PERIODICA POLYTECHNICA SER. MECH. ENG. VOL. 35, NO. 4, PP. 217–234 (1991)

ENERGY SAVING AND ENVIRONMENTAL PROTECTION AT RECIRCULATING COOLING WATER SYSTEMS

J. HALÁSZ[1], P. GŐSI[2] and P. KOSTKA[2]

[1]University of Campinas (UNICAMP)
[2]Institute for Electric Power Research (VEIKI)

Received: January 20 1992.

Abstract

This paper deals with the problems of energy saving and environmental aspects (water consumption, drift emission, noise level) of cooling systems using mechanical draft wet cooling towers. Graphs have been developed to represent correlation between cooling performance, size of cooling towers, and fan power requirement for the most common operating range. Tracing of distinct cold water temperature is shown, below which reduction of fan power is advisable. Generalized charts are presented to provide values on cooling tower evaporation loss and make-up water demand.

Keywords: Wet cooling tower, mechanical draft, energy saving, environmental protection.

Introduction

Rejection of waste heat to ambient air is often necessary in various fields of industry. In some cases this heat dissipation is directly related to different processes: in power plants at steam condensing; at refrigeration cycles and air conditioning systems at refrigerant vapor condensing; in the chemical industry at cooling of various liquids and gases. Sometimes it is necessary to reduce the temperature of certain structural elements, e. g. at furnaces or at injection moulds, etc. Very often these problems can be solved in the most economic way by cold water systems using wet cooling towers of mechanical draft type (with fans) (*Fig. 1*).

The most important requirements for cooling systems operating with recooled recirculated water are as follows:

- low cold water temperature
- low investment costs
- operation in accordance with required cooling demand
- moderate electric energy consumption
- high cooling capacity
- moderate required head for pumping
- small make-up water demand
- low water treatment cost

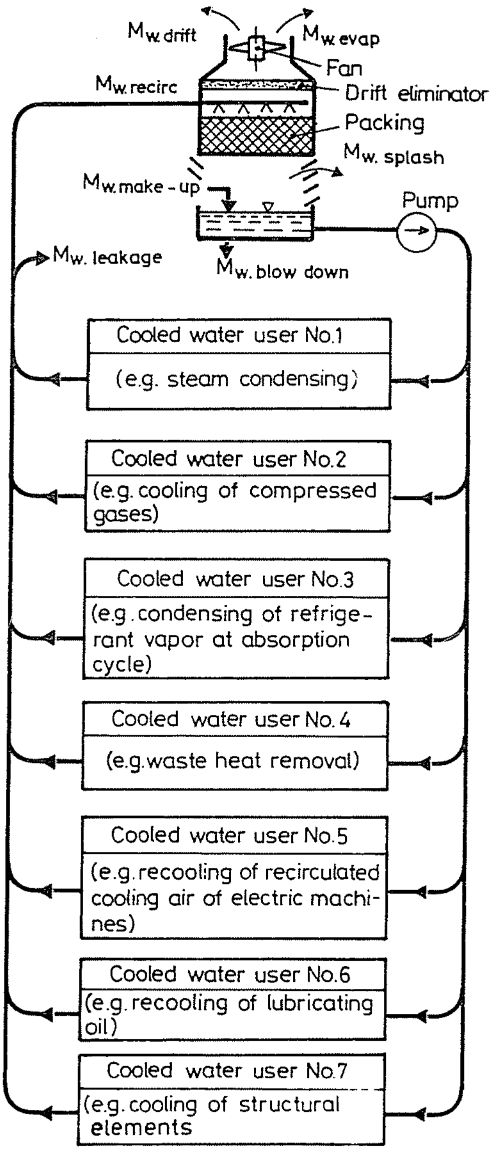

Fig. 1. Scheme of a cooling cycle with mechanical draft cooling tower supplying cold water for different users

- minimum maintenance demand
- safe winter operation
- long durability

– low noise level
– minimum drift loss, moderate fogging
– esthetic appearance.

Energy saving operation and environmental problems have an increasing importance in the field of recirculating cooling water systems, too.

Correlation between Cooling Performance, Size and Fan Power of Cooling Towers

In order to choose the most economic cooling system it is necessary to perform optimum calculations, simultaneously considering the process using cold water and a number of variations of cooling systems. This method is frequently used at power plant calculations, but it is less common at different industrial (process) cooling systems. The reason is that investigation is neglected between the correlation of energy consumption of complex processes and the enthalpy of ambient air [1]. In several cases cooling tower manufacturers do not provide enough data for clients to allow elaborate approximate calculations for choosing the most adequate equipment and operational mode.

On the basis of many calculations diagrams are provided to demonstrate correlation between the variation of cooling performance, size and fan power of mechanical draft cooling towers in the practically possible operational range.

Fig. 2 demonstrates the variation of counterflow cooling tower packed area as a function of $(a/\Delta t_w)$ approach/cooling range. If $a/\Delta t_w = 1$ the packed basis area is 100 %. The variation of the packed area has been determined by maintaining air velocity constant belonging to the 100 % basis area while changing approach and cooling range ratio.

