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Numerical analysis of performance of cooling towers

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SUMMARY

Cooling towers (CT), as a part of thermal power plants, have a major importance from the thermodynamic point of view. They represent a necessary part of a thermo-energetic facility needed to cool down the cooling water from the condenser, which cools the steam coming from the turbine thus closing the thermodynamic cycle. One of the basic parameters involved in evaluation of the performance of a cooling tower is the so-called number of transfer units (NTU). In this paper, the performance of cooling towers operating as a part of Units 5 and 6 in the Thermal Power Plant Kakanj (TPPK) in Bosnia and Herzegovina will be analyzed. For the purpose of the numerical evaluation of the NTU value, real data are taken from the TPPK. Based on the obtained results, suggestions will be given for improvement of the cooling towers performance.

Key words: *Cooling tower, number of transfer units, numerical integration, heat transfer.*

1. INTRODUCTION

In many industrial facilities, especially in thermal power plants, it is necessary to remove heat from a part of the facility into the surroundings, in order to keep the local temperature from rising or to close a thermodynamic cycle. In some applications heat can be removed by direct contact of the working fluid with the surrounding air, but in most cases cooling with water would be more economical. In cases where enough water supplies are available, once-through cooling system can be used, whereas if the water supply is insufficient or the temperature of water available is too high, cooling systems with circulating water are used. Main part of such systems are cooling towers, in which heat is transferred from the cooling water to the atmospheric air stream. Thus, the main task of a cooling tower is to enable heat to be transferred conveniently and economically from the plant to the atmosphere.

In this paper, the performance of cooling towers operating within Units 5 and 6 in the Thermal Power Plant Kakanj (TPPK) will be analyzed using real data taken from the TPPK, as well as the results of previous studies, which were performed for the purpose of improving the cooling system performance as a whole. Geometry and other parameters of the cooling towers were fixed in the last reconstruction of the cooling system, after which 14 cooling towers remained available with fills made of hard PVC plates with dimensions $600 \times 1000\ mm$, $0.8\ mm$ thick, which are fitted over each other in three layers. The height of the fill $1800\ mm$ and its cross-sectional area of $130\ m^2$ give the total fill volume of $234\ m^3$ per tower. Fans used for forced air circulation produce air mass flow rate of *370 m³/s* in each tower. It was guaranteed after the reconstruction that the section of 14 cooling towers could be used for cooling a maximal amount of *17500 m³/h* cooling water, i.e. *1250 m³/h* in each tower, with a cooling range of *9.4°C*. This means that for the atmospheric air temperature *15°C* and relative humidity *70 %*, the temperature of cooled water leaving the tower should be *19.2°C* with the cooling range of *9.4°C*.

2. COOLING TOWER MODEL

Generally accepted analysis of heat and mass transfer in cooling towers was developed by a German professor Frederick Merkel (1925) [1] and his model with the derivation of basic energy conservation equations can be found in references. Schematic of a direct-contact counter flow cooling tower, with fill volume V [m^3], extended water surface per unit fill volume a [m^2/m^3], water and air mass flow rates $\dot{m}_w$ and $\dot{m}_a$ is shown in Figure 1.

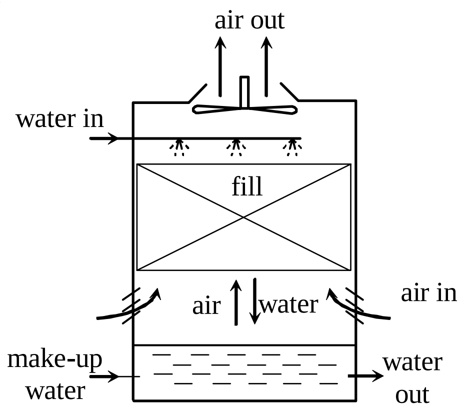


Fig. 1 Schematic of a direct-contact counter flow cooling tower

Heat and mass transfer from the water droplet into the main air stream is shown in Figure 2. Water droplet at temperature t_w is surrounded by air at temperature t_a, enthalpy h_a and humidity ratio x_a. The interface is assumed to be a film of saturated air having temperature t_s, enthalpy h_s and humidity ratio x_s. The total heat transfer rate from the droplet to the interfacial film can be expressed with the following equation:

