 40 to 50 % capacity improvement can be obtained. On the other hand, if a very high tower is used for a very easy duty, the performance can be adversely affected when film-type packing is installed. In this instance, airflow rate is more critical to performance than the performance level itself.

However, for a reasonable performance level, a 20 % increase in capacity can be achieved for a tower of average design. An increase of 20 % in capacity may be equated to a 20 % increase in the water flow rate at the same temperature level, or an approximately 20 % decrease in the approach to the wet bulb at the same flow rate and heat load.

Distribution System. Some older towers have open flume or trough distribution systems. This type of system, especially in multi-cell installations, is hard to balance. The problem is then compounded when water loadings change in the process. Flooding or dry spots also detract from the tower's effectiveness.

Sometimes the gravity feed clogs at the downspouts. Each downspout has a diffusion deck, or splash plate, under it. If these are broken or out of line, then the effective cooling volume is reduced. Here, the change to a positive pressure spray-type distribution system reduces balancing problems. An advanced design header-lateral spray system ensures a good water pattern over the entire fill area and permits the passage of more air through this area of the tower.

Again, an analysis of the tower in operation reveals possible areas for improvement. Water spray may not fully cover upper fill layers. Occasionally, water is halfway down through the fill before it is evenly distributed across the entire plan area of the tower.

The obvious solution to this problem is to modify the nozzles so that the spray pattern is altered, or corrected. This simple improvement often adds capacity to an existing tower.

Some older, high-pressure up spray towers have been converted to low-pressure down spray units with improved results. This process entails raising the distribution system level, which permits additional rows of fill decks to be installed. While this modification may increase capacity for as much as 12 to 15 %, there will also be an overall, small decrease in pumping head.

Drift Eliminators. Many older towers have extra heavy drift eliminators. Others have close-spaced eliminator blades of a 45^o angle. Both designs are found to be too conservative for some requirements. One reason is that these designs naturally restrict the air flow. By replacing them with staggered drift eliminator blades at a 60^o instead of a 45^o angle, more air is admitted through the tower and the result is additional capacity.

Drift eliminator modifications are usually made when the drift eliminator requires replacement. A counter flow tower improved by this simple change can increase capacity from 4 to 5 %. On a 5 ^{o}C approach tower, this is equivalent to almost 0.5 ^{o}C colder water.

In some cases, an extra pass of drift eliminators can be installed in order to enable the change of fan or gear to draw the maximum air flow rate without causing a drift loss problem.

Mechanical Equipment. If additional capacity is expected from the existing cooling tower, more air movement is generally needed. But many installations already operate at the maximum rated power. In such cases, additional readings have to be taken at the motor leads in order to determine how close the motor is running to full load or nameplate amperage. If there is room for additional load, the fan blade-angle has to be changed. Since power varies as the cube of airflow rate, nothing much can be done without a larger motor.

Conversely, an increase in the air rate varies as the cube root of the increase in power. But a substantial increase of power may lead only to a relatively small increase in capacity, as summarized in Table 12.3.

Table 12.3: Motor Power versus Flow Rate

Motor Power Change [kW]	Motor Power Change [%]	Approximate Flow Rate Change [%]
25 to 30	20.0	6.0
30 to 40	33.3	10.0
40 to 50	25.0	7.5
50 to 60	20.0	6.0
60 to 75	25.0	7.5
75 to 100	33.3	10.0

If the unit is installed with the fan pitch angle at the maximum efficiency level, a change in fan speed may be appropriate. Changing the gear ratio or motor achieves that.

Major modifications include increasing the fan size. But this is rarely done, as increasing the flow rate by increasing fan speeds strains the limits of the tower from another viewpoint as a velocity that is too high through the tower and drift eliminators can cause problems from excess carryover.

Full effectiveness can be realized from a larger motor with an increased pitch angle or fan speed. Here, capacity improvement of as much as 10 % can be achieved.

Fan Stack. The importance of fan stack design for top performance cannot be overemphasized. It is not practical to change or increase the motor size on many existing units. One reason is that any change of the electrical service to the cooling tower site might be unreasonably costly. Here, the installation of a parabolic ventury-type fan stack may be the answer because, with this velocity recovery-type fan stack, the fan is capable of delivering 6 to 7 % more air through the tower with the same motor. 7 % more air means 7 % more tower capacity.

Partitioning. Some larger multi-cell towers are built for operation only at design load. Some of these units have two fans per cell and there is no partition in the plenum area. If one fan is shut down for repairs, or if the tower is operated with only one fan per cell for any reason, the mechanical draft feature will be rendered practically ineffective. This is because the operating unit will then be drawing most of its air through the adjacent fan opening, thus bypassing the fill area, which is extremely wasteful.

But even if plenum areas are partitioned properly between fan cells (in counter flow towers) the transverse partition should be extended down to the top of the louver level. Only then can the effective counter flow principle be realized. If the unit is installed in a wide-open area, a longitudinal partition has to be installed in order to prevent blow through when high winds hit at 90^o to the longitudinal axis of the tower. When the plant operates at partial load, proper partitions between fans and cells will greatly improve operating efficiency.

7. **Efficiency Obtained by Proper Operation and Maintenance**. The cooling tower will be the focus of more and more attention as increased effort is directed to the more efficient use and conservation of water. The operating and maintenance manual for the cooling tower should be studied by operating personnel in order to make sure that peak efficiency of the equipment is realized.

Consultations with a trained cooling tower engineer will enable the proper assessment of tower components. *The distribution system (air water pattern) is the key for the tower's operating efficiency* and by an experienced survey the small difference needed to maintain the design capacity level may be found. The dividends obtained by following such a procedure will more than justify the comparatively small amount of time invested.

8. **Energy Audit**. The main purpose of an energy audit is to find opportunities to reduce energy consumption and reduce energy costs. An energy audit is a process which can be divided into the following steps:

Step 1: Boundary Identification and Physical Inspection

The first step in this process is to define the boundary of the system and establish the mass and energy balances over the defined boundaries of the energy system. It includes the physical inspection of the system and all the elements that form the system. An example of a cooling system is presented in Fig. 5. Three main parts can be identified:

– Cooling Tower System (within the boundary)
– Distribution System of Cooling Water
– End-users

The task of the cooling tower is simply to remove the heat generated by end-users. If the cooling tower is 'big', it may accomplish the duty by cooling water temperature, for example, from 35 to 30 ^oC. If it is 'small', it may cool the water in the same process from 38 to 33 ^oC. In both cases, the heat removed from the process to be cooled will be the same, if the flow rate is the same. The size of the cooling tower, the flow rate and the wet bulb temperature determine the inlet and outlet water temperatures, but not the difference between them.

In this phase of the system analysis, the measuring points have to be identified. In Fig. 12.5, the following points are identified:

- **M1: Supply Water.** The flow rate of water and its temperature has to be measured.
- **M2: Return Water.** The temperature of return water has to be measured.

If measurements are performed, the heat capacity of cooling system will be:

$$Q\,[kW] = M_w\ [kg/s] \cdot 4.19\ [kJ/(kg\ ^oC)] \cdot \left(T_{w,out} - T_{w,in}\right)[^oC] \qquad (12.5)$$

For the proper operation of end-users, it is sometimes very important that the temperature of the supply water is as designed.

The equation above defines the **load** of the system and it is obvious that the load varies with time and depends on the processes that have to be cooled. Load versus time is the **load profile** and for any energy analysis it must be defined.

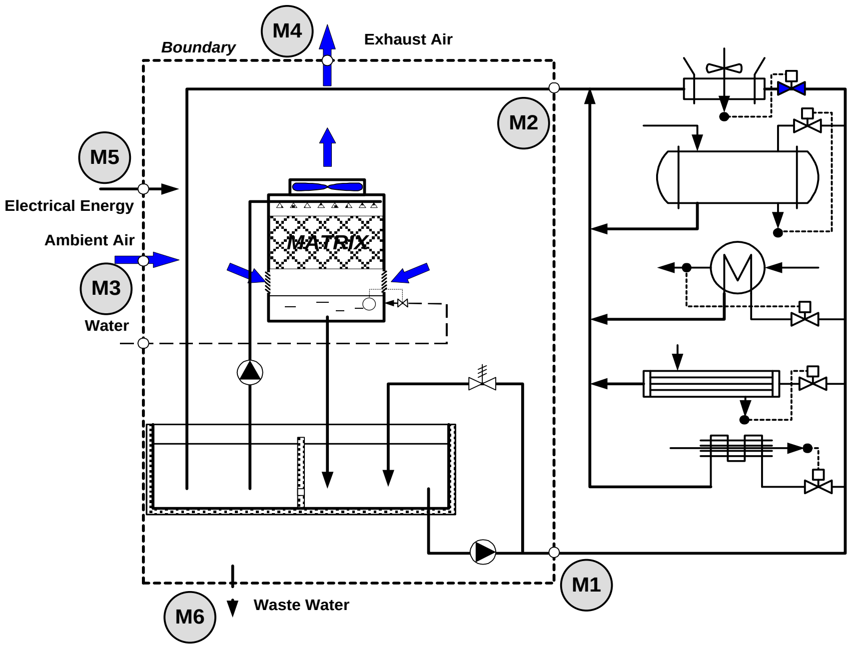

Figure 12.5: Possible Scheme of Cooling System

- **M3: Ambient Air Parameters.** In a cooling tower system water is cooled by moist air and, because of that, the parameters of the air must be known. Of all moist air parameters, the most important for cooling tower operation is wet bulb temperature. It is only wet bulb temperature that strongly influences the enthalpy of air. The influence of other parameters such as dry bulb temperature and pressure are within negligible ranges which occur in the normal operation of cooling towers. The enthalpy of air versus wet bulb temperature is presented in Fig. 12.6.