Considering that investigation has been carried out on the relative variation of the area, the velocity of the air may vary with wide range of limits, the absolute value of it will not influence this ratio. (As a consequence, it is possible to find all variations of different cooling ranges, packings and ambient wet bulb temperatures at constant air velocity and barometric pressure in the area marked by the border curves). It can be seen from the diagram that one has to be careful in choosing lower values of the approach/cooling range ratio. Cooling tower size and price increase too rapidly in this area. (E. g. by choosing the approach/cooling range ratio instead of the value 1 to be 0.6, the necessary area can be greater by minimum 40 % but sometimes even by 120 %).

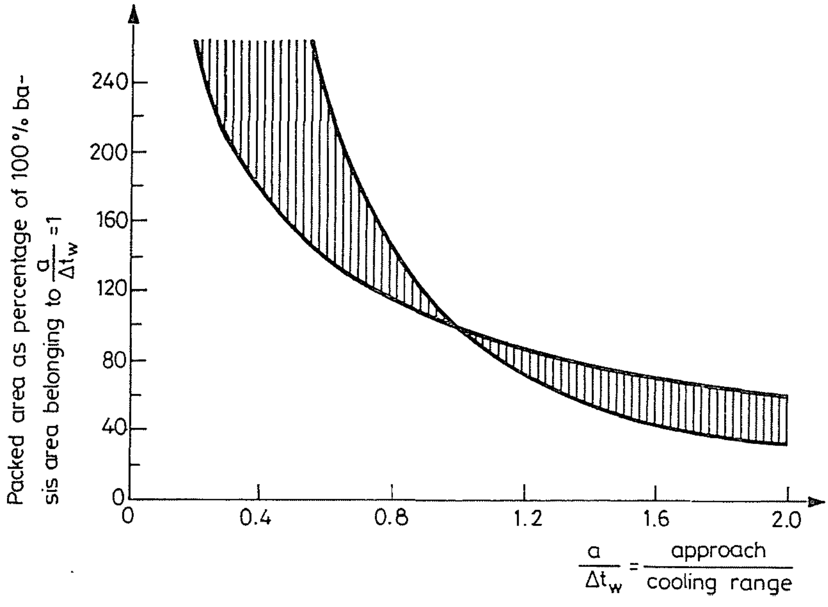

Fig. 2. Variation of cooling tower area as a function of approach/cooling range (Characteristic data of the tested region: $p_{\rm bar}$ = 1.0bar; $5.0 \leq t_{wb} \leq 30$ °C; $3.0 \leq \Delta t_w \leq 21.0$ °C; $1.2 \leq C \leq 3.0$; $0.5 \leq m \leq 0.9$).

In *Fig. 3* one can find values of 100 % basis areas belonging to different approach/cooling range ratios. Decreasing the approach/cooling range ratio from 0.6 to 0.4 there is an increase of 25 to 65 % in the necessary area. On the other hand, by increasing the approach/cooling range ratio from 0.6 to 0.8 the original basis area can be reduced to 65–85 %.

Fig. 4 shows the correlation between fan power and cooling tower performance by varying the cooling capacity through changing fan speed of an already operating cooling tower. Cooling performance has been expressed also by the value of the approach/cooling range ratio. Similarly to the diagram at *Fig. 3,* the values of the 100 % basis power consumption belong to the 0.6; 0.8; 1; 1.2; 1.4 and 1.6 values of approach/cooling range ratio.

For example, decreasing fan speed at constant water flow rate and cooling range, and increasing the value of approach/cooling range ratio from 0.8 to 1.0 (e. g. from 8 °C/10 °C to 10 °C/10 °C) the approach and the cold water temperature will increase by 2 °C. Fan power, however, will be maximum 60 % of the original value.

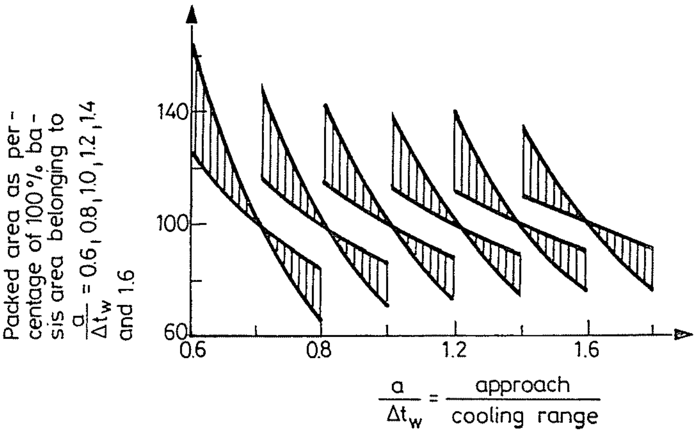

Fig. 3. Variation of cooling tower area as functions of 100 % basic areas belonging to different approach/cooling range values. (Characteristic data of the tested region: $p_{\text{bar}} = 1.0$ bar; $5.0 \leq t_{wb} \leq 30$ °C; $3.0 \leq \Delta t_w \leq 21.0$ °C; $1.2 \leq C \leq 3.0$; $0.5 \leq m \leq 0.9$).

