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Modelling the Thermo-Hydraulic Performance of Cooling Networks and Its Implications on Design, Operation and Retrofit

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1. Introduction

The set of heat transfer equipment through which heating and cooling takes place within a process is known as the heat exchanger network. Heat exchanger networks are made up of three main subsystems: a) the heating network, where thermal energy from external sources is supplied to a process, b) the heat recovery network, where thermal energy available within a process is recovered in order to minimize the external energy consumption, and c) the cooling network, where the low grade thermal energy from a process is rejected to the environment. A cooling network is an essential part of a cooling system; its design, operation and retrofit is looked at in this chapter.

The main components of a cooling system are the set of coolers or cooling network, the cooling system and the flow system which includes the pumping system and the pipeline. All the components of a cooling tower are linked and failure to appreciate this relation may lead to operating difficulties and excessive operating costs. In this chapter we look in detail at the interaction between these three components in the light of a thermo-hydraulic model.

The thermal analysis is based on the application of the thermal effectiveness model originally developed for single heat exchangers. On the hydraulic side, the water distribution through the network depends on the pressure drop across the various branches that make up the network; this in turn has implications on the thermal performance of the coolers.

During operation cooling systems are subject to changes in operating conditions that affect their heat removal capacity. These changes take place as a result of any of the following: plant throughput variations during the year; variations in ambient conditions that affect the performance of cooling towers; the performance of heat exchangers gets deteriorated over time due to fouling, etc. In practice, a cooling systems is commonly designed for point conditions; normally these point conditions correspond to the case where the whole system is well overdesigned. One problem with this practice is that the operating costs are much higher than they should be. This problem can be addressed by the proper application of a thermo-hydraulic model. In this context, this chapter shows the way a thermo-hydraulic model for cooling systems is developed and introduces its application in design.

Cooling systems are normally designed for fixed conditions; for instance, they are specified by a given cooling load, water flow rate, pumping capacity and heat transfer surface area in

coolers. Over design in all of these components is a common practice in industrial applications; such practice guarantees that the cooling utility is always available regardless of the expenditure of power. Under such circumstances, it is common to find that the operation of the cooling system remains unaltered even when plant throughput changes; however, one consequence of this mode of operation is the rise in operating costs due to pumping. The way to keep pumping power consumption at a minimum is by means of a flexible cooling system. A flexible cooling system is one that is able to adapt to the process heat removal demands as the plant throughput varies; such a systems is depicted in Figure 1; in this diagram, the cooling duty is represented by the length of the rectangles. Three different situations are shown: a) cooling duty for 100% throughput, b) cooling duty for for reduced throughput and c) cooling duty for for increased throughput.

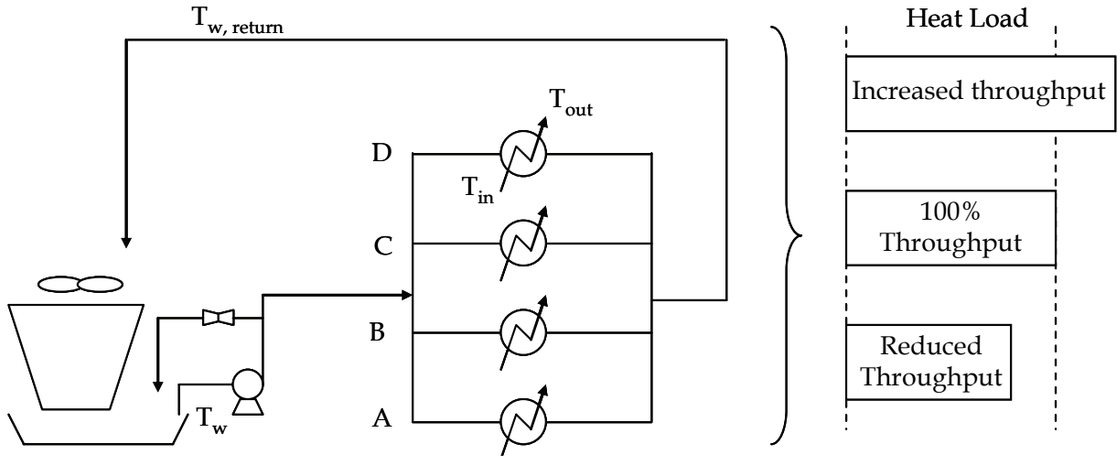


Fig. 1. Flexible cooling water system designed to supply the required heat load for changed plant throughput.

A cooling tower is one of the main thermal components of a cooling system; since in the operation of these types of units atmospheric air is used to cool down hot water, its thermal performance is strongly affected by the ambient conditions such as: wet bulb temperature and air relative humidity (Castro et al., 2000). The need then arises for a model that allows a quick and reliable determination of the thermal performance of cooling systems when they are subject to changed operating conditions.

The experimental and theoretical investigation of the thermal performance of cooling towers has drawn the attention of many researchers either experimentally or theoretically (Bernier, 1994; Jameel-Ur-Rehman et al., 2004; Gharagheizi et al., 2007; Chengqin, 2006; Nenad, 2001). Expressions to apply the thermal effectiveness method directly to cooling towers were developed by Jaber and Webb (1989). A more detailed and rigorous expression for the thermal effectiveness applied to cooling towers was presented by El–Dessouky et al. (1997). The calculation of the rigorous thermal effectiveness takes into consideration the ratio of the tangent to the enthalpy curve in two points: the midpoint between the tower inlet and outlet temperatures and the midpoint between the water outlet temperature and the wet bulb temperature. The thermal effectiveness model can be extended to determine the overall thermal effectiveness of a network of coolers. Cooling systems as integrated structures have also been studied either for design, improved operation and debottlenecking and water conservation (Giorgia et al., 2009; Majozi & Moodley,2008; Panjeshahi et al., 2009).