The analytical relation of enthalpy versus wet bulb temperature can be represented by an analytical expression as follows:

$$h = 1.618366 \cdot 10^{-3} \cdot t_{WB}^3 - 2.703963 \cdot 10^{-2} \cdot t_{WB}^2 + 2.512014 \cdot t_{WB} + 4.574424 \qquad (12.6)$$

The variation of ambient air parameters can be great during a day or month and will cause the variation of cooling tower heat capacity and supply water temperature.

The flow rate of air has to be known as it influences the cooling tower's heat capacity.

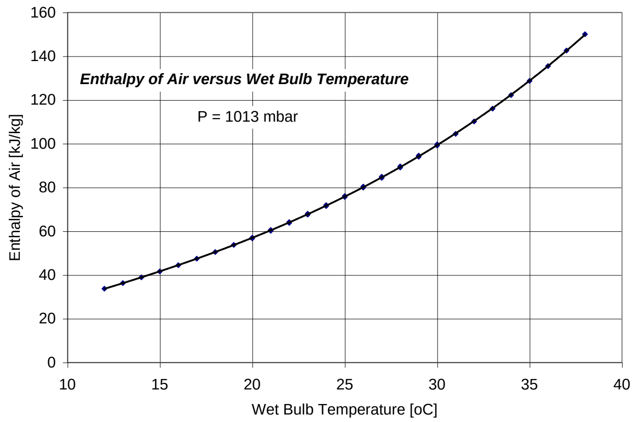

Figure 12.6: Enthalpy versus Wet Bulb Temperature (the Influence of Dry Bulb Temperature is Negligible)

M4: Exhaust Air. The evaporation process within the cooling tower increases the absolute humidity of exhaust air. Evaporation loss is practically the same for both counter- and cross- flow cooling towers. As the air is almost saturated, the measurement of relative humidity or wet bulb temperature is very difficult.

M5: Electrical Energy. Electrical energy is used for running the pump(s), fan(s) and auxiliary equipment. The pump which runs water throughout the cooling tower is called the **circulating pump**. The pump or pumps that are used for supplying end-users with cooling water are **supply pumps** and, if there is a pump which returns the water from end-users to the cooling tower, it will be called the **return pump**.

Most of the electrical energy is used for running the pumps. The power of a fan's electrical motor is typically from 25 to 35 % of the circulating pump’s power.

M6: Waste water. The blow down is necessary in order to prevent the concentration of dissolved solids from increasing to the point where it may precipitate and scale up heat exchangers and the cooling tower fill, or reach unacceptable levels for other reasons.

Economics plays an important role in the selection of water treatment methods. It is possible to prescribe ideal water for each cooling job, but the costs of equipment and chemicals are unacceptable for most installations. So, treatment must be tailored to fit the size of any type of cooling-water system currently in use.

The calculation of blow down is explained above, but the blow down loss is small compared to the flow of circulating water flow.

During physical inspection, all of the relevant parts of the cooling tower system have to be checked and repaired if necessary. In the previous section, all of the relevant elements are mentioned. The potential measurement points and instrumentation have also to be checked. The methods of any additional measurements necessary for energy audit have to be analyzed.

Step 2: Equipment Specification

An energy audit entails measurements, technical calculations and analysis. This means that a great deal of technical data must be known. The most important thing is to know what are the design data and/or standards for any measured and written value in the daily log sheets. This means that the energy flows entering the boundary must always be known. The waste energy and material must also be known.

The typical sets of data for any cooling tower are as follows:

Name plate data

Type of cooling tower			
Manufacturer			
Age			
Type and characteristics of matrix			
Design parameters:			
- *Water temperatures in*	$t_{w,in}$	^oC	
- *Water temperatures out*	$t_{w,out}$	^oC	
- *Mass flow rate of water*	L	kg/s	
- *Wet bulb air temperature in*	$t_{a,WB}$	^oC	
- *Mass flow rate of air*	G	kg/s	

- Control system: ____
- Total annual operating time: ____________________________________
- Total annual quantity of water supplied to the system: __________________
- Instrumentation and metering system: ____________________________
- Housekeeping procedure: _______________________________________
- Water analysis report:
  - *– pH =* ___________
  - *– Total dissolved solids of cooling tower water* (TDS_{CTW}) *=* ________
  - *– Total dissolved solids of make-up water* (TDS_{MuW}) *=* _____________
- Cooling tower fan:
  - *– Mass flow rate of air*
  - *– Head of fan*
  - *– Power of electrical motor*
  - *– Type of fan*
  - *– Type of transmission*
  - *– Fan characteristics*
- Cooling tower pump (circulating pump):
  - *– Mass flow rate of water*
  - *– Head of pump*
  - *– Power of electrical motor*
  - *– Type of pump*
  - *– Type of transmission*
  - *– Pump characteristics*
- Supply water pump
  - *– Mass flow rate of water*
  - *– Head of pump*
  - *– Power of electrical motor*
  - *– Type of pump*
  - *– Type of transmission*
  - *– Pump characteristics*

Step 3: Daily Log Sheets and Maintenance Records Analysis

The daily log sheet normally gives information on water temperature and pressure in and out, dry bulb temperature and relative humidity or wet bulb temperature, and current (or electrical energy consumption – kWh) of the electrical motors of the cooling system. In factories, regular water flow measurements are rarely found and air flow measurements almost never.

Maintenance records provide the opportunity to estimate the condition of the analyzed energy system. Preventive and regular maintenance always reduce maintenance costs significantly and increase the reliability and prolong the lifetime of equipment.

These matters have to be analyzed and improved within the implementation of the energy management system.

Daily, weekly or/and monthly reports are desirable.

Step 4: Meteorological Data

Meteorological data could be very important for energy audit analysis. Many energy systems depend strongly on meteorological data (heating, drying, air conditioning, etc.). In the case of cooling tower systems the most dominant influence comes from the wet bulb temperature of the air. This value does not vary by much. For example, the design wet bulb temperature in San Francisco and Bogota is 18 ^{o}C and for Hanoi is 30 ^{o}C. However, this value influences the dimensions dramatically and, later, the operation of cooling towers.

These data can be obtained from a local meteorological institution.

Step 5: Measurements

The analysis performed during the realization of the actions described in Steps 1–3 has to be used for the preparation of the measurement plan. How detailed the plan will be depends on the expected results.

This means that in this phase the **target** must be clearly identified.

For example, if the water temperature difference is higher than designed, the cause can be a low water flow rate, problems with the bed water spraying system, damage of the cooling tower filling, increased load due to new machines installed, or a scaling problem at the machine heat exchanger, etc. In this case, all potential causes must be checked and quantified. The water flow rate has to be measured, the list of installed machines and their cooling load prepared and the spray system assessed as either ‘poor’ or ‘good’, etc.

This procedure provides an opportunity to eliminate some of the causes and to solve the problems of cooling system operation. All of these actions will result in energy efficiency improvement and, at the same time, reduction in energy consumption.

The measurements necessary for cooling system analyses have already been defined in Step 1 and presented pictorially in Fig. 12.5. The frequency of measurement, the type of instruments used, their accuracy and the organization of measurements have to be the subject of independent analysis.

Step 6: Load Profile

A typical daily load is presented in Fig. 12.7. The presented heat load meets the real production program. There is an automatic control system for the temperature of fluid to be cooled. The **design temperature** of cooling tower water out is 32 ^{o}C and the **design temperature difference** is 5 ^{o}C. However, as ambient parameters vary during the year, the load profile has to be defined for any characteristic period of the year.

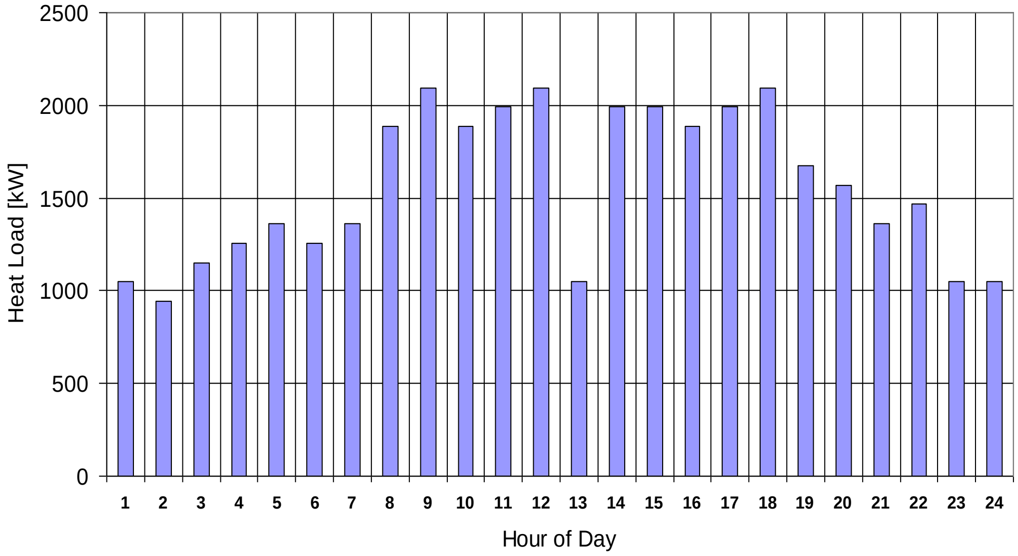

Figure 12.7: Typical Load Profile

Step 7: Calculation and Analysis

It is very difficult to prescribe the types of calculation needed during an energy audit. But, it is possible to identify the calculations that are used most often. As it has already been mentioned, the energy audit is based on everyday measurements performed by technical staff and reported in daily log sheets or by performing some advanced measurements prepared in advance with some special task which will result in energy savings.

