**Governing equations:**

Individual components of the cycle are modelled thermodynamically where mass and energy conservation equations are applied as follows:

The mass conservation equation,

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0 \tag{1}$$

The energy conservation equation,

$$Q + W = \sum \dot{m}_f h_f - \sum \dot{m}_{in} h_{in} \tag{2}$$

where $\dot{m}$ is the mass flow rate in kg/s, $Q$ is the heat input in kW, $W$ is the work output in kW, $h$ is the specific enthalpy in kJ/kg, and subscripts $in, out$ and $f$ represent the inlet and outlet and final conditions, respectively.

The heat exchanger calculations are based on the effectiveness model. In this model, Thermoflex uses a unique definition of effectiveness, particularly in calculating the maximum possible heat transfer. The effectiveness is defined as per Equation 3.

$$\epsilon = \frac{Q}{Q_{max}} \tag{3}$$

The actual amount of heat transfer is calculated from the given input of effectiveness and calculated $Q_{max}$. The maximum heat transfer $Q_{max}$ is calculated based on Equation 4.

$$Q_{max} = C_{min} * (T_{h,in} - T_{c,in}) \tag{4}$$

Where $T$ is temperature, the subscripts $c$ and $h$ represent cold and hot stream respectively and the minimum stream capacitance $C_{min}$ is calculated as per Equation 5,

$$C_{min} = \min (C_{cold}, C_{hot}) \tag{5}$$

The stream capacitance is calculated based on,

$$C_{cold} = m_{cold} * \frac{h_{c,in} - h_{c,out,max}}{T_{c,in} - T_{c,out,max}} \tag{6}$$

where, $T_{c,out,max} = T_{h,in}$ and $h_{c,out,max} = f(P_{c,out}, T_{c,out,max})$

Thermoflex discretise the heat exchanger to 13 zones or (defined by the user) and perform this calculation in every zone. The $Q_{max}$is reduced if the minimum pinch constraint is violated. The conductance is calculated based on Log Mean Temperature Difference (LMTD) method for every zone and summed to the total heat exchanger.

In case of the economiser, the cold outlet temperature is set as,

$$T_{c,out,max} = T_{sat} - T_{sub} \quad (7)$$

The steam generator, surface condenser and the feed water heaters were initially set up to match the design conditions. The size of all heat exchangers in the power block was then determined using the Log Mean Temperature Difference (LMTD) method, where LMTD is defined as follows:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (8)$$

where,

$$\Delta T_1 = \left(T_{Hot,in} - T_{Cold,out}\right) \quad (9)$$

$$\Delta T_2 = \left(T_{Hot,out} - T_{Cold,in}\right) \quad (10)$$

where $T_{Hot,in}$ and $T_{Cold,in}$ represent the hot and cold fluids inlet temperatures whereas $T_{Hot,out}$ and $T_{Cold,out}$ is the hot and cold fluids outlet temperatures. The area can be obtained from the Equation 11.

$$Q = U\ A\ LMTD \quad (11)$$

where $U$ is the overall heat transfer coefficient, $A$ is the area, Q is the heat duty and $LMTD$ is the Log Mean Temperature Difference. The turbine and pump work is determined by calculating the outlet stream enthalpies. For turbine,

$$h_{out} = h_{in} - \left(h_{in} - h_{out,s}\right) * \eta_{turb} \quad (12)$$

where, $\eta_{turb}$ is the turbine efficiency and $h_{out,s} = f(P_{out}, S_{in})$

For pump,

$$h_{out} = h_{in} + (h_{out,s} - h_{in})/\eta_{pump} \quad (13)$$

where, $\eta_{pump}$ is the pump efficiency and $h_{out,s} = f(P_{out}, S_{in})$

The performance of all heat exchangers in off-design conditions were modelled using the "Thermal Resistance Scaling" method. Thermal Resistance, R is the reciprocal of thermal conductance and is defined as:

$$R = \frac{1}{UA} = \frac{LMTD}{Q} \tag{14}$$

The off-design performance is then calculated through the off-design thermal resistance ($R_{OD}$) from the ratio between the design ($\dot{m}_{DES}$) and off-design ($\dot{m}_{OD}$) feed water mass flow rates. The exponent (x) can be found since the mass flow rates are known.