The work by Picon-Nuñez et al. (2007) shows such application where they demonstrate that the thermal response of a network of coolers subject to changes in operating temperatures is independent of the structure. This finding, along with the concept of flexible design of single heat exchangers becomes the basis for undertaking the design of flexible cooling systems.

2. Hydraulic model

It has been shown that the pressure loss due to valves and fittings is proportional to the fluid velocity raised to an exponent. In the turbulent region, this exponent varies between 1.8 and 2.1. However, in the majority of practical cases the pressure drop is calculated as a function of the square of the velocity.

In the case of water distribution systems, it can readily be demonstrated that the pressure drop through heat exchangers, valves and fittings can be related to volumetric or mass flow rate by means of equation (1):

$$\Delta P = KV^2 \tag{1}$$

Where ΔP is the pressure drop, K is a loss constant and V is the volumetric flow rate. The term V can be expressed as a function of the fluid velocity and the flow area as:

$$V = uA \tag{2}$$

Where u is defined as:

$$u = \left(\frac{V}{\pi d^2} \right) \tag{3}$$

The pressure drop through a straight pipe can be expressed as:

$$\Delta P = 2f \left(\frac{l}{d} \right) \rho u^2 \tag{4}$$

Where f is the friction factor, l is the pipe length, d is the pipe diameter and u is the fluid velocity. Substitution of equation (3) into equation (4) gives:

$$\Delta P = 2f\rho \left(\frac{l}{d} \right) \left(\frac{V}{\pi d^2} \right)^2 \tag{5}$$

From equation (5), the loss term (K) is defined as:

$$K = 2f\rho \left(\frac{l}{d} \right) \left(\frac{1}{\pi d^2} \right)^2 \tag{6}$$

The advantage of using the volumetric flow rate instead of velocity in the determination of the pressure drop is that it remains constant as the fluid flows through the various components. In a typical cooling system branch the minimum components are: valves, pipeline and heat exchangers; so, application of Equation (1) to these components leads to:

$$\Delta P_{valve} = K_{valve} V^2 \tag{7}$$

$$\Delta P_{pipe} = K_{pipe} V^2 \tag{8}$$

$$\Delta P_{Ex} = K_{Ex} V^2 \tag{9}$$

Equation (9) represents an alternative way of determining the pressure drop through heat exchangers as a function of the volumetric flow rate. The determination of the value of K for a heat exchanger requires an iterative approach where the geometry that meets the pressure drop for the given flow rate is sought.

A cooling network is made up of a series of branches each one containing the same elements mentioned above. Overall, the various branches that constitute a cooling network can be arranged in parallel, series or a combination of both.

One of the main factors that affect the thermal and hydraulic performance of a cooling network is the way water flow rate is distributed. For instance, branches located at a given altitude will carry less water than a similar branch located at a ground level. In the same way, as the total flow resistance increases in a branch the water flow rate reduces. The pressure drop through the various branches arranged in parallel that make up a cooling system is always the same (Figure 2). The water flow rate per branch can be calculated from:

$$V_A = \sqrt{\frac{\Delta P}{K_A}} \tag{10}$$

and

$$V_B = \sqrt{\frac{\Delta P}{K_B}} \tag{11}$$

Now, the total flow rate is obtained from:

$$V = V_A + V_B = \sqrt{\frac{\Delta P}{K_A}} + \sqrt{\frac{\Delta P}{K_B}} \tag{12}$$

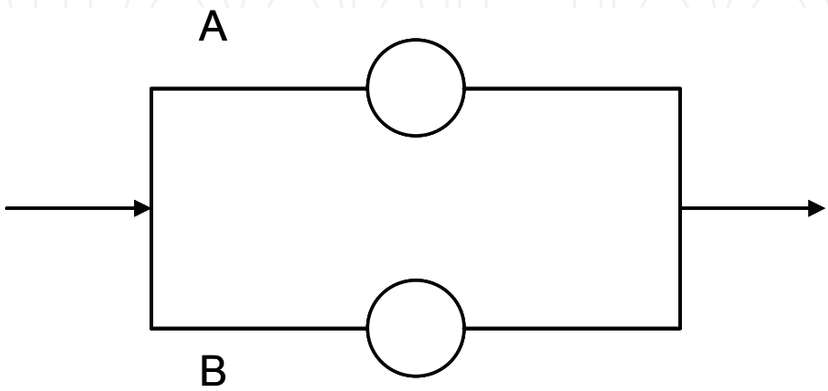


Fig. 2. Schematic of a two branch cooling network arranged in parallel.

The value of K for the whole system in Figure 2 is given by equation (13) where the terms K_A and K_B represent the resistances involved in each branch (pipeline, valves, fittings and heat exchanger):

$$K_{A+B} = \frac{\Delta P}{V^2} = \frac{1}{\left(\sqrt{\frac{1}{K_A}} + \sqrt{\frac{1}{K_B}}\right)^2} \tag{13}$$

For the case of piping, the K loss term is a function of the velocity whereas for valves and fittings its value is independent of it. The values of K for pipe fittings are given in Table 1.