Cooling capacity and fan power demand of obsolete cooling towers can be made more favorable by installing more efficient packing, water distribution system and drift eliminators of lower pressure drop. Such upgrading can be characterized by a ratio U:

$$U = \frac{M_{w,\,\text{recirc},\,u}}{M_{w,\,\text{recirc},\,o}}\,. \tag{1}$$

The denominator of this ratio is the original water flow, and the numerator is the increased water flow rate which can be cooled down by the upgraded tower to the same temperature (at unchanged cooling range, fan power and inlet air properties) as the original water flow was cooled to by the tower before upgrading. U can reach the values of 1.3 to 1.4 by upgrading old, obsolete cooling towers. Modernized cooling towers can be operated with smaller fan power to cool the water flow rate to the original temperature, or with the original fan power to produce lower cold water temperature. *Figs 5* and *6* show characteristics of the two possibilities. *Fig. 5* shows fan power decrease as a function of parameter U. *Fig. 6* supplies data for the case of constant fan power after upgrading. In this case cooled water temperature, and thus, approach/cooling range ratio decrease as a function of U.

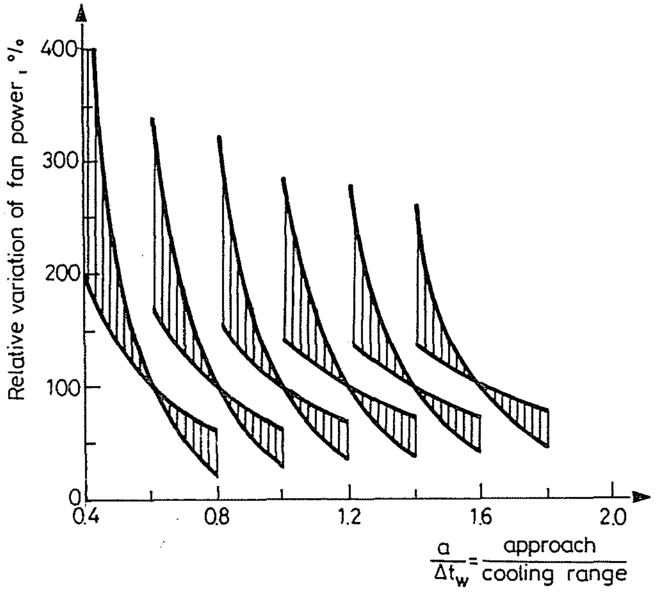

Fig. 4. Correlation of fan power and cooling performance (Characteristic data of the tested region: $p_{\text{bar}} = 1.0\,\text{bar}$; $5.0 \leq t_{wb} \leq 30$ °C; $3.0 \leq \Delta t_w \leq 21.0$ °C; $1.2 \leq C \leq 3.0$; $0.5 \leq m \leq 0.9$).

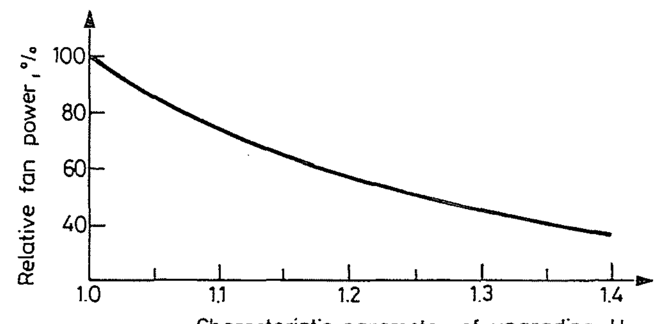

Fig. 5. Fan power variation with characteristic parameter of upgrading, U. (Characteristic data of the tested region: $p_{\text{bar}} = 1.0\,\text{bar}$; $5.0 \leq t_{wb} \leq 30$ °C; $3.0 \leq \Delta t_w \leq 21.0$ °C; $1.2 \leq C \leq 3.0$; $0.5 \leq m \leq 0.9$).

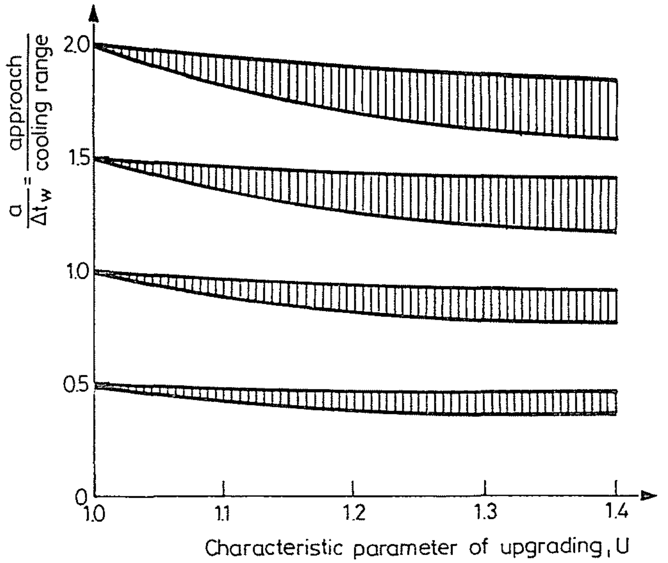

Fig. 6. Approach/cooling range variation as a function of U at constant fan power before and after upgrading (Characteristic data of the tested region: $p_{\mathrm{bar}} = 1.0\,\mathrm{bar}$, $5.0 \le t_{wb} \le 30\ ^\circ\mathrm{C}$; $3.0 \le \Delta t_w \le 21.0\ ^\circ\mathrm{C}$; $1.2 \le C \le 3.0$; $0.5 \le m \le 0.9$).

