THEORETICAL POSSIBILITY OF THE MAISOTSENKO CYCLE APPLICATION TO DECREASE COLD WATER TEMPERATURE IN COOLING TOWERS

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In any explanation of cooling tower operation, it is shown that the ambient wet bulb temperature is a physical limit for water cooling. The main advantage of the Maisotsenko cycle (M-cycle) is the possibility to have the cold water temperature lower than the ambient wet bulb temperature, without violation of any physical principle. The idea is based on the fact that the ambient wet bulb temperature depends not only on the absolute humidity value, but also on a current air temperature. On the contrary, the dew point temperature depends only on the absolute humidity value. If we can provide lower temperature of the air (with the same absolute humidity) at the air inlet of the cooling tower, then the wet bulb temperature for such air will be lower. Therefore, if we can cool the air entering the cooling tower, without the change of its absolute humidity, it will be possible to cool the water more and, for some conditions, its temperature could be lower than the wet bulb temperature of the surrounding air at some distance from the cooling tower. However, the main problem is how to cool the air entering the cooling tower, very simply and without significant energy consumption. A theoretical possibility to solve this problem by application of the M-cycle is under analysis in this paper.

KEY WORDS: cooling tower, wet bulb, dew point, Maisotsenko cycle

1. INTRODUCTION

Air cooling methods due to moistening have been well known for centuries. However, saturated humid air, because of its high humidity, is not desirable for air conditioning systems or many technological processes. The idea to cool the air without adding water and without using any refrigerating equipment was proposed by one research group, a member of which was V. Maisotsenko, many years ago in the USSR (Tsimerman et al., 1976). At that time, the idea was called "indirect-evaporative cooling."

The following is a short description of the device, implementing such a cooling method. In some imaginary device there are two groups of channels. Incoming air
is divided into two flows. One flow is going through the channels, where it makes contact with the water and therefore its temperature decreases due to energy consumption during water evaporation. This airflow is called "working airflow." Through the other group of channels another airflow is moving. Its absolute humidity remains constant, while temperature decreases due to sensible heat transfer to the "working airflow" through the dividing walls; the outlet air temperature will be lower than the air temperature outside. This useful airflow is called "product airflow." Cooled by this method air may be supplied to an air conditioning system directly, or for further processing, or in any other equipment/system as cooling media.

The theoretical cooling limit for such a process, in case of the special channels’ design and organization of working and product flows, will be the dew point of the surrounding, nonprocessed air.

It is necessary to mention that during simple moistening of the surrounding air it is impossible to attain a lower temperature than the wet bulb temperature of that air, which is considered as the lowest limit during evaporative cooling for all cooling towers.

It is not difficult to explain why indirect-evaporative cooling allows this limit to be overcome. Wet bulb temperature depends on dry bulb temperature and absolute humidity. During an air temperature decrease in the constant humidity process the wet bulb temperature will also decrease. The example in Fig. 1 shows two moistening processes: along the line 1–1 wet bulb and along the line 1′–1′ wet bulb. For humid air conditions, corresponding to points 1 and 1’, the air humidity is the same. Therefore it becomes clear that the wet bulb temperature is a variable value for humid air with constant absolute humidity and depends on current air temperature. The dew point depends only on absolute humidity.

The idea of how to apply the indirect-evaporative cooling phenomenon was under development by different research groups, which may be seen from numerous patents (Tsimerman et al., 1977a,b, 1993, 1994; Gershuni et al., 1990; Maisotsenko and Gershuni, 1990).

However, the really "breakthrough" design was proposed by Maisotsenko (1995) and it was a practical step for implementation of this insightful idea to air conditioning systems. An operating commercial model of the air conditioning system based on the Maisotsenko design is described in the Coolerado Corporation brochure (2006).

In the scientific literature this process received the name the Maisotsenko cycle or M-cycle; even so it cannot be considered as a traditional cycle, because according to the definition, "a thermodynamic cycle consists of a series of thermodynamic processes transferring heat and work, while varying pressure, temperature, and other state variables, eventually returning a system to its initial state" (Cengel and Boles, 2002). However, we shall use generally recognized terminology for this phenomenon in the present article.

Research on heat exchanger design perfection is still ongoing (Maisotsenko et al., 2002, 2005a,b, 2007; Gillan et al., 2007). In numerous scientific papers physi-
cal aspects of the devices, operating on the M-cycle principle, were described in
detail and therefore all this information is not repeated here. For details one may
refer, for example, to Gillan (2008), Gillan et al. (2011), and Wani et al. (2012).
In the present paper the possibility of M-cycle application mainly for evaporative
cooling towers’ modification will be under analysis.