3. Thermal model

The exchanger thermal effectiveness (ε), represents the ratio of the actual heat load to the maximum load that can thermodynamically be transferred. From this definition it can be shown that the exchanger thermal effectiveness can be represented by the ratio of the temperature change that the stream that exhibits the CP_{min} undergoes to the maximum temperature difference that exists between streams. The CP_{min} stream is the one whose product heat capacity-mass flow rate is the lower of the two. For the temperature profiles shown in Figure 3, it can be shown that the expression for the thermal effectiveness according to the definition above is given by:

$$\varepsilon = \frac{T_1 - T_2}{T_1 - t_1} \tag{14}$$

The exchanger thermal effectiveness can also be expressed as a function of Cr ($Cr=CP_{min}/CP_{max}$), the number of heat transfer units (NTU) and the exchanger flow arrangement. Expressions that relate these terms for a number of flow arrangements have been developed (Kays & London, 1984). For instance, for a one tube pass and two tube passes shell and tube heat exchanger, the expression that relates: ε, NTU and Cr is:

$$\varepsilon = 2\left\{1 + C_r + \left(1 + C_r^{\,2}\right)^{1/2} \left[\frac{1 + exp\left(-NTU\left(1 + C_r^{\,2}\right)^{1/2}\right)}{1 - exp\left(-NTU\left(1 + C_r^{\,2}\right)^{1/2}\right)}\right]\right\}^{-1} \tag{15}$$

The number of heat transfer units is defined as:

$$NTU = \frac{U\,A}{CP_{min}} \tag{16}$$

Where U is the overall heat transfer coefficient and A is the heat transfer surface area. From the heat balance equation it can be shown that:

$$C_r = \frac{CP_{min}}{CP_{max}} = \frac{t_2 - t_1}{T_1 - T_2} \tag{17}$$

Type of fitting or valve	Loss coefficient (K)	Type of fitting or valve	Loss coefficient (K)
45° elbow, standard	0.35	Plug disk, open	9.0
45° elbow, long radius	0.2	¾ open	13.0
90° elbow, standard	0.75	½ open	36.0
Long radius	0.45	¼ open	112.0
Square or miter	1.3	Angle valve, open	2.0
180° bend, close return	1.5	Y or blowoff valve, open	3.0
Tee, standard, along run, branch blanked off	0.4	Plug cocks, θ=5°	0.05
Used as elbow, entering run	1	10°	0.29
Used as elbow, entering branch	1	20°	1.56
Branching flow	1.0	40°	17.3
Coupling	0.04	60°	206.0
Union	0.04	Butterfly valve, θ = 5°	0.24
Gate valve, open	0.17	10°	0.52
¾ open	0.9	20°	1.54
½ open	4.5	40°	10.8
¼ open	24.0	60°	118.0
Diaphragm valve, open	2.3	Check valve, swing	2.0
¾ open	2.6	Disk	10.0
½ open	4.3	Ball	70.0
¼ open	21.0	Foot valve	15.0
Globe valve, bevel seat, open	6	Water meter, disk	7.0
½ open	9.5	Piston	15.0
Composition seat, open	6.0	Rotary (star-shaped disk)	10.0
½ open	8.5	Turbine wheel	6.0

Table 1. Loss coefficient for pipe fittings (Perry & Chilton, 1999).

From equation (14) it can readily be shown that the outlet temperature of the CP_{min} is given by:

$$T_2 = T_1 - \varepsilon \left(T_1 - t_1 \right) \tag{18}$$

And combining equation (14) and (17) we have for the outlet temperature of the CP_{max} stream:

$$t_2 = t_1 + C\, \varepsilon \left(T_1 - t_1 \right) \tag{19}$$

A closer look at equation (15) indicates that the thermal effectiveness is a function of C_r and NTU; when a temperature change enters a heat exchanger, this change does not have an effect upon the value of C_r or the value of NTU, so the thermal effectiveness remains unchanged. However, when mass flow rate changes, this has an effect upon the overall heat transfer coefficient which in turn affects the value of NTU , the value of C_r and therefore affects the thermal effectiveness. So, the thermal effectiveness changes and it has to be updated for the new exit temperatures to be determined. In the case of coolers, the temperature rise of water is normally kept low due to fouling problems; so large flow rates are normally handled making this stream in the majority of cases, the CP_{max} stream. So, equation (19) can be used to determine the cold stream outlet temperature.

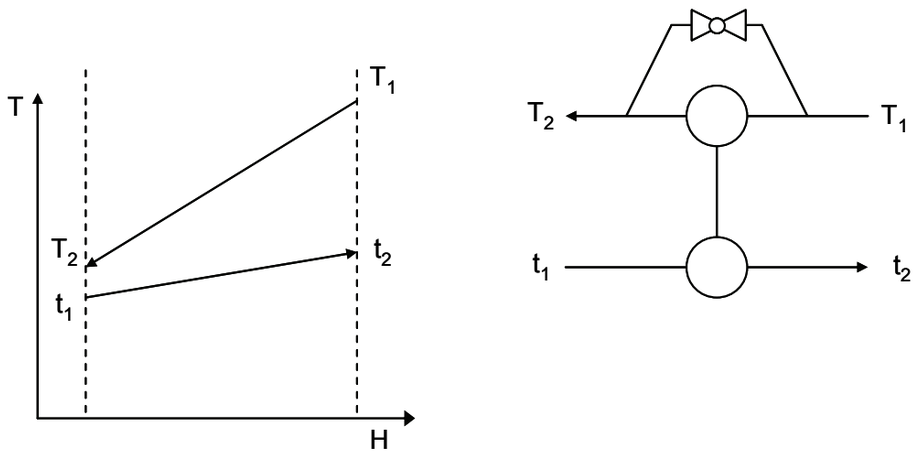


Fig. 3. Temperature profile for a heat transfer between two streams.