Energy Saving Operation

Although there is a unequivocal correlation between the cooled water temperature produced by the cooling tower and the costs or savings in the processes using the cold water, it can only be expressed as functions of numerous parameters. For instance, with increasing condensing temperature, turbine electric power output is decreasing, or compressor power demand of a refrigeration cycle is increasing. Cost variation characteristics of such installations are illustrated in *Fig. 7*. The curve presented is valid for a number of constant parameters like constant inlet steam flow rate and steam properties at turbine cycles. Energy consumption and often the amount of the most valuable fractions at several chemical processes depend on the cooling water temperature. To avoid cold water temperature increase above certain critical limits greater energy consumption is necessary in some cases. (E. g. to achieve lower cold water temperature cooled water is further refrigerated with vapor compression or absorption systems, or in order to decrease the outlet temperature of the circulated water in some installations, water velocity and volume flow rate is increased.)

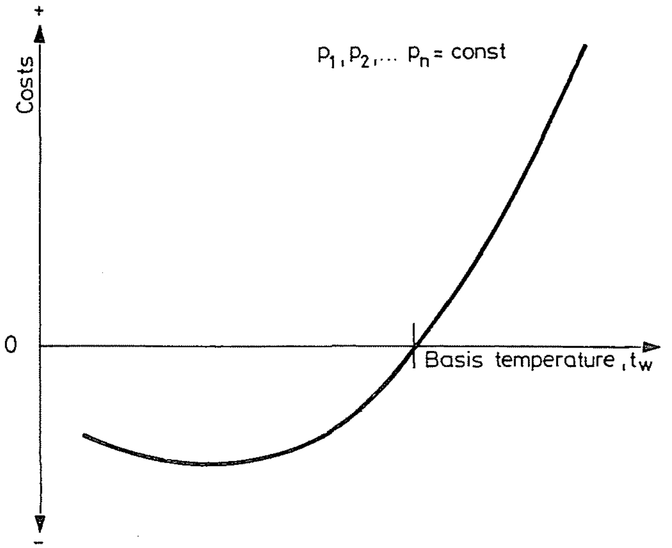

Fig. 7. Costs versus cold water temperature

Correlation between costs and cold water temperature of cooling systems providing cold water for various different processes is similar to the curve shown in the upper part of *Fig. 8.* In most cases this curve has various break points in the region of lower temperatures [1]. Distinct cold water temperature can be determined from the diagram (valid for a given series of parameters), below which it is advisable to decrease cooling tower fan power, and above which it is more economic to run cooling tower fans at maximum power consumption. The method to find this distinct cold water temperature is shown in *Fig. 8.*

In the upper part of the figure cost versus cold water temperature is shown. Positive cost values are presented above basis temperature t_b, while negative values (savings) are located below t_b. In the lower part of the figure cooling tower fans' operational costs are presented as a function of cold water temperature at various ambient air wet bulb temperatures and fan revolutions assuming continuous (e. g. adjustable frequency drive) speed control. Distinct temperature t_d appears where incremental cost saving ΔK_{fan} due to decreased fan speed compared to the original 100 % fan speed (in the figure at $t_{wb} = 15$ °C wet bulb ambient air temperature) is equal to the incremental cost increase $\Delta K_{\text{process}}$.

Savings due to fan speed control at lower than t_d distinct temperature can be determined from the diagram. Considering t_{wb} = 5 °C wet bulb ambient air temperature and fan operation at 100 % speed, cold water

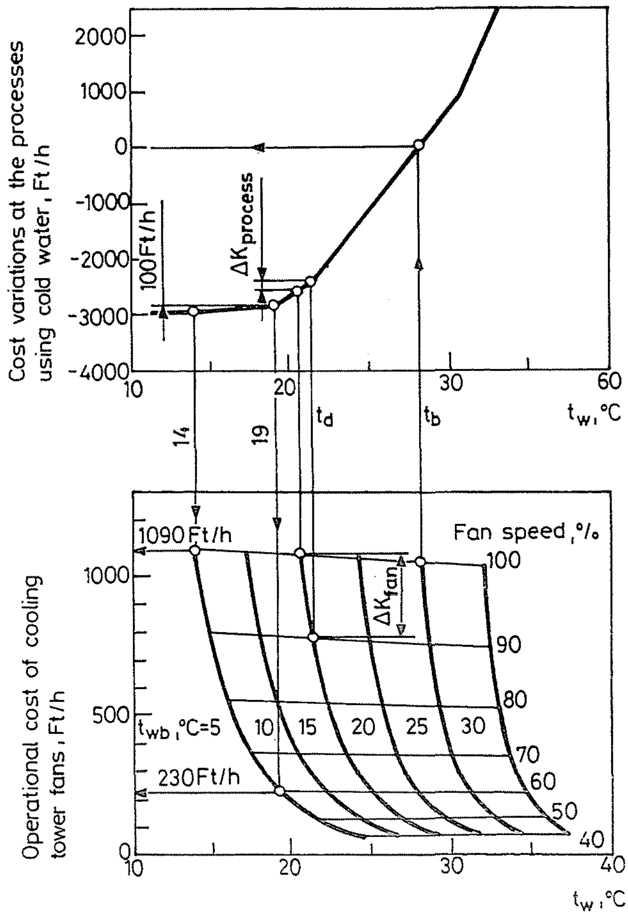

Fig. 8. Determination of distinct cold water temperature t_d ($\Delta K_{\text{process}} = \Delta K_{\text{fan}}$)

temperature is 14 °C. However, fan speed should be decreased if water temperature is lower than distinct temperature t_d. Fans operated at 60 % speed provide 19 °C cold water temperature instead of 14 °C. According to the figure the extra cost in the process due to temperature increase is only 100 Ft/h. On the other hand, fan power consumption cost will change from 1090 Ft/h to 230 Ft/h due to speed decrease from 100 % to 60 %. Thus, economy will be $1090 - (230 + 100) = 760$ Ft/h.