2. THEORETICAL POSSIBILITY OF M-CYCLE APPLICATION
TO EVAPORATIVE COOLING TOWERS

Practical technical device with detailed description for realization of M-cycle in
cooling towers is not under analysis on the current stage. Based on already-devel-
oped calculating schemas and existing software, an attempt is made to prove the

FIG. 1: Ramzin enthalpy-absolute humidity diagram with examples of air parameters
theoretical possibility to achieve lower temperatures of cooled water with the help of M-cycle application.

It should be noted that in a single operating cooling tower the only possibility to achieve the theoretical cooling limit, i.e., the wet bulb temperature, is to provide operation of the cooling device with very low water load and large heat-and-mass transfer area in the unit volume of the fill-and in any case the wet bulb temperature of the surrounding air will be the insuperable obstacle for water cooling.

On the stage of theoretical confirmation of M-cycle possibilities to enhance a cooling tower’s operation by decreasing the cooled water temperature, we will neglect the dimensions’ problem of any real heat-and-mass transfer device for product air processing. However, it should be kept in mind that in an air-to-air heat exchanger, where heat will be removed from product airflow to working airflow, the temperature difference between airflows depends on the heat transfer area and the heat exchanger’s efficiency.

It should be noted that in case of a conditioning system, cooling of the air without moistening and without application of refrigerating equipment looks very attractive. Such air may be supplied directly into living or industrial areas. Also it may face such further processing as additional cooling or drying/moistening. Due to M-cycle cooling the air may be organized practically without electricity consumption, and that fact significantly increases the entire system’s energy efficiency. It is obvious that for cooling of the working airflow it is necessary to organize some water supply for evaporation. However, evaporation is a very efficient process and due to the evaporating of 1 kg of water it is possible to remove from the air about 2.5 MJ of thermal energy. If for the same purpose a refrigerating system is used with the COP = 3 (EER = 10.23), then its electricity consumption will be 0.23 kWh. During these evaluations we are neglecting energy consumption of the water pump due to its small value.

In the case, for example, of the state of Arizona (USA) the cost of water for the commercial consumer is less than 0.1 cent/liter, while the cost of electricity in the day time in summer may be 17–24 cents/kWh. Therefore application of M-cycle to air cooling, and especially in conditioning systems, is economically reasonable. Water and electricity prices are available from the Internet websites of Tucson Water (2012) and APS (2012).

In the case of the cooling tower the goal is to cool the circulating water. If one succeeds to cool somehow the incoming air by 2–3°C without changing its absolute humidity, then the final temperature of the cold water at the exit of the cooling tower could be decreased by a considerable value. If the consumers of the cold water are power plant condensers, then it becomes possible to provide normal operation of the steam turbine in the hottest period. It is well known that the efficiency of the steam turbine decreases slightly when condensation temperature/pressure grows until some upper limit. However, when the said limit is exceeded, a significant drop of generating power is observed. Therefore, for such an operational situation the possibility of additional decrease of the cold water temperature
becomes crucial and extra spending to achieve that goal becomes economically reasonable.

In order to confirm the decrease of the cold water temperature in the cooling tower due to M-cycle application, calculations of a real cooling tower with an exhaust fan and with cell dimensions $8 \times 8 \times 9$ m and fill height 1.36 m were carried out. Air consumption for the chosen fan unit was 478,000 m$^3$/h, while the cooled water flow was 192 m$^3$/h.

Calculations were carried out with the help of software developed in our company, simulating the cooling tower operation. Reliability of this software was confirmed by good correlation of the calculation results and data from field tests of real cooling towers.

An assumption was made that incoming air with the certain initial parameters ($t_{air} = +25^\circ$C and relative humidity $\varphi = 43.6\%$) could be cooled without any change of its absolute humidity in some heat exchanger, based on M-cycle. Two variants were simulated, when air was cooled by 5$^\circ$C and by 10$^\circ$C (Figs. 2a and 2b).

**FIG. 2:** (a) Simulation results for additional cooling of incoming air by 5$^\circ$C, (b) simulation results for additional cooling of incoming air by 10$^\circ$C.
For the first variant, the cold water temperature practically reached the cooling limit and was obviously lower than the cold water temperature of the conventional cooling tower by a meaningful value of 1.4°C.

For the second variant, simulation results met the expectations: The cold water temperature at the exit of the fill was below the temperature limit of the conventional cooling tower, i.e., the wet bulb temperature of the unprocessed surrounding air, by 2°C. However, precooling of the incoming air to such a low temperature in some imaginary device might be an unrealistic task.