For the case of shell and tube heat exchangers, heat transfer coefficients can be updated as the water flow rate changes using the expressions below:

$$h_{tube}^{N} = F_{tube}^{0.8}\, h_{tube}^{o} \tag{20}$$

and

$$h_{shell}^{N} = F_{shell}^{0.6}\, h_{shell}^{o} \tag{21}$$

Where the terms F_{tube} and F_{shell}, represent the ratio of the original mass flow rate to the new mass flow rate; h is the heat transfer coefficient and the superscripts (o) and (N) indicate the original and new condition respectively.

Figure 4 shows the temperature and volumetric variables involved in a cooling network. For temperature and flow rate variations, the thermal performance of the network can be determined by solving the model which is made up of NV number of variables and the same

number of equations. The number of variables in such a network is given by (Picon-Nuñez et al., 2002):

$$NV = S + 2C + M \tag{22}$$

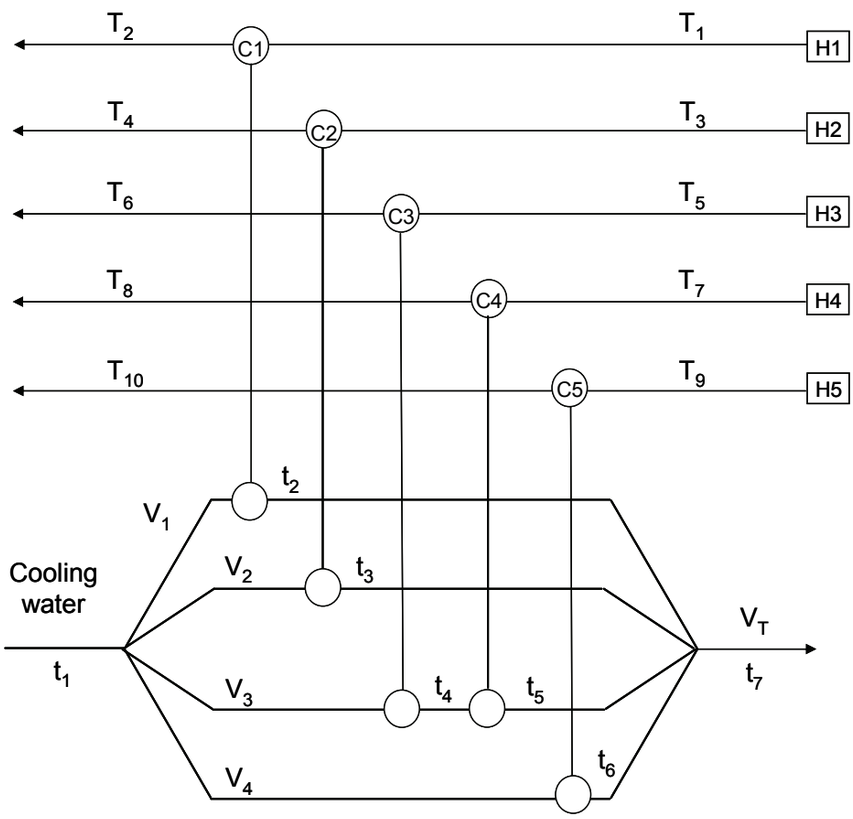


Fig. 4. Temperature variables in a water cooling network.

Where S is the number of streams; C is the number of coolers and M is the number of mixing points. So, for the case shown in Figure 4 we have that NV= 6+2x5 +1=17. The solution of the hydraulic model described in section 2 determines the values of the volumetric flow rates in the branches, so we only need to determine the set of equations in order to solve for the outlet temperatures of the system. These equations are: a) the stream inlet temperatures; b) process streams outlet temperatures (Equation 18); c) cooler outlet temperatures (Equation 19) and d) outlet temperature at mixing points. From an energy balance around the mixing point we have that:

$$t_m = \sum_{i=1}^{n} \left(\frac{V_i}{V_T} \right) t_i \tag{23}$$

Where tm is the temperature of water at the mixing point, Vi is the water volumetric flow rate of branch i and VT is the total water volumetric flow rate handled by the system. The known variables for a given network are: inlet temperatures, water flow rates, exchanger thermal effectiveness and Cr values.

4. Flexible operation of single heat exchangers

During operation, a heat exchanger is subject to variations in operating conditions that may result in a reduction or increase of the exchanger heat duty. A single heat exchanger that exhibits flexible operation has the following essential features: its installed surface area is carefully oversized according to the expected increase on the process throughput and is fitted with a bypass system (see Figure 3). The way these two features are used in operation in order to maintain the required heat load is as follows: a) Normal operation: the bypass operates partly opened; b) Increased throughput: in order to compensate for the required increase in heat load removal the bypass closes thus increasing the water flow rate through the cooler and its heat removal capacity. This mode of operation also applies in situations where the ambient wet bulb temperature increases; c) Reduced plant throughput: the bypass is further opened to reduce the water flow rate through the exchanger, thus reducing the heat transfer coefficient and consequently the heat load. This mode of operation also applies when the web bulb temperature goes down.