$$dq_w = \dot{m}_w c_{pw} dt = k_w a\left(t_w - t_s\right) dV \qquad (1)$$

where q_w [W] is the total heat transfer rate, $\dot{m}_w$ [kg/s] is water mass flow rate, k_w [$W/m^2{\cdot}K$] is the total heat transfer coefficient from the droplet to the air film, V [m^3] is fill volume, or cooling volume and a [m^2/m^3] is extended water surface per unit fill volume which is the actual surface over which heat and mass transfer takes place.

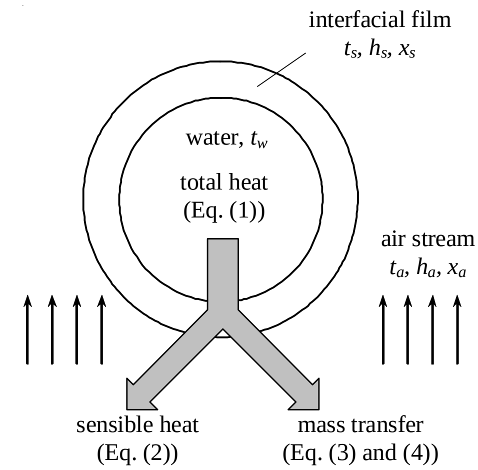


Fig. 2 Heat and mass transfer from water droplet into the air

This heat flux is then transferred from the interface to the main air stream in the fill by two means: as sensible heat transfer and as mass transfer, i.e. heat transfer with mass transfer.

Sensible heat transfer from the interfacial film into the air stream can be defined by the following equation:

$$dq_s = k_s a(t_s - t_a)\ dV \qquad (2)$$

where q_s [W] is sensible heat transfer rate and k_s [$W/m^2{\cdot}K$] is the total heat transfer coefficient film the interfacial film to the air stream.

On the other hand, mass transfer due to water evaporation can be written as follows:

$$d\dot{m}_w = K'\, a\left(x_s - x_a\right) dV \qquad (3)$$

where $\dot{m}_w$ [kg/s] is mass transfer rate due to evaporation, K' [$kg/m^2{\cdot}s$] is mass transfer coefficient from the interface to the main air stream, x_s [kg_w/kg_a] is humidity ratio of saturated air at the interface and x_a [kg_w/kg_a] is humidity ratio of the main air stream.

Heat flux due to evaporation of water is then defined by the following equation:

$$dq_e = h_{fg} d\dot{m}_w = h_{fg} K'\, a\left(x_s - x_a\right) dV \qquad (4)$$

where q_e [W] is heat transfer rate due to evaporation and h_{fg} [J/kg] is latent heat of evaporation.

By combining the two coefficients k_s and K' into one coefficient, a dimensionless coefficient known as the Lewis number is obtained and due to Merkel, it may be assumed to be equal to one, i.e.:

$$\frac{k_s}{K'\, c_{pa}} = 1 \qquad (5)$$

where c_{pa} [$J/kg{\cdot}K$] is a specific heat of moist air.

Noting that heat lost by the water must be equal to heat gained by the air, the total heat flux from the droplet into the main air stream can be written as:

$$dq_w = dq_s + dq_e, \qquad (6)$$

which can be rearranged, by combining Eqs. (1), (2) and (4) with Eq. (5) to the following form:

$$\dot{m}_w c_{pw} dt = \dot{m}_a dh = K' a(h_s - h_a)\, dV \qquad (7)$$

where $\dot{m}_a$ [kg/s] is mass flow rate of the main air stream.