Software 10: ***Cooling Tower Calculation*** offers solutions to four problems for both counter or cross flow cooling towers:

1. ***DETERMINATION OF PERFORMANCE INDICATORS***
2. ***COMPARISON OF COOLING TOWER PERFORMANCE INDICATORS WITH DESIGN DATA (ACCEPTANCE TEST)***
3. ***INFLUENCE OF MASS FLOW RATES OF AIR AND WATER VARIATION***
4. ***CALCULATION OF WATER MASS FLOW RATE FOR GIVEN LOAD AND WET BULB TEMPERATURE***

The cooling tower performance indicators are:

1. ***Number of Transfer Units (NTU):***

$$NTU = \frac{h_c \cdot A}{c_{p,m}} \tag{12.7}$$

$c_{p,m}$ = mean isobaric specific heat of moist air, [kJ/(kg K)]

2. ***Heat capacity (Q):***

$$Q = L \cdot 4.19 \cdot (T_{w,out} - T_{w,in}) \; [kW] \tag{12.8}$$

3. Range (r)

$$r = T_{w,in} - T_{w,out} \quad [^{o}C] \tag{12.9}$$

4. Approach (a)

$$a = T_{w,out} - t_{a,wb,in} \quad [^{o}C] \tag{12.10}$$

5. Cooling degree (η_{CT})

$$\eta_{CT} = \frac{T_{w,in} - T_{w,out}}{T_{w,in} - t_{a,wb,in}} \quad [-] \tag{12.11}$$

Step 8: Energy Conservation Opportunities

Generally speaking, after improving the existing cooling tower system by curing the deficiencies found during the physical inspection of the site and satisfying optimal performance indicators, there are only a few opportunities for reducing energy consumption:

a) *Operational procedure improvement to meet the real load and design supply water temperature*

The ambient air wet bulb temperature will vary during the day and during the year. For any analysis of the cooling tower system it has to be known in relation to weather conditions. The change of heat capacity versus ambient wet bulb temperature for a particular counter flow cooling tower is presented in Fig. 12.8. The outlet water temperature is also presented. In the case of more than one cooling tower, sequential control has to be used. It can be done manually or automatically. For low wet bulb temperatures often only one cooling tower will be enough. This procedure demands the measuring of wet bulb temperature and load.

b) *Implementation of variable speed drive control systems for pumps and fans*

The more advanced technique used for adjusting the operation of cooling system to meet the real load entails the implementation of variable speed drive control of pumps and fans.

The capacity of a cooling tower depends strongly on water and airflow rates (Figs 12.9 and 12.10).

c) *Improvement of the control system and operational procedure of end-users*

The analysis performed in this chapter assumes that the load is given. This means that the operation of the end-users has not been subjected to analysis. However, it has to be stressed that the analysis of end-users' operation has also to be done. For example, in many industries there is no automatic closing of cooling water valves when a machine is not in operation. If an electromagnetic valve is installed, after stopping the machine the cooling water valve will be closed automatically or will be closed after the time necessary for cooling the part of the relevant machine. If such valves are installed, the load profile will be changed.

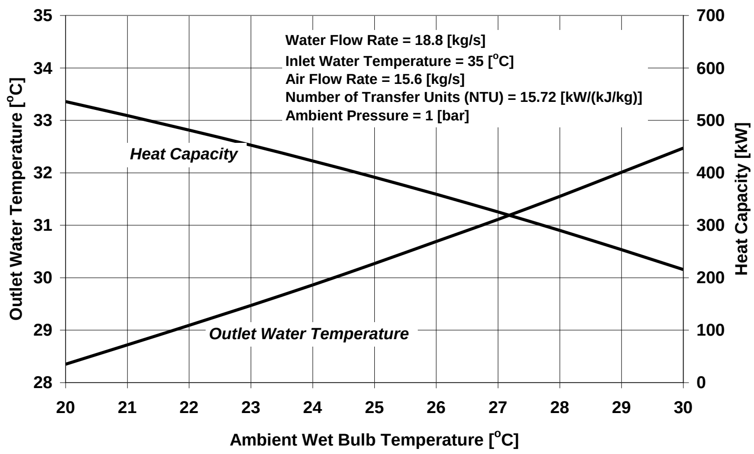

Figure 12.8: Heat Capacity and Outlet Water Temperature versus Ambient Wet Bulb Temperature

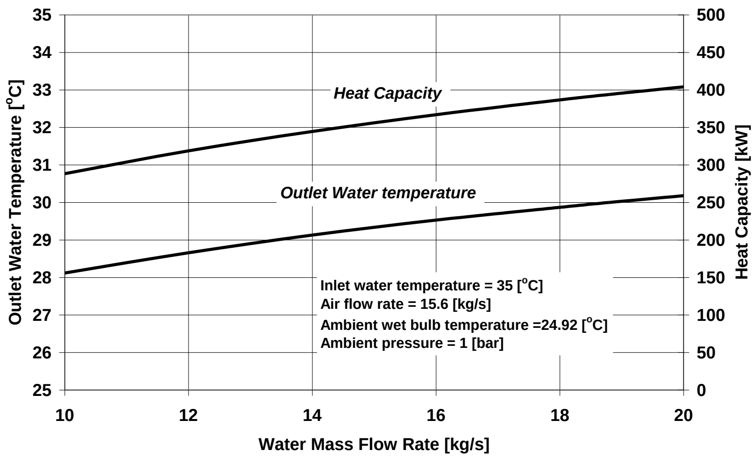

Figure 12.9: Heat Capacity and Outlet Water Temperature versus Flow Rate of Water and Constant Air Flow Rate

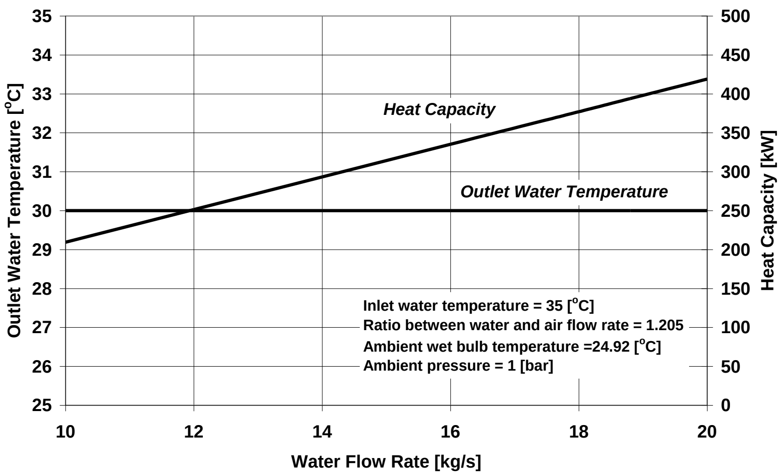

Figure 12.10: Heat Capacity and Outlet Water Temperature versus Water Flow Rate for Constant Ratio of Water Flow Rate and Air Flow Rate

Step 9: Financial Evaluation and Cost Benefit Analysis (see Toolbox 3)

This step will not be evaluated here. With precise technical and technological goals to improve existing cooling tower energy efficiency, financial evaluation can be done in direct contact with equipment manufacturers. With their prices and costs and financial capabilities, it is easy to prepare the elements for a final decision.

9. **Energy Audit Example**

Step 1: Boundary Identification and Physical Inspection

- **Scheme of the System:**

The boundaries of the system are presented in Fig. 12.11. At the same time, the measuring points are also defined.

The overflow valve is opened if the pressure after the supply pump exceeds 5 barg. This has to protect the pump from overloading.

A physical inspection will show that there is no visible damage to the system.