One should see that these encouraging results were received for very small water loads – 3 m³/m²⋅h. Based on the certain simulating software, to cool the water below the wet bulb temperature of the surrounding air for larger water flow rate’s values was impossible, though in all simulations the cold water temperature was lower than for a conventional cooling tower.

2.1 Principle Difference between Processes in Cooling Tower and Air Conditioning System

An air conditioning system and a cooling tower differ by their end product. In cooling towers the goal is to cool water and, therefore, cooling of the incoming air in the heat exchanger, working on the M-cycle principle, is an interim process before the water cooling heat transfer process starts. For the air conditioning systems a typical circulation factor is equal to 3 to 10 to 30 volumes of the ventilated zone. If the assumed area of the office or production factory is 100 m² with a ceiling height of 3 m, then the air rate will be 900–9000 m³/h. However, even for a small cooling tower with a cross section dimension 4 × 4 meters, i.e., 16 m², the air volume flow rate will exceed 70,000 m³/h.

For practical realization of the M-cycle the main component is a heat exchanger device in which heat flux transfers from product airflow to working airflow, and its design could be very complicated. In order to achieve lower product air final temperature, airflow may be divided several times: Initially all incoming air is divided into product and working flows; then, at the next stage, the product airflow is divided again into product flow and working flow with a visible decrease of product air volume flow rate. If necessary this division may be repeated again and again; however, the part of the useful product flow from the initially taken surrounding air will decrease significantly (Gillan, 2008). The required design for such a process will be very complicated.

It becomes clear that in case of a cooling tower to cool incoming air according to the M-cycle, even with one division of this incoming flow, it is necessary to have an air-to-air heat exchanger with high air volume flow rate performance.

In Fig. 3 there are three schemas, which utilize fundamentally different approaches to implement the M-cycle into a cooling tower design. The drawing in Fig. 3a shows the imaginary design of the fill, which permits the cooling of incoming air without any moistening before this air starts to work as cooling media for
FIG. 3: Cooling tower implementing M-cycle principle (a) with very complicated internal design of the fill, (b) with separated ventilation systems with exhaust fans for working airflow and for operation of the cooling tower, (c) with multipurpose forced-draft fans: 1 – air entrance for precooling; 1’ – entrance of the working airflow; 2 – air exit of the additionally cooled air; 3 – air-to-air heat exchangers similar to Coolerado air conditioners’ design; 4 – fill with general design; 4’ – fill with special design containing dry channels and traditional wet channels for water cooling; 5 – water distribution system; 6 – drift eliminator; 7 – water basin; 8 – main exhaust fan; 9 – additional exhaust fans for working airflow; 10 – forced-draft fans
water flow in the cooling tower. The air pass should be very complicated and the fill design is expected to be therefore very complex. A fan unit with special characteristics will be required for overcoming the very high aerodynamic resistance. In Fig. 3b the cooling tower has air-to-air heat exchangers of Maisotsenko type at the air inlets, while the rest of the cooling tower design is conventional. Additional fans are used for organization of the working airflow. In Fig. 3c exactly the same air distribution schema as was proposed by Maisotsenko (1995) is realized. However, for a cooling tower it may not work, because of the considerable aerodynamic resistance to airflow in the fill, especially when it is with water flow.

3. PHYSICAL PROBLEMS OF M-CYCLE APPLICATION TO THE COOLING TOWERS

During most of the year properly designed cooling towers can successfully cool water to the level required by technological process. Water cooling problems begin only in the hottest season.

On the Ramzin enthalpy-absolute humidity chart (Fig. 1) air conditions for the hot period of the year are shown by the following marks: Point 1 corresponds to a hot and dry zone, point 2 corresponds to the Moscow region, and point 3 is a representative condition for some regions of the northwest of Russia. For each point the wet bulb temperature is also shown (points 1′, 2′, and 3′). Each two points (1 and 1′, 2 and 2′, 3 and 3′) are assumed to have the same enthalpy, which is physically reasonable.

Therefore, in the ideal case, a heat exchanger, operating with implementation of the Maisotsenko cycle, may provide heat flux, limited by the temperature difference between dry bulb and wet bulb temperatures. It is obvious that to design such a perfect counterflow or crossflow air-to-air heat exchanger, in which product air could be cooled to the lowest temperature of the working airflow, i.e., wet bulb temperature, is impossible.

In recuperative heat exchangers (it should be mentioned that heat transfer between channels with product air and channels with working air will be only sensible) the usual logarithmic mean temperature difference (LMTD) is expected to be several degrees (up to 10°C), especially for gaseous media.