This equation expresses heat transfer rate from the interfacial film to the main air stream, but conditions at the interface are indeterminate. If an overall coefficient K, based on enthalpy of saturated air h_w at water temperature is introduced, the above equation can be written as:

$$\dot{m}_w c_{pw} dt = \dot{m}_a dh = Ka(h_w - h_a)\, dV \qquad (8)$$

which leads to the main integral equations for evaluation of the tower characteristic:

$$\frac{KaV}{\dot{m}_w} = \int_{t_{w,in}}^{t_{w,out}} \frac{c_{pw}}{h_w - h_a} dt \qquad (9)$$

and:

$$\frac{KaV}{\dot{m}_{zr}} = \int_{h_{a,in}}^{h_{a,out}} \frac{dh}{h_w - h_a} \qquad (10)$$

where indices *in* and *out* stand for inlet and outlet of air from cooling tower. In engineering practise the integral on the right hand side of Eq. (10) is referred to as the number of transfer units (NTU), which actually shows how many times a mean enthalpy potential (h_w–h_a) is consisted in a water temperature difference ($\Delta t = t_{w,in} - t_{w,out}$). Thus, the definition of one transfer unit would be:

$$\frac{c_{pw}\Delta t}{(h_w - h_a)_{mean}} = 1 \qquad (11)$$

An explanation of Eq. (9) can be given with help of the graph in Figure 3, where enthalpy of air is plotted against water temperature.

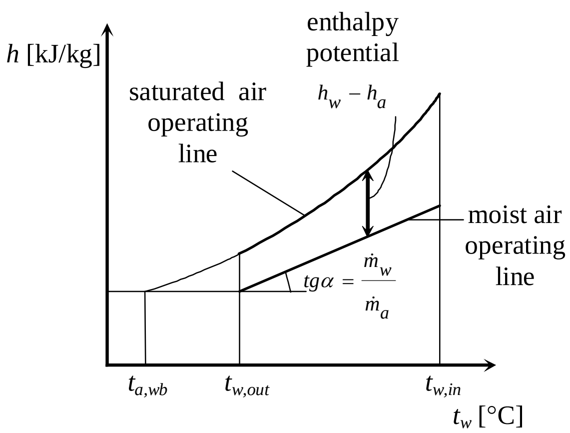


Fig. 3 Graph enthalpy - temperature (h-t diagram)

Eq. (7) implies that the air operating line is a straight line and saturated air operating line, also called water line, is obtained by calculation of enthalpy of saturated air at water droplet temperature. The conditions of water at the tower inlet and outlet are defined by temperatures $t_{w,in}$ and $t_{w,out}$ respectively, and the temperature $t_{a,wb}$ is the wet-bulb temperature of the inlet air, which should be approached by outlet water temperature in case of an infinite fill cross-sectional area. This can be clearly illustrated by Eq. (9) where an increase of the fill surface leads to the decrease of the enthalpy difference (h_w–h_a), whereas in a limiting case of an indefinite surface, the enthalpies and their corresponding temperatures $t_{w,out}$ and $t_{a,wb}$ become equal.

3. NUMERICAL CALCULATIONS

Cooling system of Units 5 and 6 in the TPPK is currently half-circulating, i.e. a part of the cooling water is rejected into a nearby river and the rest is cooled in the cooling towers. In the studies performed recently [2] it is suggested that Units 5 and 6 switch from the half-circulating to a full-circulating cooling system, trying to achieve less water supply from the river, which would make the cooling system more independent on water level and conditions in the river. According to these studies, the cooling towers would need to cool an amount of approx. *27,000 m^3/h* of cooling water (*1,930 m^3/h* in each tower). Hydraulic simulations of such cooling system were performed using a well known software EPANET 2.0 for hydraulic simulations. Results, (see Figure 4) show a non-uniform pressure and water mass flow rate distribution over the cooling towers, which could be successfully eliminated by reconstruction steps suggested in Ref. [2]. These simulations were firstly performed for the case of partial reconstruction, providing only the circulating cooling system, and later on for the case of full reconstruction, which would give nearly equal water mass flow rate in each tower.