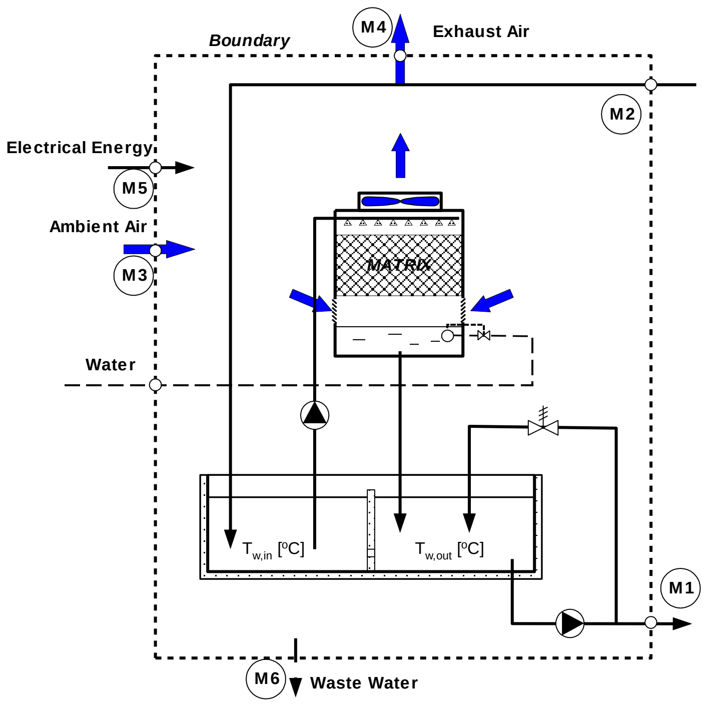

Figure 12.11: Scheme of the Cooling System

Step 2: Equipment Specification

- **Cooling Tower Specifications (as designed):**

LBC 600 – counter flow		
Ambient wet bulb temperature	=	28 ^{o}C
Water-in temperature	=	37 ^{o}C
Water-out temperature	=	32 ^{o}C
Water flow rate	=	6736 l/min
Air flow rate	=	3750 m^3/min
Power of fan motor	=	14.7 kW

- **Circulating Pump Specifications:**

At working point (as designed):

Centrifugal		
Flow rate	=	112.3 l/s (6738 l/min)
Head	=	25 mWC
Power of electrical motor	=	39.33 kW
Speed	=	1500 rpm

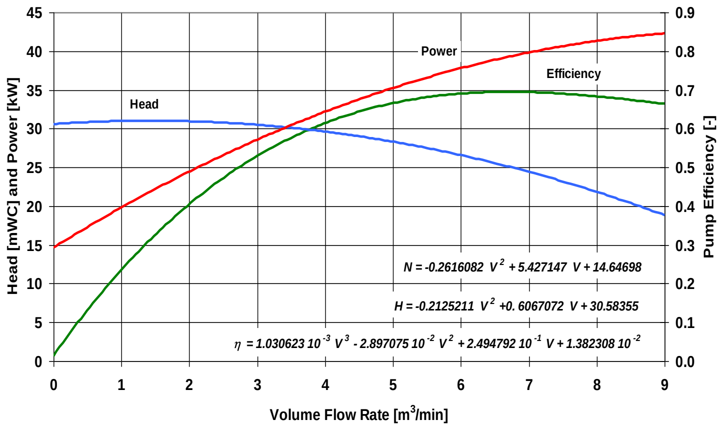

Figure 12.12: Head, Power and Pump Efficiency versus Volume Flow Rate of Circulating Pump

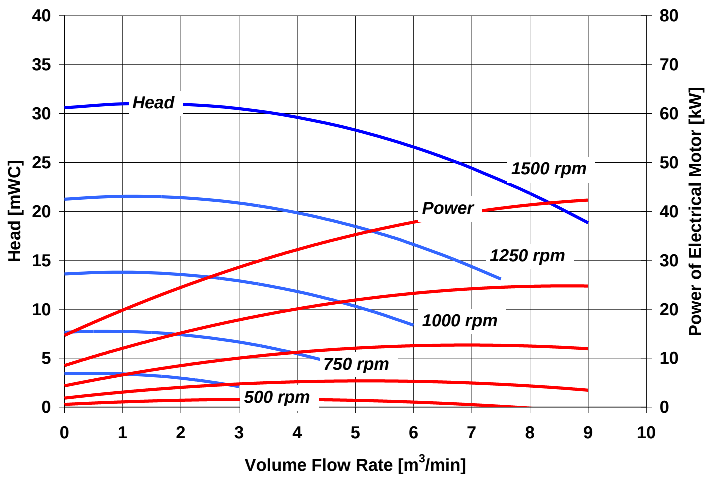

Figure 12.13: Head and Power versus Volume Flow Rate of Circulating Pump for Different Speeds

- **Supply Water Pump Specifications:**

At working point (as designed):
Centrifugal

Flow rate	=	150 l/s (9000 l/min)
Head	=	55.6 mWC
Power of electrical motor	=	115.5 kW
Speed	=	1750 rpm
Fluid temperature	=	from –15 to 180 ^{o}C
Maximum working pressure	=	12 bar
Material	=	Cast iron casing
Inlet diameter	=	250 mm
Discharge diameter	=	200 mm
Impeller diameter	=	342 mm

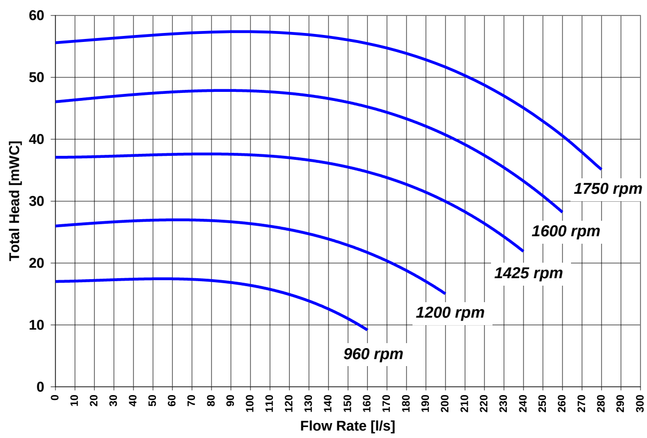

Figure 12.14: Head versus Volume Flow Rate of Supply Pump for Different Speeds

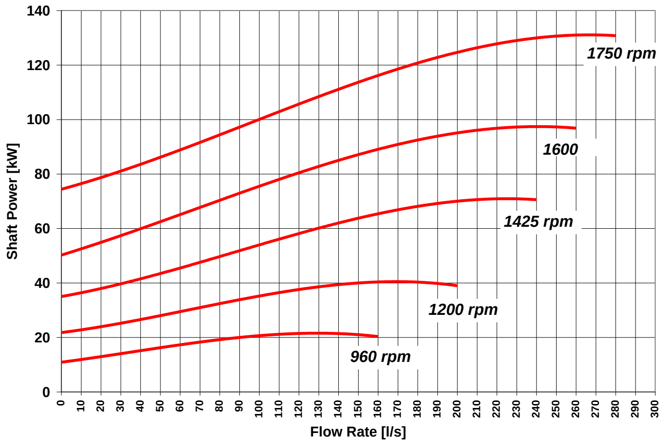

Figure 12.15: Shaft Power versus Volume Flow Rate of Supply Pump for Different Speeds

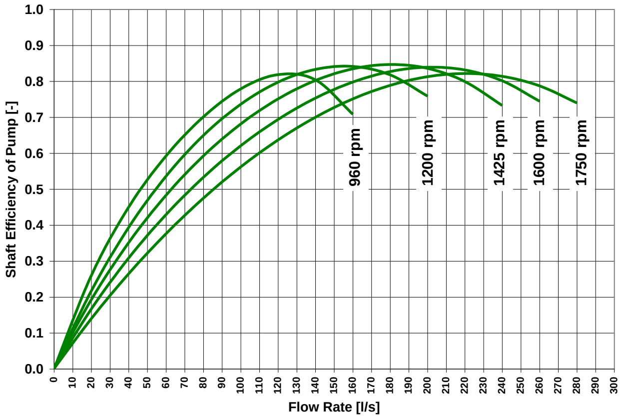

Figure 12.16: Efficiency versus Volume Flow Rate of Supply Pump for Different Speeds

Step 3: Daily Log Sheet and Maintenance Record

The system operates for 8000 hours annually. The production is more or less stable and varies from 75 to 90 % of installed capacity.

Daily log sheets contain the water temperature in and out and the pressure of water after the supply pump. Only the current for the whole system is measured. These measurements show that consumption is more or less constant.

Water and air flows have not been measured.

Water analysis has been performed once every day. pH and conductivity have been measured and controlled. In compliance with performed measurements, chemicals have been added manually.

The water temperature after supply pump is approximately 32 ^oC and return water temperature from end-users is approximately 35 ^oC. This means that the temperature difference is around 3 ^oC. As the flow rate can be assumed to be constant (constant measured current), it can be concluded that the supply water flow rate is greater than designed (5 ^oC).

From equipment specifications it can be recognized that the flow rate of the circulating pump is 6736 l/min and the supply water pump flow rate is 9000 l/min. There is no reason for the supply pump flow rate to be greater than the circulating pump flow rate.

By inspecting the maintenance records, it is found that three years ago the supply pump was replaced by a new one and it was decided then to buy a bigger pump for no special reason. After that period, the temperature difference dropped from approximately 5 ^oC to 3 ^oC.

The variation of the supply water temperature difference shows that there are significant variations in the heat load.

There have been no complaints from production departments about the operation of the cooling system. This means that during the whole year it has operated within the design's anticipated ranges.