The size of the heat exchanger is determined for the expected set of conditions which require the largest surface area. This corresponds to the case where throughput and cooling water temperature are increased. The flow diagram of Figure 5 shows an approach for the design of a heat exchanger given a K value. Chemical plants often need to install a new cooler in an existing cooling network. This task should be undertaken by considering all the components of the network. It is necessary to define some operational data to design the new heat exchanger, for instance: allowable pressure drop, maximum heat load and a reference volumetric flow. Heat exchangers are related to K values from their geometrical features; therefore, it is possible to add a K resistance into an existing cooling water network for an expected hydraulic and thermal behaviour; the aim of this approach is to determine an approximate size of the new heat exchanger and to assess the performance of the whole network; this is, the way water flow rate is distributed and the effect upon the existing exchangers. Finally, after calculating these parameters, the heat transfer surface area and the number of tubes are determined.

5. Flexible design of cooling water systems

The principles described for the case of a flexible design of a single heat exchanger can be applied to cooling systems. Coolers are designed for the largest expected production rate, so they will be over designed for normal operation. When the process operates under normal production rate, the excess of water is returned back to the cold pond by means of a bypass. As the production rate increases, the bypass closes increasing the water flow rate through the system and thus the heat removal. For the case of lower production rate, the bypass is further opened and more water is recycled back to the cold water pond. When the water flow rate returned back to the cold pond equals the flow capacity of a single pump, one of the pumps can be taken out of service. On the other hand, as the cooling duty increases, the pump is set back into operation. An alternative means of controlling the water flow rate is the use of a pump velocity control system. This approach has the benefit of avoiding large power demands with the shut down and starts of the pumps. Further analysis is needed for best choosing the right control system.

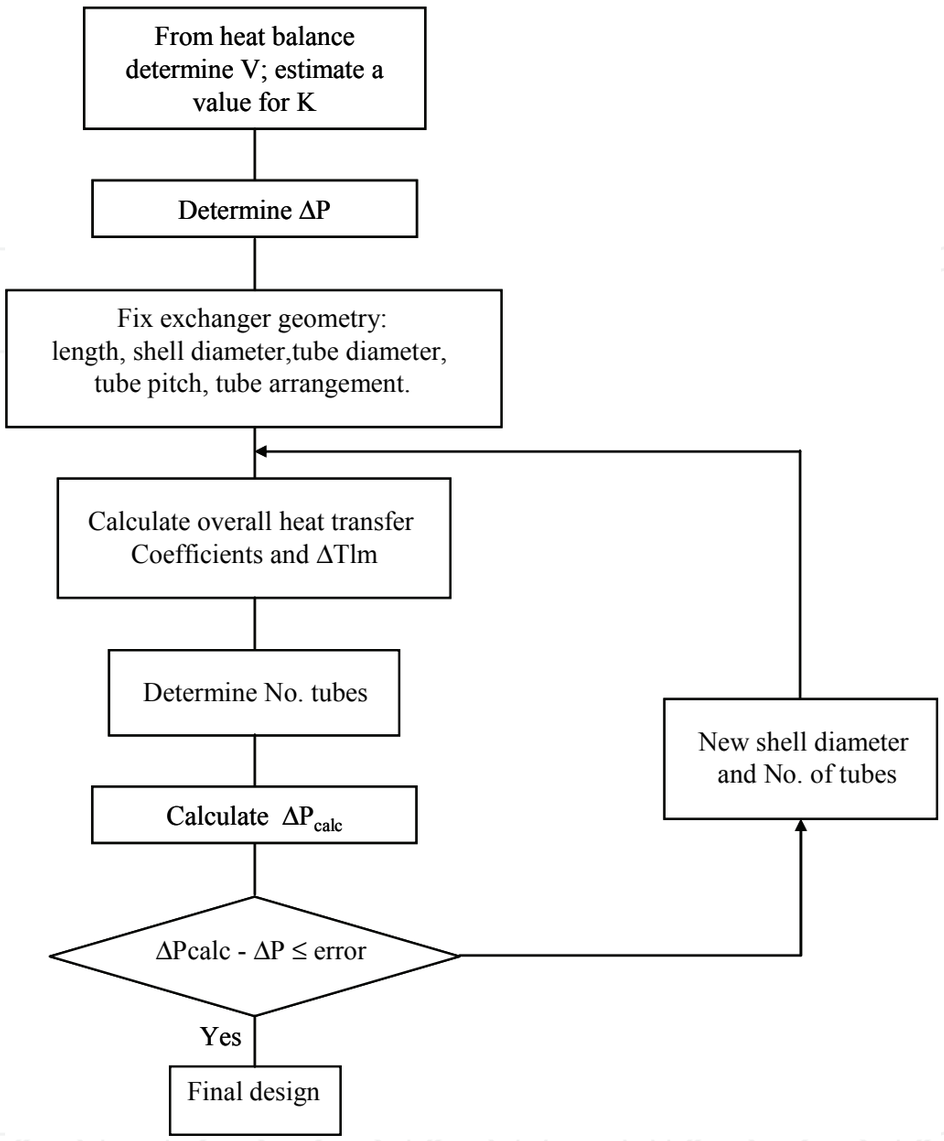


Fig. 5. Algorithm for the design of a heat exchanger in existing cooling network.