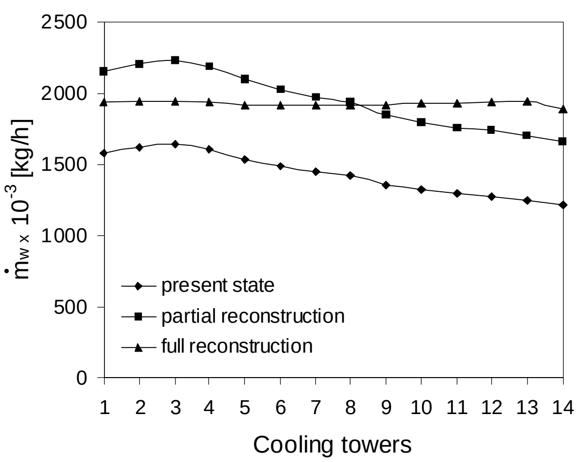


Fig. 4 Water mass flow rates for cooling towers

In order to analyze the performance of cooling towers, it is necessary to calculate the value of the integral in Eq. (9), since it represents the tower characteristic. The integral was evaluated numerically using the method of Chebyshev and the method of mean enthalpy potential, both implemented in a computer program written in FORTRAN, whereby nearly equal values were obtained.

In Chebyshev method, the following approximation formula was used:

$$\frac{KaV}{\dot{m}_w} = c_{pw}\left(t_{w,in} - t_{w,out}\right)\frac{1}{4}\left(\frac{1}{\Delta h_1} + \frac{1}{\Delta h_2} + \frac{1}{\Delta h_3} + \frac{1}{\Delta h_4}\right) \tag{12}$$

where Δh_i, $i=1,\ldots,4$ $[J/kg]$ are values of enthalpy differences, which are calculated for temperatures $t_i=t_{w,out}+c_i(t_{w,in} - t_{w,out})$ and c_i are the coefficients of Chebyshev with values $c_1=0.102673$, $c_2=0.406204$, $c_3=0.593796$ and $c_4=0.897327$.

For calculations using mean enthalpy potential, Ref. [3], the following formula was used:

$$\frac{KaV}{\dot{m}_w} = c_{pw}\Delta t\sum_{i=1}^{n}\frac{1}{\left(h_w - h_a\right)_{mean,i}} \tag{13}$$

where mean enthalpy differences $(h_w–h_a)_{mean,i}$ are calculated for equal temperature intervals Δt. The integral was calculated in 50 points over the height of the tower, i.e. for temperature increment $\Delta t=0.188°C$. For the given water parameters (inlet temperature *28.6°C*, outlet temperature *19.2°C*, mass flow rate *1250000 kg/h*) and air (inlet temperature *15°C*, specific humidity *70 %*, mass flow rate *1531.800 kg/h*) the calculated value for the cooling tower characteristic is: $\left(KaV/\dot{m}_w\right) = NTU = 1.758$. This value corresponds to the water outlet temperature *19.2°C* with cooling range *9.4°C*. The saturation pressures were calculated using the following formula taken from Ref. [4], with accuracy of *0.3 %* within the temperature interval from –35°C to +35°C:

$$P_s = 6.11exp[(17.67t_s)/(t_s+243.5)] \tag{14}$$

where P_s $[mbar]$ is saturation pressures, t_s $[°C]$ is saturation temperature.

4. RESULTS

The variation of enthalpy of air with water temperature in the tower fill is plotted in Figure 5. The enthalpy values are calculated for aforementioned parameters, based on air flow rate fixed by the fans and the water flow rate of *1250 m³/h*. Since the cooling towers work with different water flow rates shown in Figure 4, which are not equal to *1250 m³/h*, the outlet temperature of water will be different for each tower. In order to get more insight in how the tower characteristic would change with water outlet temperature and mass flow rate, values of water outlet temperature are varied between *17°C* and *23°C* with the increment of *0.1°C* and tower characteristics are calculated for each tower, taking into account the real water mass flow rate for each tower. The calculations were performed for the actual state, as well as for the situations after partial and full reconstruction suggested in Ref. [2], and the results of calculations are shown in Figures 6, 7 and 8.