Step 4: Meteorological Data

These Values Appear Annually Approximately 1752 (20 %) Hours (For Pressure 1.013 Bar)

MEAN Max	**T [°C]**	**RH [%]**	**t_{wb} [°C]**	**h [kJ/kg$_{da}$]**	**x [-]**	**ρ [kg/m³]**
Jan	32.0	89.0	30.41	101.59	0.02710	1.138
Feb	32.7	91.0	31.39	106.95	0.02891	1.134
Mar	33.7	91.0	32.36	112.50	0.03066	1.130
Apr	34.9	89.0	33.20	117.54	0.03213	1.124
May	34.0	92.0	32.81	115.13	0.03156	1.128
Jun	33.1	90.0	31.63	108.27	0.02926	1.133
Jul	32.7	91.0	31.39	106.95	0.02891	1.134
Aug	32.5	92.0	31.35	106.71	0.02890	1.135
Sep	32.3	94.0	31.45	107.30	0.02921	1.136
Oct	32.0	94.0	31.16	105.67	0.02870	1.137
Nov	31.6	91.0	30.33	101.13	0.02709	1.140
Dec	31.3	88.0	29.59	97.26	0.02570	1.142
Average	**32.7**	**91.0**	**31.42**	**107.25**	**0.02901**	**1.134**

These Values Appear Annually Approximately 5256 (60 %) Hours

MEAN	**t [°C]**	**RH [%]**	**t_{wb} [°C]**	**h [kJ/kg$_{da}$]**	**x [-]**	**ρ [kg/m³]**
Jan	25.9	71.0	22.00	64.05	0.01491	1.170
Feb	27.4	75.0	23.97	71.61	0.01726	1.162
Mar	28.7	75.0	25.16	76.54	0.01866	1.156
Apr	29.7	75.0	26.08	80.50	0.01981	1.152
May	29.2	78.0	26.09	80.53	0.02002	1.153
Jun	28.7	78.0	25.62	78.50	0.01943	1.156
Jul	28.3	78.0	25.25	76.91	0.01897	1.157
Aug	28.1	79.0	25.21	76.76	0.01899	1.158
Sep	27.8	82.0	25.38	77.44	0.01938	1.159
Oct	27.6	81.0	25.04	76.03	0.01891	1.160
Nov	26.9	76.0	23.66	70.37	0.01698	1.164
Dec	25.6	71.0	21.73	63.05	0.01464	1.171
Average	**27.8**	**76.6**	**24.60**	**74.36**	**0.01816**	**1.160**

The mean temperature and relative humidity are calculated by using following formulae:

$$t_{mean}(daily) = \frac{1}{8} \cdot \sum_{n=1}^{8} t_n \quad RH_{mean}(daily) = \frac{1}{8} \cdot \sum_{n=1}^{8} RH_n \tag{12.12}$$

where t_n and RH_n are the dry bulb temperature and relative humidity, respectively, of the ambient air measured every 3 hours starting from 00:00.

The monthly mean dry bulb temperature and relative humidity are calculated as average daily mean temperature and relative humidity:

$$t_{mean}(monthly) = \frac{1}{M} \cdot \sum_{m=1}^{M} t_{m,mean}(daily) \quad RH_{mean}(monthly) = \frac{1}{M} \cdot \sum_{m=1}^{M} RH_{m,mean}(daily) \tag{12.13}$$

where M is the number of days in a month.

Using everyday maximum and minimum values measured by maximum and minimum probes and the following formulae it is possible to calculate MAX and MIN temperatures and relative humidities:

$$t_{MAX}(\text{monthly}) = \frac{1}{M} \cdot \sum_{m=1}^{M} t_{m,MAX}(\text{daily}) \quad RH_{MAX}(\text{monthly}) = \frac{1}{M} \cdot \sum_{m=1}^{M} RH_{m,MAX}(\text{daily}) \tag{12.14}$$

$$t_{MIN}(\text{monthly}) = \frac{1}{M} \cdot \sum_{m=1}^{M} t_{m,MIN}(\text{daily}) \quad RH_{MIN}(\text{monthly}) = \frac{1}{M} \cdot \sum_{m=1}^{M} RH_{m,MIN}(\text{daily}) \tag{12.15}$$

These Values Appear Annually Approximately 1752 (20 %) Hours

MEAN Min	**t [°C]**	**RH [%]**	**t_{wb} [°C]**	**h [kJ/kg$_{da}$]**	**x [-]**	**ρ [kg/m^3]**
Jan	21.0	48.0	14.31	39.94	0.00741	1.194
Feb	23.3	54.0	17.14	47.92	0.00962	1.183
Mar	24.9	55.0	18.65	52.58	0.01081	1.176
Apr	26.1	55.0	19.67	55.89	0.01162	1.171
May	25.6	60.0	20.05	57.16	0.01232	1.172
Jun	25.4	62.0	20.19	57.63	0.01259	1.173
Jul	25.0	62.0	19.84	56.45	0.01229	1.175
Aug	24.9	63.0	19.90	56.67	0.01241	1.175
Sep	24.6	64.0	19.79	56.29	0.01239	1.176
Oct	24.3	64.0	19.52	55.41	0.01216	1.178
Nov	23.1	58.0	17.58	49.24	0.01022	1.184
Dec	20.8	51.0	14.59	40.68	0.00778	1.195
Average	**24.1**	**58.0**	**18.44**	**52.15**	**0.01097**	**1.179**

It is estimated that MAX mean and MIN mean temperatures and relative humidities appear approximately 1752 hours per year and mean ones approximately 4256 h/year.

Step 5: Target Setting and Measurements

The previous analysis shows that there are no significant technical problems in the operation of the analyzed system. However, the electrical energy consumption of this system is estimated to be approximately 5 % of the total electrical energy consumption. The estimation is based on measurements of the total electrical energy consumption (what is paid) and the electrical energy consumed by this system and the compressed air system (which is on the side of the end-users of the cooled water) as they have a kWh meter. The participation of this system in electrical energy consumption is estimated on the installed power of both systems.

The main targets of the energy audit are formulated as follows:

- ***Improve the efficiency of the system by implementing the Variable Speed Drive (VSD) control system in both the pumps and the fan.***
- ***Analyze the possible consequences for the operations of end-users that can be produced by the implementation of VSD control.***

The basic assumptions of the technical and financial analysis are:

- The load will not be changed in the near future.
- The price of electricity will not change in the next 2–3 years.

The following plan for measurement is prepared (readings or measurements are done hourly):

Cooling Tower

- Water flow measurement of circulating water
- Water temperature in
- Water temperature out
- Air flow measurement
- Dry bulb temperature measurement
- Relative humidity of ambient air
- Pressure of air

Supply Water System

- Water flow measurement of supply pump
- Supply water temperature
- Return water temperature

The list of instruments is not presented. The standard equipment is used.

The results are shown in the form of a graph in Figs 12.17–12.22. Figure 12.17 shows the dry and wet bulb temperatures. It is easy to conclude that the wet bulb temperature (which is the most important for cooling tower operation) is very stable. The mean value is 26.6 [^{o}C]. This temperature is quite high generally speaking, but for the area of Bangkok (Thailand), it is so very often (the designed wet bulb temperature for Bangkok is 28 ^{o}C).

The air flow rate (Fig. 12.18) is measured by an anemometer. The velocity of the air is measured at 24 points around the cooling tower suction area. The average velocity is multiplied by area and density to get the average air flow rate. The average flow rate is 72.0 [kg/h] with StDev = 5.1 [kg/s] or 7.1 [%].

The average supply water flow is 141.1 [kg/s] (Fig. 12.19) with a standard deviation of 4.4 [kg/s]. The StDev is only 3.1 [%] and can be the error of measurement. Error includes both instrumentation errors and errors of method. The power of the electrical motor is not measured.

The average circulating water flow (Fig. 12.21) is 110.0 [kg/s] with a standard deviation of 2.5 [kg/s] or 2.8 %. The power of the electrical motor is not measured. The comment on this measurement is the same as for supply water measurement.

The temperatures of water are measured by immersing the thermometer (thermocouple). The bush is installed in the pipeline after the supply water pump and return water tank. The circulated water temperature is measured in the same way after the circulating pump and in the collecting basin at the bottom of the cooling tower (Figs 12.21 and 12.22).

Average values are used for further calculations because the variation of flow measurements is not significant and can be within the range of measurement errors. This means that it is not possible to find a correlation between measured flow rates, measurement errors and what happens on the side of end-users.