The design approach for flexible cooling systems involves thermal and hydraulic aspects, this is, it considers: pipe length, height, fittings, valves, cooling tower, pump capacity and heat exchanger performance. The approach starts by specifying the process and water temperatures, flow rates and physical properties; the values of K for each exchanger related to their specified pressure drop. For a flexible design the maximum expected production rate is considered. Then lengths and diameters for the various flow system components are determined and so their K value. For each branch, the summation of the K values is determined and then the value for the whole system is obtained by adding the K values of the various branches. Given the total water flow rate for the system, the total pressure drop is calculated. Figure 6 shows a flow chart for the thermo-hydraulic design of water cooling networks.

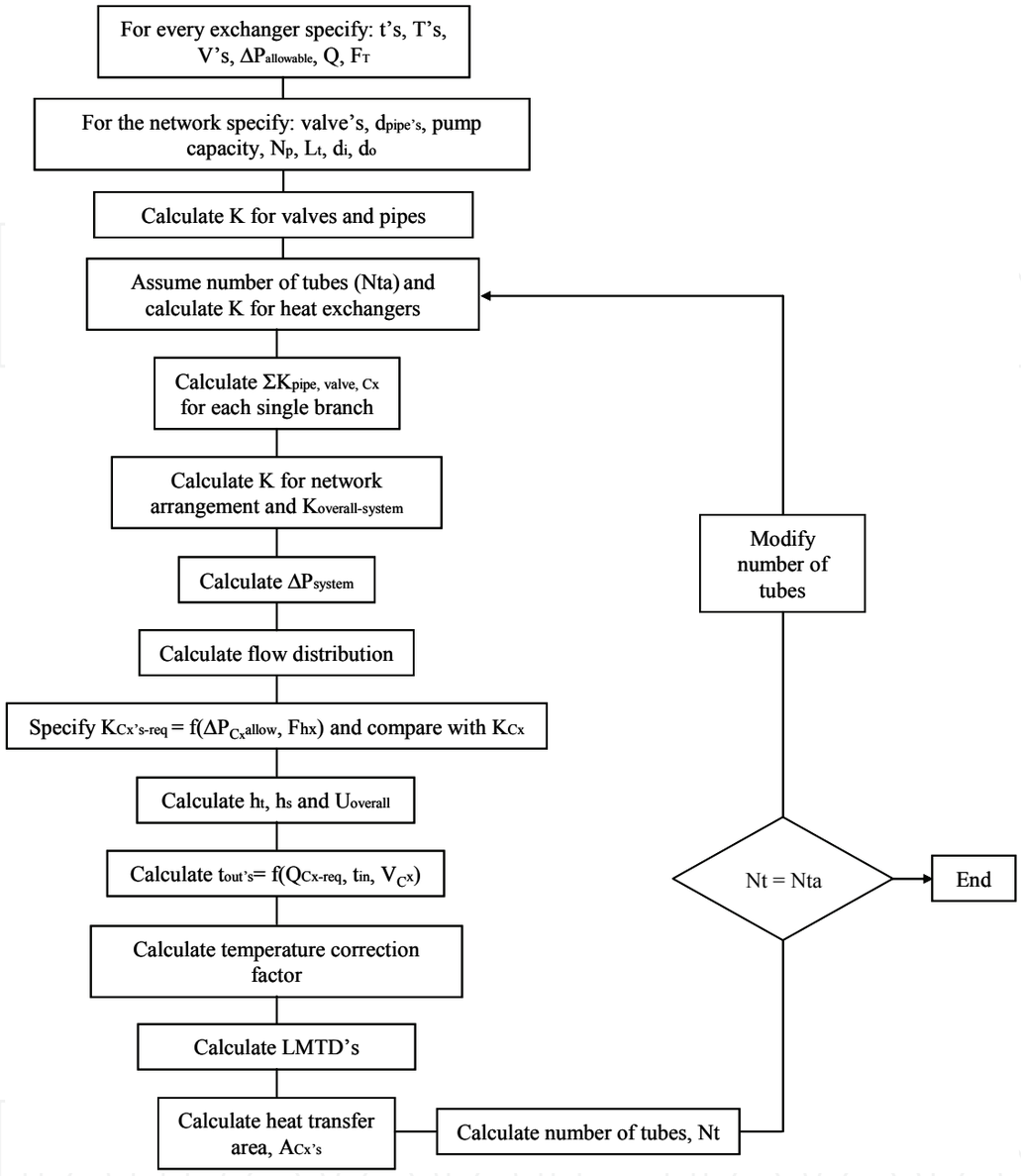


Fig. 6. Algorithm for the thermo-hydraulic design of a cooling network.