The working regime of cooling towers corresponding to the water outlet temperature 19.2°C and mass flow rate *1250 m³/h*, with *NTU=1.758*, is indicated in Figures 6, 7 and 8 (black square). It can be seen in Figure 6 that only towers 12, 13 and 14 can satisfy the aforementioned requests, because their water mass flow rates are nearly equal to *1250 m³/h* (Figure 4). It is obvious that, due to non-uniform water distribution to the towers, their thermal performance is not equal. In other words, the towers will not cool the water to the same outlet temperature. In consideration of their thermal performance, cooling towers 1 to 11 are overloaded, whereas towers 13 and 14 are slightly under loaded. Such situation would get even worse if a circulating cooling system would be introduced, which can clearly be seen from Figures 7 and 8. After switching to the circulating cooling system, the towers would need to cool the amount of *27000 m³/h* water instead of *17500 m³/h* within the same cooling range. At much greater water mass flow rates none of the cooling towers could have the characteristic equal to *1.758*, cooling the water to the outlet temperature *19.2°C* at the same time. In order to achieve such water temperature at the outlet, the towers should, provided the other parameters are unchanged, have a characteristic greater than *1.758* (in Figures 7 and 8 this value is even greater than 2). This in fact means that, with the actual fixed value of the tower characteristic *1.758*, the towers would cool the water to the temperature greater than *20°C*, which would not satisfy the requests.