The greatest energy saving and the longest life of rotary machines can be achieved by continuous adjustable frequency speed control. Investment costs are often returned within two years [2]. Similar, but somewhat less favorable is the use of variable pitch fans whose fan blade pitch can be continuously changed during fan operation. Considerable energy saving can be accomplished by operating the optimum number of fans in cooling towers having more cells and fans. Systems having one fan only should be provided with at least two speed fan motors. Energy consumption of an off and on cycling operation is higher considerably and this method shortens the life of rotary structural elements, too [5].

Cooling system operation at optimum performance requires cold water flow rate adjustment followed by cooling range variations as well. Cold water flow rate change can be accomplished by stopping one or more pumps or by some other control methods. Too low velocities, however, should be avoided (especially in critical parts of heat exchangers) because they allow the suspended solids to settle out.

Different distinct cold water temperatures t_d belong to different cold water flow rates and cooling ranges. Therefore, in controlling fan speed, actual water flow rate should be considered, too.

To achieve energy saving operation of a cooling system, fouling of heat exchange surfaces should be avoided (or at least reduced to an acceptable level). Fouling is caused by sedimentation of dissolved and suspended solids, biological impurities (algae, bacteria, fungi) and corrosion. Water treatment, filtration and chemical cleaning play an important role. On-load, continuous cleaning of heat exchangers has a special importance, too. Two on-load cleaning methods are used the most frequently: cleaning by balls or by brushes. *Fig. 9* provides experimental data of possible improvements of 4.5 MW turbine condensers [3]. Tests were carried out in a chemical plant on two identical heat exchangers. Removing fouling continuously from the tubes by a cleaning brush system, heat transfer coefficient of the condenser even improved during the first 30 days, then remained at a constant level. At the same time heat transfer decreased and water pressure drop increased considerably at the reference (not cleaned) heat exchanger as a function of time. Cold water was produced by cross flow cooling tower using filtered river water as make-up water. If instead of continuous cleaning periodical cleaning is applied, then the optimum intervals and times of cleaning will result in lower but still significant energy savings, too.

As a summary, it can be stated that cooling systems need greater attention from the point of view of energy saving as well. A few measurements and calculations carried out on a cooling system can provide data on simplified energy saving operation, on upgrading and maintenance, too.

Continuous computer analysis is needed, however, to optimum operation of more complicated systems.

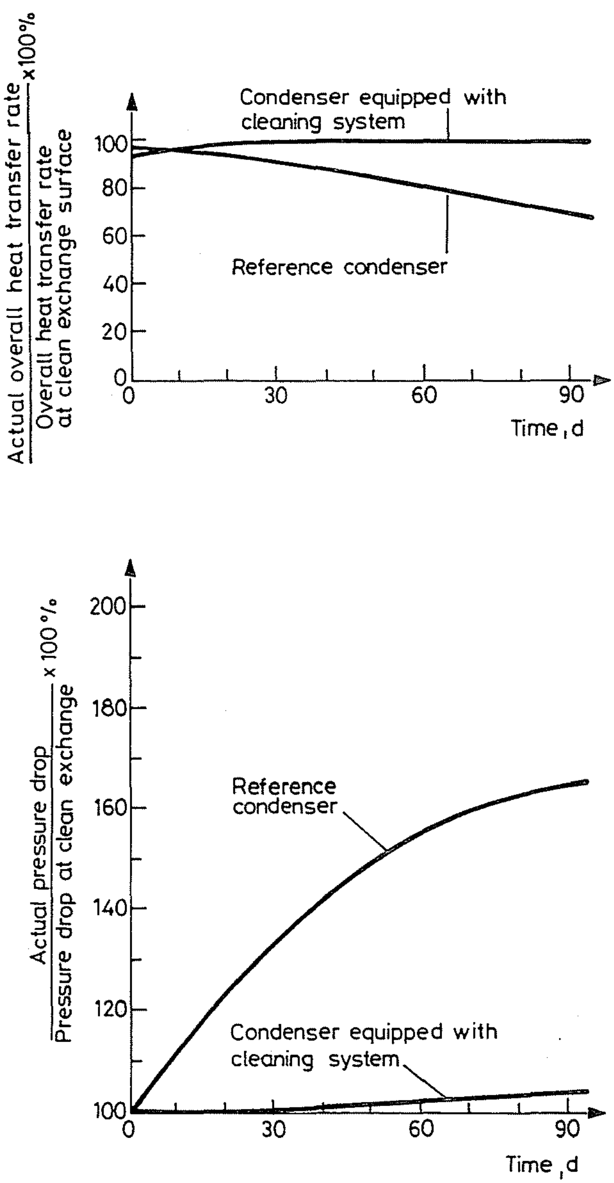

Fig. 9. Heat transfer and pressure loss at a condenser equipped with on-load brush cleaning system and at a reference condenser (measured values).