6. Case study

Using the principles introduced in the sections above, the design of a flexible cooling system will be looked at in more detail. A cooling system is designed to operate for a range of plant throughputs, namely: 120%, 100% and 80%. Heat exchangers are initially designed for the maximum production rate where the cooling demand is 3.65 MW; cooling water is available at 20°C and the maximum return temperature is 40°C with a water flow rate of: 0.0438 m3/s. The process information for the streams is given in Table 2. The allowable pressure drops, water flow rates and the corresponding K values are presented in Table 3. In Table 4 the geometry of the exchangers such as the tube passes, tube length and tube inner and outer diameter and the K values for the case of water control valves are also given. The

information related to the pipe lengths, diameters, heights and control valves is provided in Table 5. When the various components of the system are brought together, the cooling water is redistributed which in turn affects the thermal performance of the exchangers. So, the new water distribution is determined and the results are given in Table 6. The calculated pressure drops through branches are shown in Table 6.

Stream	Inlet temperature (°C)	Outlet temperature (°C)	CP (W/°C)	Heat duty (kW)
H1	50	30	8	400
H2	50	40	42.53	1,014
H3	85	40	16.8	1,906
H4	85	65	4.24	205
C1	20	40	5.14	400
C2	20	40	12.85	1,014
C3	20	40	23.13	1,906
C4	20	40	2.57	205

Table 2. Stream process data.

Exchanger	ΔPallowable (kPa)	Water flow rate (m3/s)	K
Cx1	40	0.0052	285,239
Cx2	40	0.0129	330,432
Cx3	40	0.0232	117,771
Cx4	40	0.0026	6,162,219

Table 3. Pressure drop, water flow rate and K values for heat exchangers.

Exchanger	Cx1	Cx2	Cx3	Cx4
K valve	9	9	9	9
No. Passes	2	2	2	2
length, m	6.1	6.1	6.1	6.1
tube i.d., m	0.0148	0.0148	0.0148	0.0148
tube o.d., m	0.01905	0.01905	0.01905	0.01905

Table 4. Geometrical parameters of heat exchangers.

Branch	Diameter (m)	Length (m)	Heigh (m)	Kvalve
Feed	0.35	220	0	9
A	0.25	120	5	9
B	0.25	170	7	9
C	0.2	180	10	9
D	0.2	190	10	9
Return	0.35	220	0	55

Table 5. Geometry and dimensions of the various components of the piping system.

Branch	Water flow rate (m3/s)	Pressure drop (kPa)
A	0.0020	40.03
B	0.0186	42.19
C	0.0112	41.95
D	0.0121	42.30
Total flow rate	0.0438	

Table 6. Water flow rate re-distribution through branches.

Under these conditions, the design of the heat exchangers has to be revisited and new dimensions determined for each exchanger. What is sought is that each exchanger removes the required heat load; on achieving this target, the water outlet temperature in some cases goes above the established limit of 40°C. The relation between surface area and K values can be seen in Tables 3 and 7; exchanger Cx3 has the largest surface area and the lowest K value, the opposite applies to exchanger Cx4 that exhibits the smallest surface area and largest K value.

	Cx1	Cx2	Cx3	Cx4
t_{out} (°C)	27.92	41.70	44.43	44.91
ΔP (kPa)	40.00	40.00	40.00	40.00
Surface Area (m2)	130	110	196	14
No. of tubes	356	301	537	37

Table 7. Revisited geometry of heat exchanger for the final water flow rate distribution.

The thermo-hydraulic model described in this work is used to demonstrate the flexible operation of the cooling system; for the various scenarios the process stream flow rates are multiplied by the required factors to simulate the increase or reduction in production rate. The network presented in Figure 1 is used as case study. Tables 8, 9 and 10 show the network water flow rates and temperatures for 120%, 100% and 80% throughput respectively.

Branch	Water temperatures (°C)		Process temperatures (°C)		Total tower cooling load (MW)	Cooling water (m^3/s)	System bypass (m^3/s)
	Inlet	Outlet	Inlet	Outlet			
A	20	28.29	50	29.05	3.65	0.0121	0.0
B	20	41.72	50	39.9		0.0111	
C	20	45.6	85	37.7		0.0186	
D	20	45.6	85	64.5		0.0020	
Water return temperature		39.8			Water flow rate (m^3/s)	0.0438	

Table 8. Water flow rate distribution and stream temperatures for a 120% plant throughput.

Branch	Water temperatures (°C)		Process temperatures (°C)		Total tower cooling load (MW)	Cooling water (m^3/s)	System bypass (m^3/s)
	Inlet	Outlet	Inlet	Outlet			
A	20	30.4	50	28.9	2.85	0.0081	0.0146
B	20	43.9	50	41.17		0.0074	
C	20	50.15	85	40.6		0.0124	
D	20	52.7	85	64.06		0.0013	
Water return temperature		43.2			Water flow rate (m^3/s)	0.0292	

Table 9. Water flow rate distribution and stream temperatures for a 100% plant throughput.

Since the scenarios that are analyzed consider seasonal variations in plant production, it is acceptable to consider that instead of just recycling water back to the cold pond, pumps can be taken out of operation for 100% and 80% plant throughput. For instance, six pumps can be selected for the condition of 120%. When production returns to 100%, one pump is taken out of operation. Further decrease in throughput to 80% is accompanied by the shutting down of a second pump.

Branch	Water temperatures (°C)		Process temperatures (°C)		Total tower cooling load (MW)	Cooling water (m^3/s)	System bypass (m^3/s)
	Inlet	Outlet	Inlet	Outlet			
A	20	32.9	50	29.51	2.05	0.005	0.0256
B	20	45.6	50	42.59		0.0046	
C	20	55.07	85	44.39		0.0078	
D	20	61.16	85	64.34		0.0008	
Water return temperature		46.82			Water flow rate (m^3/s)	0.0182	

Table 10. Water flow rate distribution and stream temperatures for a 80% plant throughput.