Therefore, the best operation of this heat exchanger might be organized for hot regions with low relative humidity values, while for areas with a mild climate the efficiency will be much lower, and for climate zones with very small difference between dry bulb and wet bulb temperatures in the ranges of 2–4°C the effect of application of such a device will be negligible.

4. TECHNICAL PROBLEMS OF THE COOLING TOWER USING M-CYCLE

It is very difficult, if not impossible, to manufacture real fill design, corresponding to the cooling tower schema from Fig. 3a. Incoming airflow should move inside
the cooling tower fill along a very complicated route. The fill should have two types of channels:

(a) in the beginning of the air route air channels should be insulated from moisture in order to provide air cooling with constant absolute humidity;

(b) only after the air is cooled, it may participate in the contact heat-and-mass transfer with the water flow of the cooling tower. The expected very large aerodynamic resistance to the airflow will be a problem for the designer.

Even this imaginary design is promising, because there is no division of incoming airflow into product and working flows and one fan unit is enough; to develop the fill with the required channels’ design is a very challenging goal. It is necessary to organize additional research work and tests in order to confirm the real possibility to design such a fill.

Another variant, similar to one already tested in air conditioning systems, considers location of the heat transfer device, working on the M-cycle principle, in front of the air inlets (Fig. 3b).

The air cooling device will be large, because it is necessary to provide the cooling of significant airflows, while the efficiency of air-to-air heat exchangers is low. Because there are limitations for the temperature difference between product and working airflows, the heat transfer area of the device should be large, in order to compensate for this shortcoming.

As an example, let us examine the same cooling tower, for which thermal calculations were carried out. Let us assume that it is necessary to decrease the inlet air temperature by 5°C, while the LMTD between product and working airflows is 5°C and the heat transfer coefficient depends mainly on convective heat transfer at the dry air side and is in the range of 25–30 W/m²·K. The required heat transfer surface will be in the range of 5800–6900 m². Let us chose the film fill’s parameters for the heat exchanger’s volume evaluation. If the ratio of full contact surface to unit volume is in the range 100–150 m²/m³, then the volume of the additional heat exchanger at the air inlet of the cooling tower will be in the range of 38–69 m³, which is below 12% from the entire cooling tower volume. Even keeping in mind the larger size of this inlet air cooler (because for better airflow larger air channel diameters are desirable), such an increase of the cooling tower volume is acceptable.

Generally, for huge cooling towers installation of additional exhaust fan units for working airflow is required. However, for small cooling towers with forced-draft fan units the division of the incoming airflow for moving through product and working channels may be organized by the same fan, as is already done in the Coolerado air conditioner (Fig. 3c). But in any case, the productivity of these fan units should be doubled, compared with the similar conventional cooling tower, because product airflow and working airflow are of the same value, but working air is exhausted in the environment without any usage in the cooling tower. Therefore energy consumption for the air supply will practically be doubled for the designs in Figs. 3b and 3c.
It is necessary to exclude recirculation of the working airflow; i.e., the used air should be removed into the environment far enough from the cooling tower air inlets.

Additional inlet air cooling will be provided by evaporation of extra water mass, but its contribution to total water consumption will be not large. For the same cooling tower, evaluation gives the additional amount of the required water at about 1.2 m³/h. During the main water cooling process about 3 m³/h will be removed with the used air. Total water makeup below 2.2% may be considered as acceptable.

One more technical problem needs to be considered: if the water that is used for working airflow cooling is not clean enough and complete evaporation is adopted without any recirculation, then wetted working flow channels could be clogged.

5. CONCLUSIONS

Solution of the technical problems, formulated in this article, is a real challenge for the designers of evaporative cooling towers and the related heat exchangers. Several goals are of the first priority:

1. It is very important to confirm in laboratory tests the real possibility to cool the water below the wet bulb temperature of the surrounding air in the cooling tower models, using Maisotsenko cycle principles.
2. The attempt should be made to develop the design of the cooling tower with the complicated fill shape without division of the airflow into working and product flows (Fig. 3a).
3. It is necessary to design an air-to-air heat exchanger similar to the already existing Coolerado heat exchangers, implementing the Maisotsenko cycle (Figs. 3b and 3c), but for larger air volume flow rates, i.e., with larger equivalent diameters of the air channels.
4. It is necessary to carry out and to analyze laboratory tests of the said designs in order to make a conclusion about the practical possibility to implement the Maisotsenko cycle for real cooling towers’ modernization.

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