Environmental Protection

Water Consumption

Make-up water quantity fed to wet cooling tower systems has special importance from the point of view of water saving and the quality of recirculated water. Water loss of up-to-date cooling systems is mostly due to evaporation loss. There are much less losses from drift, splash out, leakage and blow down. (The value of blow down may be reduced to zero).

Concentration increase of dissolved solids due to evaporation should be avoided by eliminating impurities from the recirculated water.

Three methods are used generally:

a) Continuous blow down
b) Full or side stream treatment of recirculated water
c) Combination of methods a) and b)

Determination of concentration ratio has a special importance in qualifying cooling systems. Concentration ratio ε can be calculated generally in the simplest way from the Cl^- content of the make-up and recirculated water:

$$\varepsilon = \frac{\text{Cl}^- \text{ concentration of the recirculated water}}{\text{Cl}^- \text{ concentration of the make-up water}} . \tag{2}$$

The concentration ratio can also be expressed by evaporation, blow down and other mass flow rates of water losses:

$$\varepsilon = 1 + \frac{M_{w,\text{evap}}}{M_{w,\text{blow down}} + M_{w,\text{drift}} + M_{w,\text{splash}} + M_{w,\text{leakage}}} . \tag{3}$$

Evaporation loss and make-up water quantity of wet cooling towers are generally expressed as percentage of the recirculated water mass flow rate. Evaporation loss depends on inlet ambient air properties, on cooling range and on air to water ratio and it varies within a wide range.

Fig. 10 shows a generalized chart [4] providing approximate evaporation loss of any cooling tower in the most common range of operation. Two examples help to illustrate the use of the chart. (First, state of the inlet air has to be marked in the upper left side of the chart.) Make-up water quantity can be read from *Fig. 11* as a function of known evaporation loss and concentration ratio. E. g., considering 1.5 % evaporation loss and a frequently used $\varepsilon = 4$ value of concentration ratio, make-up water quantity is 2 %. This means that blow down should be about 0.5 % including leakage, splash out and drift losses.

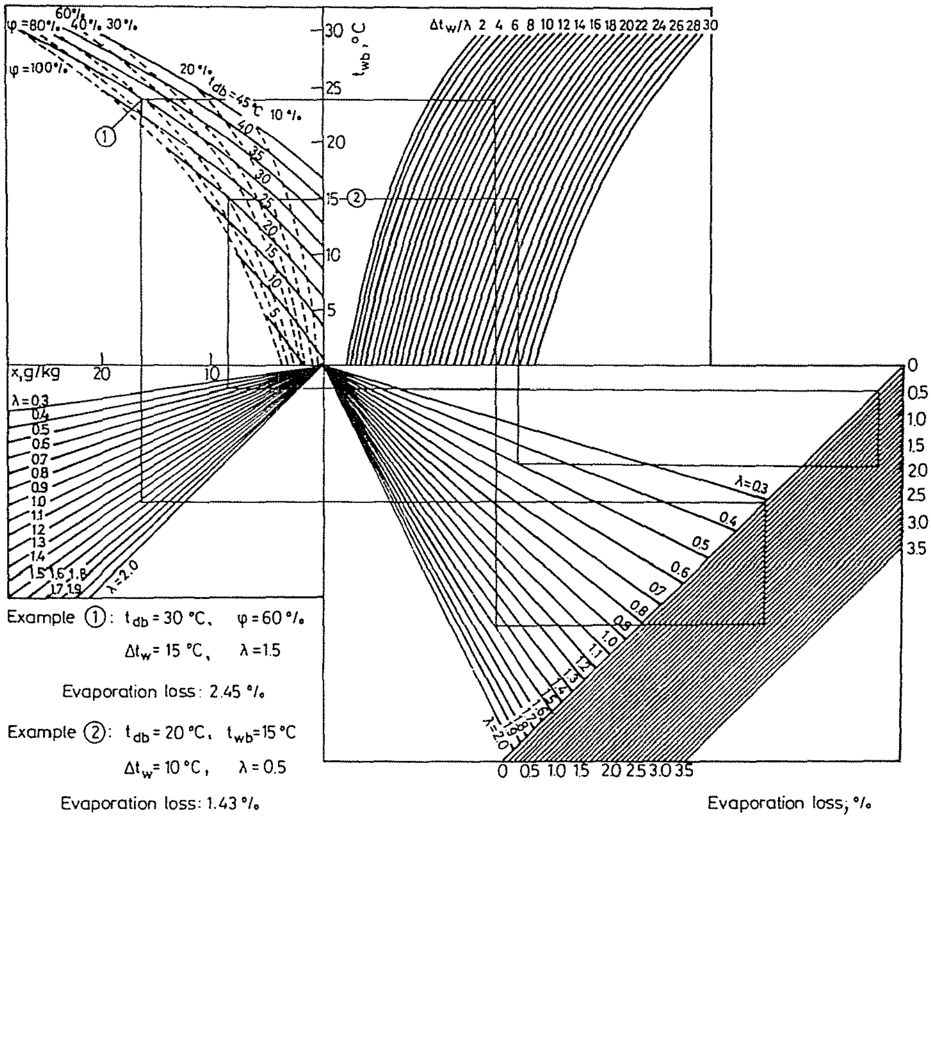

Fig. 10. Generalized chart for the determination of evaporation loss of cooling towers [4]