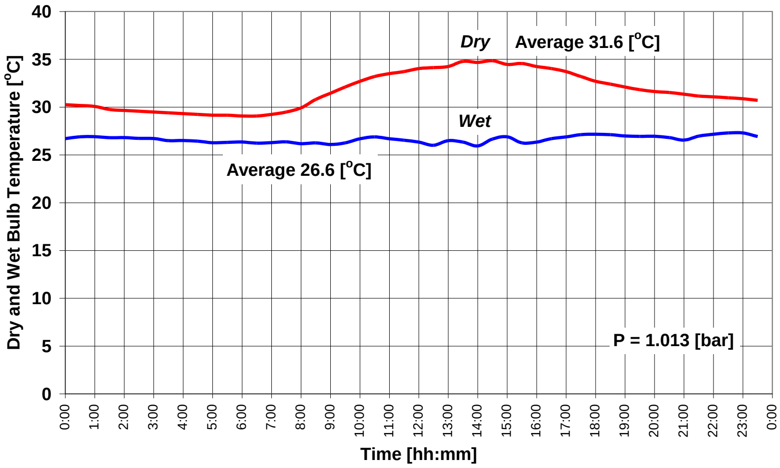

Figure 12.17: Dry and Wet Bulb Temperature of Ambient Air

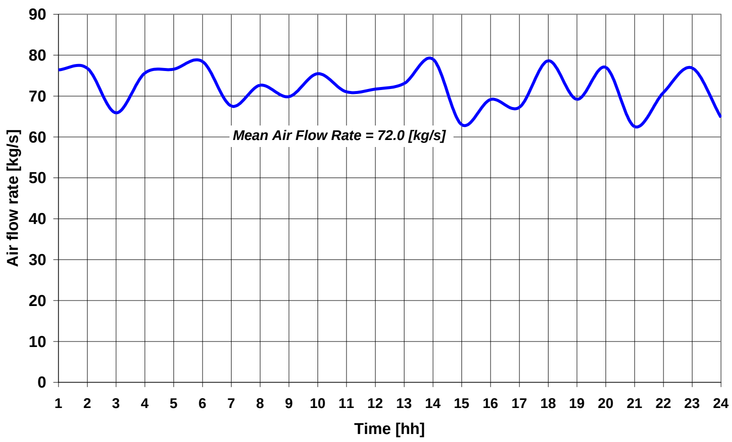

Figure 12.18: Air Flow Rate versus Time

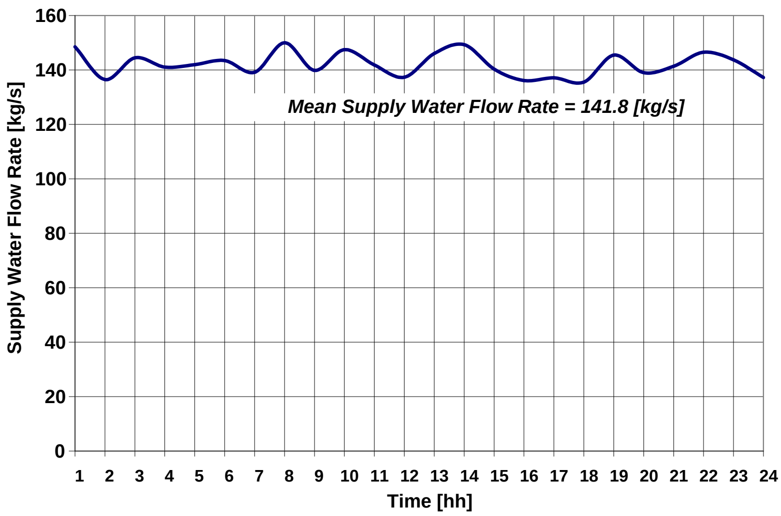

Figure 12.19: Supply Water Flow Rate versus Time

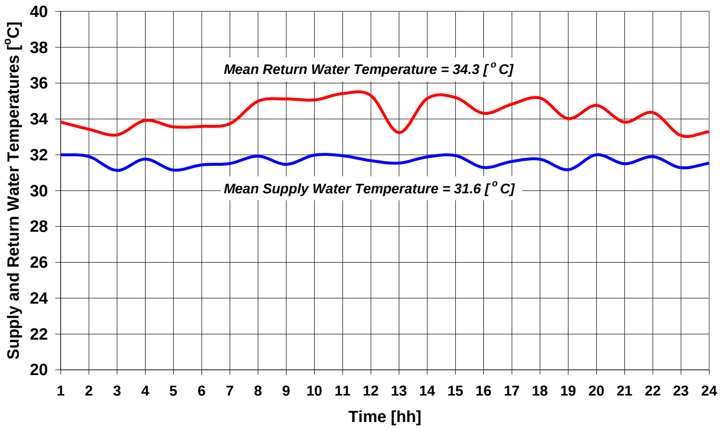

Figure 12.20: Supply Water Temperature In and Temperature Difference versus Time

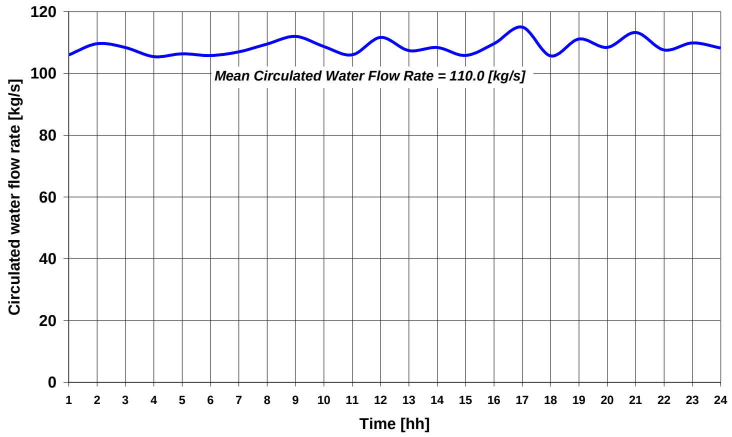

Figure 12.21: Circulated Water Flow Rate versus Time

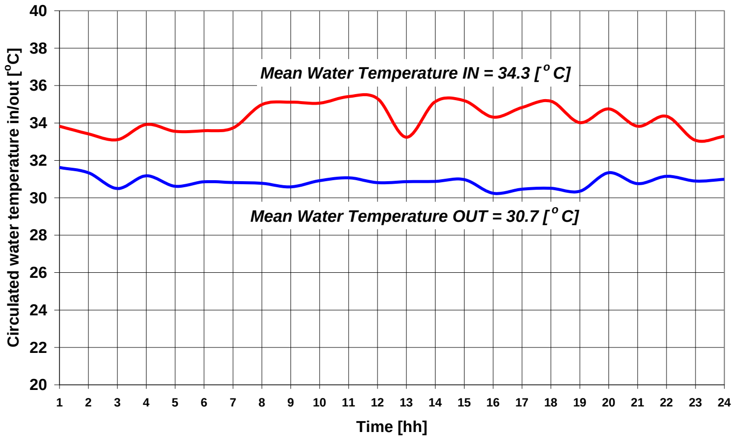

Figure 12.22: Circulated Water Temperature Out and Temperature Difference versus Time

Finally, the values relevant for further calculation are:

Mass flow rate of circulated water (L)	=	110.0 [kg/s]
Temperature of circulated water in ($T_{cw,in}$)	=	34.3 [^oC]
Temperature of circulated water out ($T_{cw,out}$)	=	30.7 [^oC]
Mass flow rate of air (G)	=	72.0 [kg/s]
Wet bulb temperature (t_{wb})	=	26.6 [^oC]
Mass flow rate of supply water (M_{sw})	=	141.8 [kg/s]
Temperature of supply water in ($T_{sw,in}$)	=	31.6 [^oC]
Temperature of supply water out ($T_{sw,out}$)	=	34.3 [^oC]

Step 6: Load Profile

The load profile represents the heat energy removed from process versus time. For the analyzed case it is presented in Fig. 12.23. This heat load meets the real production program. The water flow is controlled manually before the water enters the machine. There are no electromagnetic valves in front of the machines which will be closed when the machine is not in operation. The design temperature of the water-in is 32 ^oC and the temperature difference is 5 ^oC.

If the load is split into 12 intervals starting from 900 kW by steps of 100 kW, it is possible to calculate the frequency of load appearance over 24 hours. It is presented in Fig. 12.24. The maximum load (1950–2100 kW) appears approximately 30 % of time and the rest of the time system operates with a load lower than maximum.

The average daily load is 1562.5 kW. As the cooling tower operates with a maximum water flow rate this means that the temperatures of water in and out will be lower proportionally to the real load.

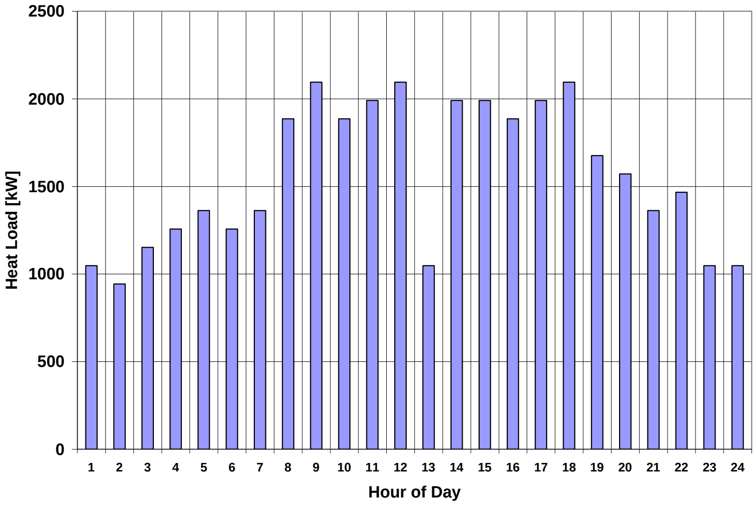

Figure 12.23: Daily Load

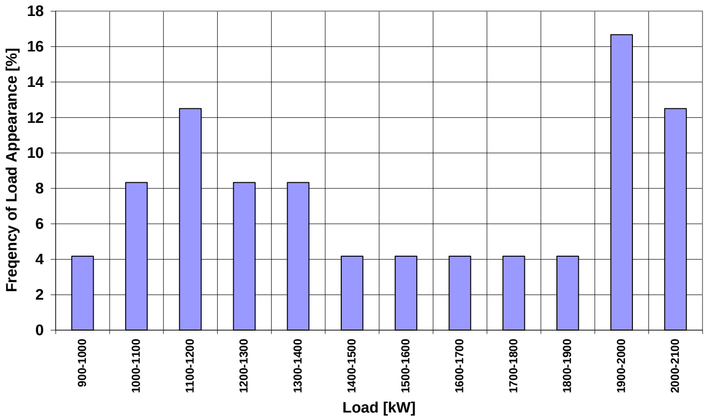

Figure 12.24: Frequency of Load Appearance during the Day

Step 7: Calculation and Analysis

For the following measurement data:

p_a = 1.013 bar
L = 110 kg/s
$T_{w,in}$ = 34.30 ^oC
$T_{w,out}$ = 31.60 ^oC

G = 72.00 kg/s
$t_{db,in}$ = 31.60 ^oC
RH_{in} = 68.2 %

by using Software (PROBLEM 1) it is possible to calculate the following cooling tower performance indicators:

NTU = 51.78 kW/(kJ/kg)

Q = 1244.43 kW
Cooling degree = 0.351
Range = 2.7 ^{o}C
Approach = 5.0 ^{o}C

By using NAME PLATE data (PROBLEM 2) and the already calculated NTU, the measured flow rates and **design** temperature of air and water and a capacity of the cooling tower of 1562.53 kW are obtained. This means that the actual capacity is lower by 33.6 % than that designed. The acceptance test shows that the manufacturer delivered a cooling tower with a lower capacity. But, as the guarantee period has expired, it is decided to accept the real capacity and to continue with the energy audit.

Step 8: Energy Conservation Opportunities

ECO1: Increasing the air flow rate

The ratio is L/G = 1.53. It is decided to increase the flow rate of the fan from 72 kg/s to 90 kg/s by replacing the electrical motor and transmission. The new ratio L/G will be 1.22. However, the capacity of the cooling tower will increase from the current 1244.43 kW to 1431.89 kW (13.1 %). The new NTU will be 60.80 kW/(kJ/kg).

ECO2: VSD control of pumps and fan

It is estimated that the most promising measure for the reduction of energy consumption is the implementation of a variable speed drive for both pumps and the fan because:

a. The maximum load appears during only 30 % of the operating time. For the rest of the operating time, the pumps can operate the system with much lower flows.
b. The supply pump is bigger than it is necessary and expends more energy for circulating water around the network with a low temperature difference. This can be solved by closing the proper valve but the energy consumption will be approximately the same.
c. All end-users are designed to operate with a supply water temperature of 32 ^{o}C and a temperature difference of 5 ^{o}C.

The scheme of the cooling system with new VSD control systems is presented in Fig. 12.25.

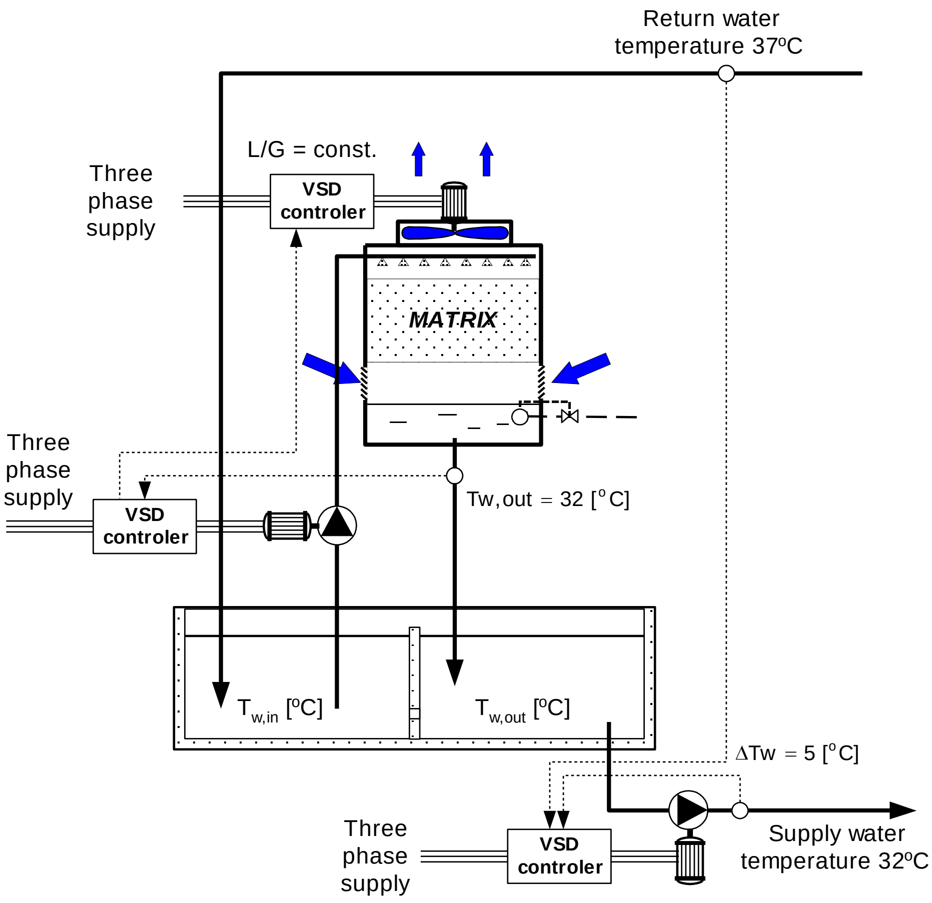

Figure 12.25: Proposed Scheme of the Cooling System

The sensors for supply and return water temperature signal the temperatures to the controller where the difference is calculated (the set temperature difference is 5 °C). The controller responds to this temperature difference and adjusts the pump flow rate.

At the same controller of the circulating pump, the cooling tower flow rate is adjusted to correspond to the outlet temperature of 32 °C. The controller of the fan follows the circulating pump flow rate to keep the L/G ration on 1.22. The positioner for this control system produces the signal proportional to the water flow rate and uses this signal to adjust the air flow rate in order to keep the ratio L/G constant. Some of the details of the control systems can be and have to be discussed with the manufacturers directly.

In the current case, the energy consumption can be estimated as follows:

$$E_{Current} = (4.7 + 39.33 + 115.5) \cdot 8{,}000 = 1{,}356{,}240 \quad [kWh] \tag{12.16}$$

However, in the case where the system reacts to the ambient wet bulb temperature and changes the load, the energy consumption calculation is more complicated. One possible approach is as follows:

(a) The ambient air parameters relevant to the calculation of energy consumption are:

MAX mean (1752 hours per year):

t_{db} = 32.7 °C $\quad$ t_{wb} = 31.42 °C

MEAN (5256 hours per year):

t_{db} = 27.8 °C $\quad$ t_{wb} = 24.6 °C

MIN mean (1752 hours per year):

t_{db} = 24.1 °C $\quad$ t_{wb} = 18.44 °C

(b) The load that appears can be divided into the following classes:

Class	***Load [kW]***	***Appears [%]***
I	900–1000	4.17
II	1000–1100	8.33
III	1100–1200	12.5
IV	1200–1300	8.33
V	1300–1400	8.33
VI	1400–1500	4.17
VII	1500–1600	4.17
VIII	1600–1700	4.17
IX	1700–1800	4.17
X	1800–1900	4.17
XI	1900–2000	16.67
XII	2000–2100	12.5

(c) The necessary circulating pump flow rates and fan flow rates for the different parameters of the ambient air and for different capacity are presented in Table 3.

It is obvious that with MAX mean ambient air parameters there are no opportunities to meet a load from 1200 to 2100 kW with water temperatures of 32/37 ^{o}C. For a capacity of 2050 kW, the return water temperature will be 40 ^{o}C and for supply water 35.87 ^{o}C. This means that for such severe ambient conditions (32.7/31.4 ^{o}) the cooling tower capacity is low to meet the load. That can imply some problems for production. In the previous period, although such a high temperature happened, it was not reported in the daily log sheet. In the daily log sheets, it was found that for only a few days the return water temperature was equal or greater than 40 ^{o}C.

Table 3: Circulating Flow Rate versus Capacity

Capacity kW	**MAX mean**	**MEAN**	**MIN Mean**
	L [kg/s]	**L kg/s]**	**L [kg/s]**
950	87.93	45.25	33.86
1050	97.19	50.02	37.42
1150	106.44	54.78	40.99
1250	115.70	59.55	44.55
1350	124.96	64.31	48.12
1450	134.21	69.07	51.68
1550	143.47	73.84	55.25
1650	152.73	78.60	58.81
1750	161.98	83.36	62.37
1850	171.24	88.13	65.94
1950	180.49	92.89	69.5
2050	189.75	97.65	73.07

The network characteristics of the circulating loop are:

$$\Delta p = H_{lift} + a \cdot V^2 \tag{12.17}$$

Constant $\mathbf{a}_C$ can be determined by using the design data for the pump and knowing that the lifting height is 10 m (from the blueprint). The constant is:

$$a_C = \frac{\Delta p - H_{lift}}{V^2} = \frac{25-10}{112.3^2} = 0.001189 \quad (12.18)$$

The flow rate and the head of circulating pump for different loads and ambient parameters are presented in Table 12.4.

The power and energy used by the circulating pump for given loads and ambient parameters are presented in Table 12.5. The total energy consumed by this pump is also presented.