7. Determining the performance under changed cooling water temperature

Flexible operation can be extended to situations where the ambient conditions affect the performance of a cooling tower; for instance: variation in the wet bulb temperature. The model is used to simulate the case study given in Table 2. The simulation proceeds by assuming a change in the inlet temperature of the cooling water; the effect upon the outlet temperatures is determined and then the water flow rate is changed in order to restore the process stream target temperatures.

Table 11 shows the case where the inlet temperature is 24°C and 18°C respectively. When the temperature of the cooling water increases to 24°C, the simulation shows that increasing the water flow rate by 8%, the process outlet temperatures can be restored to acceptable values as shown in Table 12.

The water return temperature for this case is 35.49°C and the cooling load 3.47 MW. Now, for the case where the cooling water enters at 18°C to the system, the heat removal rate will be higher than required. Therefore, the solution consists in reducing the flow rate. Table 13 shows the thermal performance of the exchangers; the cooling duty for this case is 3.53 MW and the water return temperature is 41.4°C. Since the performance is within acceptable values, the system needs not to adjust its water flow rate.

Cooling water: 24°C			Cooling water : 18°C		
Unit	Tout (°C)	tout (°C)	Unit	Tout (°C)	tout (°C)
Cx1	31.85	31.18	Cx1	27.66	26.84
Cx2	41.32	42.82	Cx2	39.32	41.16
Cx3	41	48.06	Cx3	36.32	44.42
Cx4	65.65	48.22	Cx4	63.75	44.6

Table 11. Temperature response at different cooling water inlet temperatures.

Cooling water: 24°C			
Unit	Process stream Tout (°C)	Process stream Target (°C)	Cooling water tout (°C)
Cx1	31.4	30	30.4
Cx2	40.6	40	41.87
Cx3	39.1	40	46
Cx4	65	65	46

Table 12. Process outlet temperatures after increasing water flow rate by 8%.

Cooling water: 18°C			
Unit	Process stream Tout (°C)	Process stream Target (°C)	Cooling water tout (°C)
Cx1	28.48	30	28.4
Cx2	40.6	40	42.85
Cx3	39.33	40	48
Cx4	64.93	65	48.6

Table 13. Thermal performance after reduction of cooling water temperature.

8. Conclusions

Cooling systems are, in most applications, designed for a maximum fixed heat duty and its operation remains unchanged despite the variations in plant throughput. A disadvantage of this practice is that operating costs tend to be higher than they should be. Flexible operation is the capacity of the system to deliver the required heat load within some specific bounds. For instance, specific bounds may be the following: plant throughput may vary between a minimum and a maximum from normal production rate and the wet bulb temperature, which significantly affects the performance of a cooling tower, experiences changes between seasons. When a temperature or flow rate

disturbance enters a cooling system, it propagates around until the steady state is reached. Since the steady state response of a cooling system is independent of the actual arrangement of the coolers then the heat load removal can be controlled through the use of a bypass scheme. With the operation of a flexible cooling system, the pumping power consumption is a variable that can also be controlled. Typically, the design of a cooling system is carried out by considering individual components at a time for a given set of operating conditions; when these components are placed together into a system, the water flow rate is no longer the same since it tends to re-distribute through the network depending on the resistance it encounters. The result of this phenomenon is that water flow rate distributes through the network so that the pressure drop in all branches of the system equates. So, a thermo-hydraulic model has to be implemented for the determination of the performance of a cooling system and it can be used to produce a design that meets the required flexibility. Such a model has been introduced in this work.

9. Nomenclature

A heat exchanger surface area [m^2]
Cr heat capacity mass flow rate ratio
C number of coolers in cooling network
CP_{min} minimum heat capacity mass flow rate
CP_{max} maximum heat capacity mass flow rate
d diameter [m]
h heat transfer coefficient [$kJ/m^2°C$]
F_T logarithmic temperature correction factor
K loss coefficient
Lt tube lengths [m]
M number of mixing point in a cooling network
Nt number of tubes in heat exchanger
Nta number of tubes assumed
NP number passes
NTU number of heat transfer units
NV number of temperature variables in a cooling network
Q heat load [kW]
S number of streams
T hot stream temperature [°C].
t cold stream temperature [°C]
U overall heat transfer coefficient [$W/m^2°C$]
u fluid velocity [m/s]
V volumetric flow rate [m^3/s]

Subscripts
s shell side
t tube side
i inner condition
o outer condition

Superscripts
N new condition

o original condition
m mixing point

Greek letters

ε thermal effectiveness
ΔP pressure drop (kPa)
ρ density [kg/m3]

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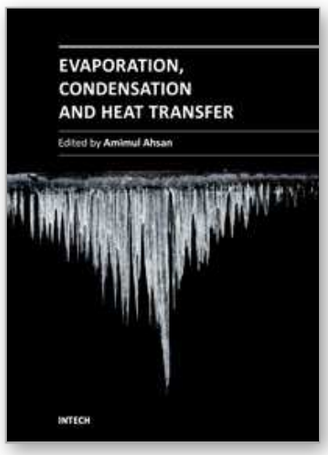
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