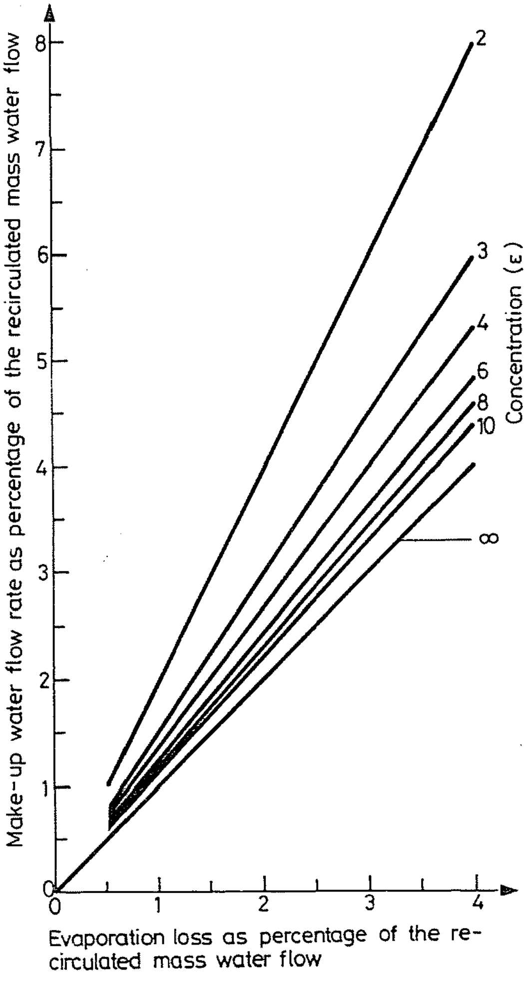

Fig. 11. Make-up water flow as functions of evaporation loss and concentration ratio

Theoretically, water consumption of wet cooling towers could be decreased to evaporation loss. *Fig. 12* shows the ratio of the actual make-up water quantity and evaporation loss as a function of concentration ratio. Increasing water costs make it necessary to operate cooling systems at higher concentration ratio values. For this, splash out, leakage and drift losses should be practically eliminated and adequate water treatment should be provided [6].

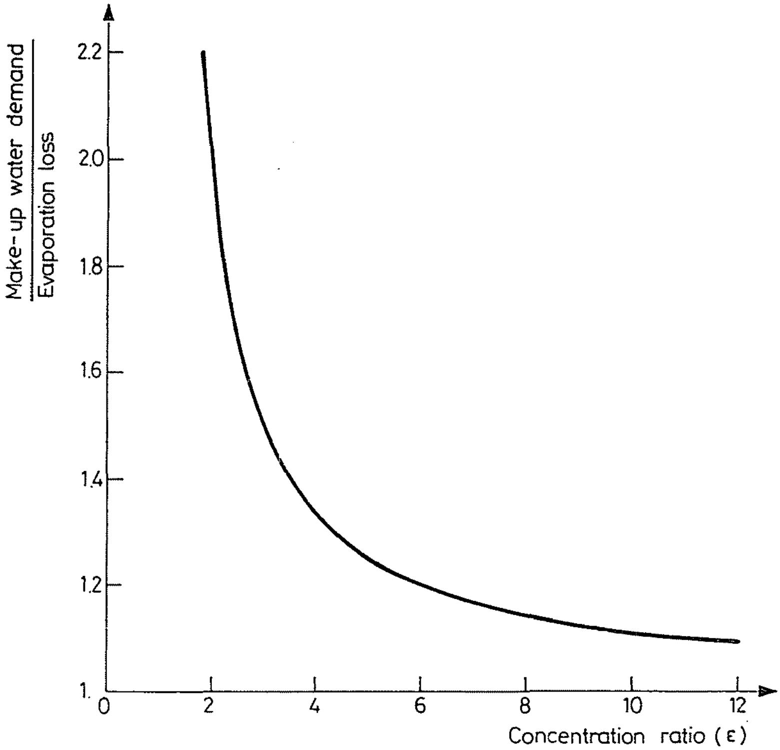

Fig. 12. Ratio of make-up water flow and evaporation loss as a function of concentration ratio

Drift Losses

Efficient drift eliminators are necessary elements of up-to-date cooling towers. Their functions are the following:

- water consumption reduction
- withholding of chemicals and other impurities in the water in order to decrease environmental pollution
- protection of the cooling tower's surroundings from drift, fast corrosion and icing
- reduction of fan blade erosion.

The most important requirements of drift eliminators are as follows:

- efficient drift elimination
- low pressure drop
- long life expectancy

- simple installation
- reasonable cost.

Drift eliminators have been significantly improved in the last decades. Earlier it was a common practice to keep drift losses of mechanical draft cooling towers under 0.2 % of the recirculated water flow rate. Today, modern devices provide a maximum of 0.02 % or even smaller values of drift losses at significantly lower pressure drops.

In addition to controlling drift carried by tower air flow, attention should be paid to decrease splash out of water at the air inlet openings of the towers, especially during windy weather, This problem can be solved by using adequate louvers.