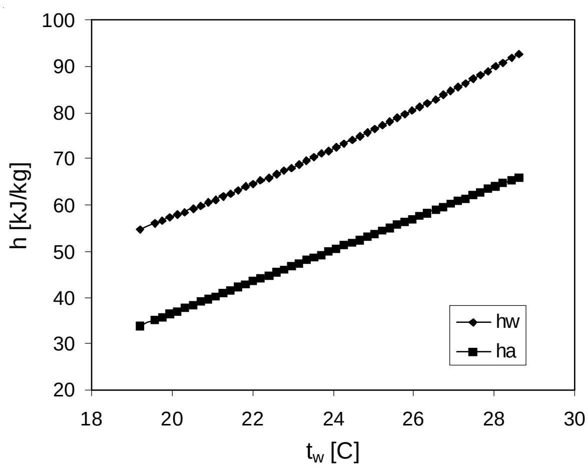


Fig. 5 h-t diagram of cooling towers

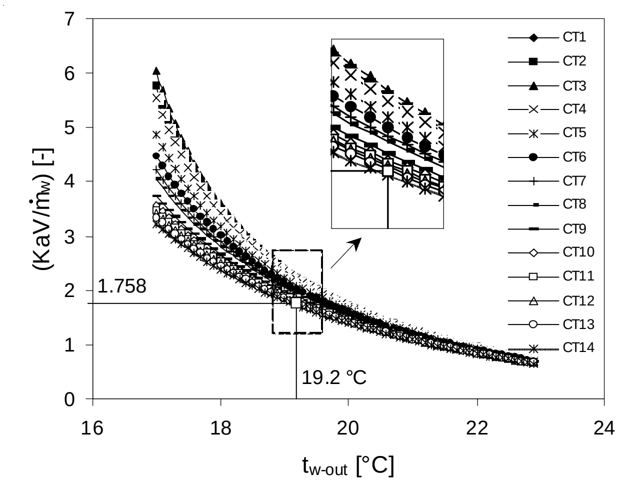


Fig. 6 Tower characteristic for present state

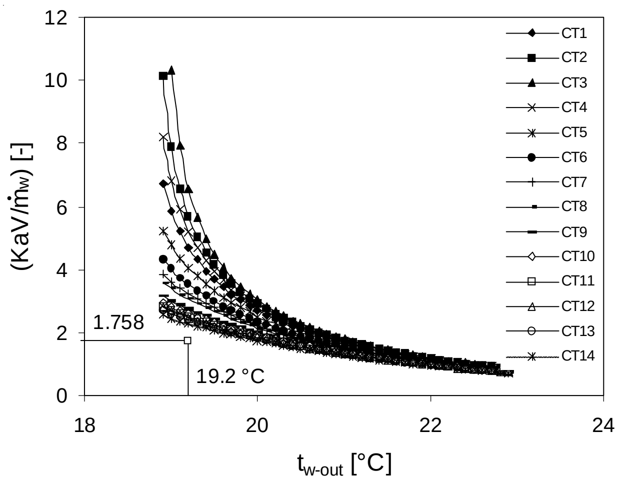


Fig. 7 Tower characteristic after partial reconstruction

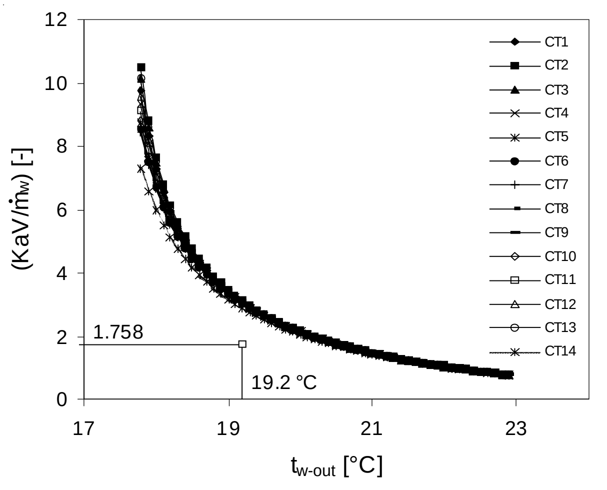


Fig. 8 Tower characteristic after full reconstruction

5. CONCLUSIONS

A general conclusion would be that, before switching to the circulating cooling system, it is necessary to perform additional reconstruction of the cooling towers in order to increase the value of the tower characteristic. According to Eq. (9) this can be achieved by increasing fill surface a·dV. The tower characteristics are also influenced by the coefficient of heat and mass transfer *K*, which, in general, depends on velocity of fluid, i.e. water flow rate. To determine this dependence, experimental measurements would be necessary, after which a minimum needed fill surface could be determined, so that other parameters, such as cooling range and condition of atmospheric air, are satisfied. Switching to the circulating cooling system itself, without aforementioned reconstructions of the cooling towers, will not give the desired results.

6. REFERENCES

[1] D.R. Baker and H.A. Shryock, A comprehensive approach to the analysis of cooling tower performance, *ASME Trans., J. Heat Transfer*, Vol. 83, No. 3, pp. 339-350, 1961.

[2] N. Neimarlija, B. Stjepović, S. Selimović and I. Plančić, Testing the real state of the cooling system in thermoenergetic Units 5 and 6 and defining the recommendations for the reconstruction of the Thermal Power Plant Kakanj, Faculty of Mechanical Engineering in Zenica, University of Sarajevo, Zenica, 2002. (in Bosnian)