$$\frac{R_{OD}}{R_{DES}} = \left(\frac{\dot{m}_{DES}}{\dot{m}_{OD}}\right)^{X} \tag{15}$$

The combination of set-points defines the thermodynamic design of the plant from which the size of the components is determined, from which the off-design performance is derived.

## Water consumption calculations:

In the wet cooling tower substantial amount of water are lost through evaporation, blowdown, drift and leakages and therefore makeup water must be continuously supplied to cope with the water losses. Moreover, water consumption in wet cooling towers depends on the ambient temperature and humidity. Therefore, the chosen location has a considerable influence on the water consumption of the plant. In a CSP plant, there are two main factors that affect the water consumption in a wet cooling system, the DNI and the size of the solar field, and the dry and wet bulb temperatures of the selected area. The DNI and the solar field size of the plant determines the amount of electricity produced, which is directly related with the amount of circulating water needed for cooling the turbine exhaust steam. As the total annual electricity production increases, so does the total annual water consumption. Dry and wet bulb temperatures also dictate the consumption of water in a wet cooling tower as it is designed based on these parameters.

The total water consumed or required by a wet cooling tower to dissipate the heat of condensation can be estimated by the following equation.

$$\dot{m}_{\mathrm{make-up}} = \dot{m}_{\mathrm{evaporation}} + \dot{m}_{\mathrm{blowdown}} + \dot{m}_{\mathrm{drift}} \tag{16}$$

where $\dot{m}_{\mathrm{evaporation}}$ represents the water loss through the evaporation process, which is responsible for the cooling of hot water coming from the condenser. When cooling water passes through an evaporative/wet cooling tower, part of it is evaporated and sent to the atmosphere. This water is considered as a loss and must be replaced. Evaporative losses account for the greatest percentage (approximately 80%) of water consumption in a CSP plant using a wet-cooled system. Merkel method and Poppe method are widely used to calculate the performance of a wet cooling tower. The Merkel method assumes that the air leaving the tower is saturated whereas the Poppe method considers the Lewis factor at each step, which means that the outlet air is not necessarily saturated. Thermoflex uses a constant value of outlet air relative humidity in its calculations for cooling tower performance. This study used a value of 100% relative humidity, equivalent to the Merkel method.

In equation 1, $\dot{m}_{blowdown}$ represents the amount of water needed for blowdown. As the cooling water in the system evaporates, the concentration of total dissolved solids (TDS) in the remaining cooling water in a closed system will increase. TDS refer to minerals, scales and other dissolved materials that can lead to scaling and be corrosive to the operational environment. Accumulation of TDS in the system is controlled by purging the system. The amount of water drained off the system is replaced with fresh water. The amount of blowdown (purge) water depends on the amount of concentration that the plant and the environment to which it may be discharged can tolerate. It is calculated as:

$$\dot{m}_{blowdown} = \frac{\dot{m}_{evaporation}}{C-1} \tag{17}$$

where C is the concentration factor.

The last element, $\dot{m}_{drift}$, in Equation 1 represents drift losses that contribute for a small fraction in the total water loss, however, it cannot be neglected. Drift losses refer to small water droplets that are taken away outside the cooling tower unit by the discharge airstream. Water lost through drift in evaporative condensers and cooling towers is usually dependant on drift eliminators and it is calculated as a percentage value of the total mass flow of the recirculating cooling water in the system.

$$\dot{m}_{drift} = \dot{m}_{cw} * f_{drift} \tag{18}$$

where $f_{drift}$ represents the fraction of drift losses.