Noise Level

Low noise level in cooling tower surroundings has an increasing importance. As a basic rule it is in most cases much more economic to install lower noise level equipment, even if it is more expensive, than to reduce high noise level of an equipment later [2]. Low noise level of larger installations can be achieved by high efficiency, low speed axial fans having a relatively great number of blades. Speed adjustment is important not only because of energy saving, but also because of noise level reduction, e. g. noise level can be reduced during the night hours.

Conclusions

Size, cost and fan brake horsepower of mechanical draft wet cooling towers increase steeply at values less than 1 of approach/cooling range. Therefore, special care should be taken in choosing the value of this ratio. The fan power consumption of renovated obsolete cooling towers can often be reduced by 50–60 % maintaining the cooling capacity. In many cases, however, it is advisable to keep fan power consumption at the original or at even higher value to provide lower cold water temperature at peak demand.

Cold water distinct temperature below which fan power consumption should be decreased can be determined in a rather simple way even in cooling systems supplying cold water for different processes. Greatest energy savings and longest life of rotary equipments can be achieved by adjustable frequency control. In order to operate systems at optimum performance, water flow rate can be changed as well.

To avoid sedimentation of suspended impurities flow rates should be kept above critical values. There are significant losses due to surface fouling in heat exchangers of the cooling circuit. To avoid these losses filtration, chemical water treatment and on-load cleaning of heat exchangers can be applied.

Water consumption, drift and noise level of wet cooling towers should be taken into consideration from the aspect of environmental protection. Due to high water costs low concentration ratios should be avoided. Up-to-date drift eliminators with low pressure drop and efficient louvers at the air inlet openings should be used. Leakages and all kinds of water losses should be eliminated, furthermore, efficient water treatment should be applied. The most efficient noise level reduction of larger induced draft equipments can be achieved by low (and adjustable) speed axial fans with relatively great number of blades.

Nomenclature

$M_{w,\text{recirc}}$	recirculated mass water flow
$M_{w,\text{evap}}$	evaporated mass water flow
$M_{w,\text{drift}}$	mass water flow lost form the tower in the from of fine droplets entrained in the exhaust air
$M_{w,\text{splash}}$	mass water flow splashed from the tower at the air inlet openings
$M_{w,\text{make-up}}$	mass water flow of make-up water
$M_{w,\text{blow down}}$	mass water flow of blow down water
$M_{w,\text{leakage}}$	mass water flow through leakages
$M_{w,\text{recirc},o}$	recirculated mass water flow before upgrading
$M_{w,\text{recirc},u}$	recirculated mass water flow after upgrading, which can be cooled down by the upgraded tower to the same temperature (at unchanged cooling range, fan power and inlet air properties) as the original water flow $M_{w,\text{recirc},o}$ was cooled to by the tower before upgrading
t_w	cold water temperature
t_{wb}	ambient air wet bulb temperature
a	approach, difference between cold water and ambient air wet bulb temperatures
Δt_w	cooling range, difference between hot water and cold water temperatures
t_d	cold water distinct temperature (see *Fig. 8*)
t_{db}	ambient air dry bulb temperature
p_{bar}	atmospheric pressure
x	absolute humidity of inlet air
φ	relative humidity of inlet air
v	air velocity at cooling tower plan packed area
P_{fan}	fan power consumption

n	fan speed
$p_1, p_2, \ldots, p_n$	parameters (steam flow of turbines, steam temperature, pressure)
Me	Merkel number
λ	air to water ratio
C	constant in the equation M= $C\lambda^m$ describing cooling tower characteristic curve
m	exponent in the equation M= $C\lambda^m$ describing cooling tower characteristic curve
ε	concentration ratio (ratio of the impurities in the circulating water and the impurities in the make-up water
$U = \frac{M_{w,\mathrm{recirc},u}}{M_{w,\mathrm{recirc},o}}$	characteristic parameter of upgrading
ΔK_{fan}	change in fan operational costs (see *Fig. 8*)
$\Delta K_{\mathrm{process}}$	cost change in the process due to change in cold water temperature

References

1. CAMPAGNE, W. v. L. – MC DONOUGHM L. J.:, How Cooling Towers Affect Process Energy Savings. *Hydrocarbon Processing,* June 1984.
2. FONSTAD, K. A.: Energy Savings Through Control of Cooling Tower Air Flow. *Energy Engineering*, Vol. 85. No. 3, 1988.
3. FODOR, L. – GŐSI, P. – KOSTKA, P. – TÓTH, J.: Üzem közbeni, kefés rendszerű mechanikus hőcserélő tisztítás, *VEIKI Közlemények*, 1990.
4. GŐSI, P.: Method and Chart for the Determination of Evaporation Loss of Wet Cooling Towers. *Heat Transfer Engineering*, Vol. 10. pp. 44–49 No. 4. 1989.
5. HENSLEY, J. C.: Cooling Tower Energy. *Heating/Piping/ Air Conditioning.* October 1981.
6. ILLÉS, I. – KELEMEN, L. – ÖLLŐS, G.: Ipari Vízgazdálkodás, VIZDOK, Budapest, 1983.

Addresses:

Péter GŐSI
Institute for Electric Power Research (VEIKI)
1051 Budapest, Zrinyi u. 1. Hungary

Judit HALÁSZ
University of Campinas
13081 Campinas, UNICAMP S. P. Brazil

Pál KOSTKA
Institute for Electric Power Research (VEIKI)
1051 Budapest, Zrinyi u. 1. Hungary