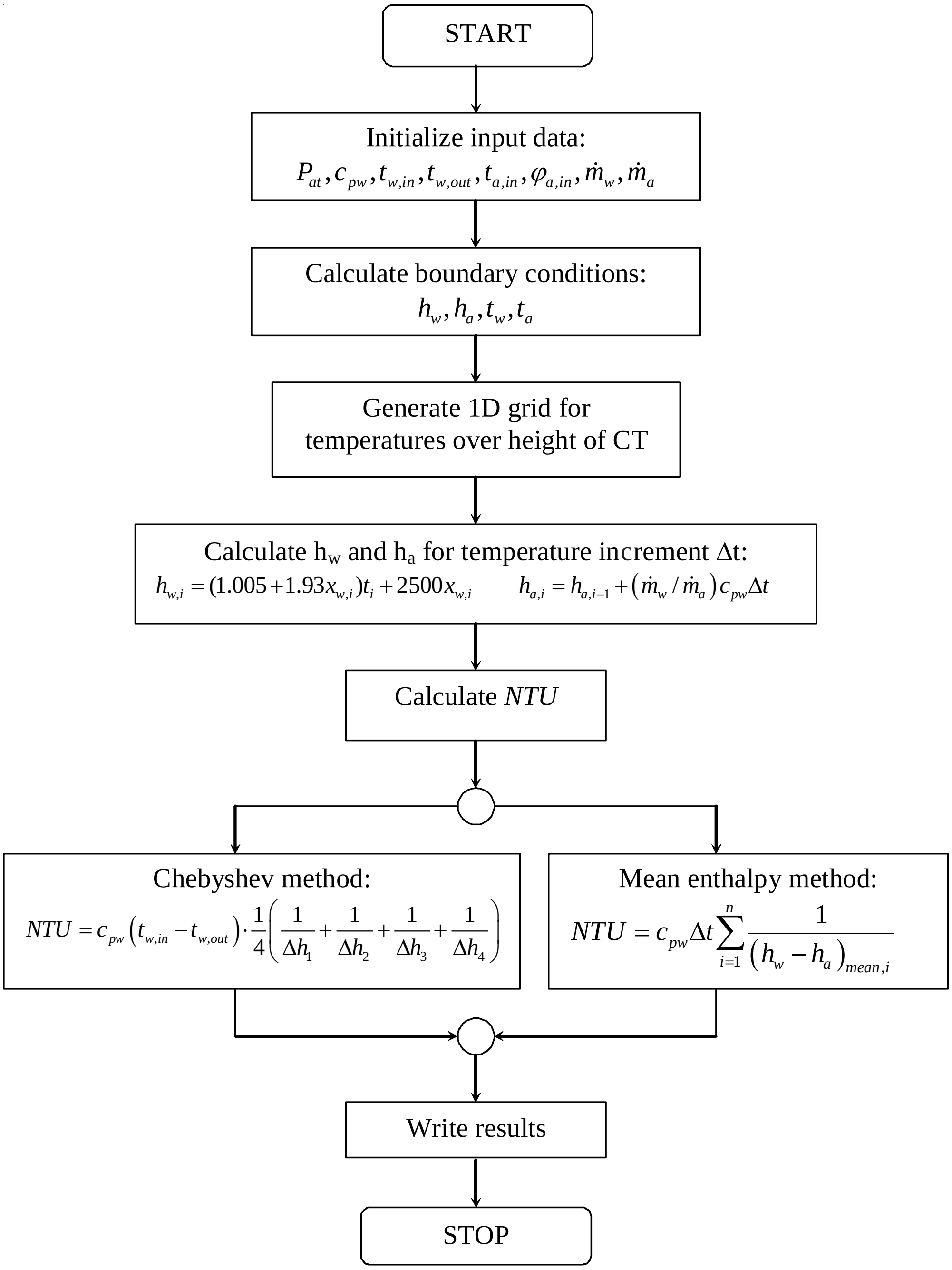
[3] J.D. Gurney and I.A. Cotter, *Cooling Towers*, McLaren & Sons Ltd., London, 1966.

[4] W.F. Stoecker and J.W. Jones, *Refrigeration and Air Conditioning*, McGraw Hill Book Company, 1982.

[5] D. Bolton, The computation of equivalent potential temperature, *AMS Monthly Weather Review*, Vol. 108, No. 7, pp. 1046-1053, 1980.

[6] P.J. Flatau, et al., Polynomial fits to saturation vapor pressure, *AMS Journal of Applied Meteorology*, Vol. 31, No. 12, pp. 1507-1513, 1992.

7. APPENDIX A – COMPUTER PROGRAM ALGORITHM



NUMERIČKA ANALIZA RADA RASHLADNIH TORNJEVA

SAŽETAK

Rashladni tornjevi (CT) kao dijelovi termoelektrana imaju veliki značaj s termodinamičke točke gledišta. Predstavljaju poseban dio termoenergetskih (potrepština) uređaja za hlađenje vode iz kondenzatora koji hladi paru koja izlazi iz turbine te na taj način zatvara termodinamički krug. Jedan od osnovnih parametara korištenih u procjeni rada rashladnog tornja je takozvani broj transfernih jedinica (NTU). Ovaj rad će analizirati rad rashlanih tornjeva koji su dio Jedinica 5 i 6 Termoelektrane Kakanj da bi se napravila numerička procjena NTU vrijednosti. Na osnovu dobivanih rezultata iznijet će se sugestije za poboljšanje rada rashlanih tornjeva.

Ključne riječi: *Rashladni toranj, broj transfernih jedinica, numerička integracija, prijenos topline.*