Table12. 4: Flow Rate, Head and Speed of the Circulating Pump

Capacity [kW]	Load/time [%]	MAX	MEAN	MIN	MAX	MEAN	MIN
		L [kg/s]			H_C [m]		
950	4.17	87.93	45.25	33.86	19.19	12.43	11.36
1050	8.33	97.19	50.02	37.42	21.23	12.97	11.66
1150	12.5	106.44	54.78	40.99	23.47	13.57	12.00
1250	8.33	115.70	59.55	44.55	25.92	14.22	12.36
1350	8.33	115.70	64.31	48.12	25.92	14.92	12.75
1450	4.17	115.70	69.07	51.68	25.92	15.67	13.18
1550	4.17	115.70	73.84	55.25	25.92	16.48	13.63
1650	4.17	115.70	78.60	58.81	25.92	17.35	14.11
1750	4.17	115.70	83.36	62.37	25.92	18.26	14.63
1850	4.17	115.70	88.13	65.94	25.92	19.23	15.17
1950	16.67	115.70	92.89	69.50	25.92	20.26	15.74
2050	12.50	115.70	97.65	73.07	25.92	21.34	16.35

Table 12.5: Power and Energy Used by the Circulating Pump (8000 h/year)

Capacity [kW]	Power [kW]			Energy [kWh]
	Max	Mean	Min	Total
950	23.88	9.96	6.85	4044
1050	29.02	11.24	7.46	9357
1150	35.02	12.66	8.13	16228
1250	41.95	14.25	8.85	12470
1350	41.95	16.01	9.64	13278
1450	41.95	17.95	10.49	7092
1550	41.95	20.10	11.41	7584
1650	41.95	22.46	12.40	8121
1750	41.95	25.03	13.48	8709
1850	41.95	27.86	14.63	9351
1950	41.95	30.93	15.88	40170
2050	41.95	34.26	17.22	32389
			Annual [kWh]:	**168 794**

The shaded area in Tables 12.4 and 12.5 indicates the maximum flows of the pump. It is approximately 6.7 % higher than the power of the electrical motor, but it is acceptable because the electrical motor is normally oversized and can cover this overload.

(d) Similarly, for the supply water distribution network, the constant $\mathbf{a_S}$ can be found. It is as follows:

$$a_S = \frac{\Delta p - H_{lift}}{V^2} = \frac{55.6-20}{150^2} = 0.001582 \quad (12.19)$$

The results of the calculation of the flow rate, head, power and energy consumption for the supply water pump are presented in Tables 12.6 and 12.7.

Table 12.6: Flow Rate, Head and Speed of the Supply Pump

Capacity [kW]	Load/time [%]	MAX	MEAN	MIN	MAX	MEAN	MIN
		L [kg/s]			H_C [m]		
950	4.17	87.93	45.25	33.86	32.23	23.24	21.81
1050	8.33	97.19	50.02	37.42	34.94	23.96	22.22
1150	12.5	106.44	54.78	40.99	37.92	24.75	22.66
1250	8.33	115.70	59.55	44.55	41.18	25.61	23.14
1350	8.33	115.70	64.31	48.12	44.70	26.54	23.66
1450	4.17	115.70	69.07	51.68	44.70	27.55	24.23
1550	4.17	115.70	73.84	55.25	44.70	28.63	24.83
1650	4.17	115.70	78.60	58.81	44.70	23.22	25.47
1750	4.17	115.70	83.36	62.37	44.70	23.11	26.15
1850	4.17	115.70	88.13	65.94	44.70	22.98	26.88
1950	16.67	115.70	92.89	69.50	44.70	22.81	27.64
2050	12.50	115.70	97.65	73.07	44.70	22.61	28.45

Table 12.7: Power and Energy Used by the Supply Pump (8000 h/year)

Capacity [kW]	Power [kW]			Energy [kWh]
	Max	Mean	Min	Total
950	43.90	24.09	21.91	9213
1050	51.68	25.23	22.52	19977
1150	60.26	26.53	23.20	32610
1250	69.15	28.06	23.94	23625
1350	77.95	29.85	24.75	25622
1450	77.95	31.95	25.66	13308
1550	77.95	34.42	26.67	13870
1650	77.95	26.51	27.81	12363
1750	77.95	26.68	29.08	12481
1850	77.95	26.79	30.53	12599
1950	77.95	26.83	32.16	50837
2050	77.95	26.81	33.99	38473
			Annual [kWh]:	**264 977**

(e) For the fan, only the flow rate and fan motor power are known. In this case, the fan's characteristics can be used to estimate the power by changing the flow rate. Of course, measurements can be performed in order to establish the performance curves, but this is deemed to be unnecessary.

Table 12.8: Flow Rate, Head, Speed, Power and Energy of the Fan

Capacity [kW]	Load/time [%]	MAX	MEAN	MIN	MAX	MEAN		Energy [kWh]
		G [kg/s]			Power [kW]			
950	4.17	72.07	37.09	27.75	7.55	1.03	0.43	738
1050	8.33	79.66	41.00	30.67	10.19	1.39	0.58	1992
1150	12.5	87.25	44.90	33.60	13.39	1.83	0.76	3927
1250	8.33	94.84	48.81	36.52	17.20	2.35	0.98	3361
1350	8.33	94.84	52.71	39.44	17.20	2.95	1.24	3638
1450	4.17	94.84	56.61	42.36	17.20	3.66	1.53	1982
1550	4.17	94.84	60.52	45.29	17.20	4.47	1.87	2167
1650	4.17	94.84	64.43	48.20	17.20	5.39	2.26	2378
1750	4.17	94.84	68.33	51.12	17.20	6.43	2.69	2615
1850	4.17	94.84	72.24	54.05	17.20	7.60	3.18	2881
1950	16.67	94.84	76.14	56.97	17.20	8.90	3.73	12704
2050	12.50	94.84	80.04	59.89	17.20	10.34	4.33	10510
							Annual [kWh]:	**48 893**

(f) Finally, after installing the new electrical motor and transmission and after implementing the VSD control system, the electrical energy consumption will be:

$$E_{new} = 168{,}794 + 264{,}977 + 48{,}893 = 482{,}664 \ \ [kWh] \tag{12.20}$$

The expected electrical energy consumption reduction will be:

$$\Delta E = \frac{E_{New} - E_{Current}}{E_{Current}} \cdot 100 = \frac{482{,}664 - 1{,}356{,}240}{1{,}356{,}240} \cdot 100 = 64.4 \ \ [\%] \tag{12.21}$$

Step 9: Financial Evaluation
(All prices in this chapter can be used only as an illustration. Current prices should be obtained from the manufacturers)

The unit price of electrical energy is 0.045 c$US/kWh (this price is calculated by taking the total annual electrical energy cost and dividing it by the total electrical energy consumption). The cost of electrical energy is then:

$$CE_{Curreent} = 0.045 \cdot 1{,}356{,}240 = 61{,}030.10 \ \ US\$/\,year \tag{12.22}$$

After the implementation of the proposed energy conservation measures the cost of electricity will be as follows:

$$CE_{New} = 0.045 \cdot 482{,}664 = 21{,}720 \ \ US\$/\,year \tag{12.23}$$

The expected cost reduction for running the cooling system is **39 310 $US/year**.

The estimated investment costs are as follows:

<u>DIRECT COSTS (DC)</u>

1. Onsite costs

Purchased-equipment costs (PEC)	
– VSD control units for:	
Circulating pump (39.33 kW)	6200 $US
Fan of cooling tower (14.7 kW)	3000 $US
Supply pump (115.5 kW)	16 500 $US
– Temperature sensors	500 $US
– Others	2500 $US
Purchased-equipment installation (33 % of PEC)	8500 $US
Piping (5 % of PEC)	1300 $US
Electrical materials (10 % of PEC)	2500 $US

Total onsite costs: 38 500 $US

2. Offsite costs

Civil, structural and architectural work	1500 $US
Service facilities	1000 $US

Total offsite costs: 2500 $US

TOTAL DIRECT COSTS:	**41 000 $US**

INDIRECT COST (IC)

Engineering and supervision (8 % of DC)	3200 $US
Construction costs (15 % of DC)	6100 $US
TOTAL INDIRECT COSTS:	**9300 $US**
TOTAL CAPITAL INVESTMENT (TCI = DC + IC):	***47 800 $US***

COST-BENEFIT ANALYSIS

$$\text{Simple Pay - Back Period} = \frac{47{,}800}{39{,}310} = 1.2 \text{ years} \tag{12.24}$$

Notation:

- A total area of the wetted surface (includes the surface area of drops of water as well as the wetted slates or other fill material), [m^2]
- a cooling tower approach, [^{o}C]
- c_p specific heat of fluid at constant pressure
- G mass flow rate of air, [kg/s]
- h enthalpy, [kJ/kg]
- h_c convection coefficient, [kW/(m^2 K)]
- L mass flow rate of water, [kg/s]
- M mass flow rate, [kg/s]
- NTU number of heat transfer units
- r cooling tower range, [^{o}C]
- RH Relative humidity of air, [%]
- T water temperature, [^{o}C]
- t air temperature, [^{o}C]
- Q heat capacity of cooling tower, [kW]
- η_{CT} efficiency of cooling tower, [-]

Subscripts:

- w refers to water
- a refers to air
- db refers to dry bulb
- wb refers to wet bulb
- in refers to inlet
- out refers to outlet

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