
4 Case studies: Mine surface cooling system

4.1 Preamble

Case studies were carried out on two deep level gold mines in the Klerksdorp region of South Africa. To maintain confidentiality, the names of the respective mines are not disclosed in this document and will subsequently be referred to as “Mine A” and “Mine B” instead.

The purpose of the case studies is to determine the effect of applying energy saving strategies, identified in Chapter 2, to the surface cooling systems of the two mines. Firstly a simulation model was developed for the baseline, which was subsequently verified. Thereafter, various energy saving strategies were applied to the model in order to predict the potential savings. Post-implementation, the strategies were validated and compared to the simulated results. This is presented in Chapter 5.

4.2 Case study 1: Mine A

4.2.1 System description

4.2.1.1 Layout

The mine cooling system consists of six refrigeration machines, six condenser cooling towers and four pre-cooling towers. The system also includes a hot and a cold dam with a confluence dam situated in between, where hot and cold water can mix together; a surge dam and three BACs. *Figure 9* shows the basic layout of all the components of the surface cooling system before project implementation.

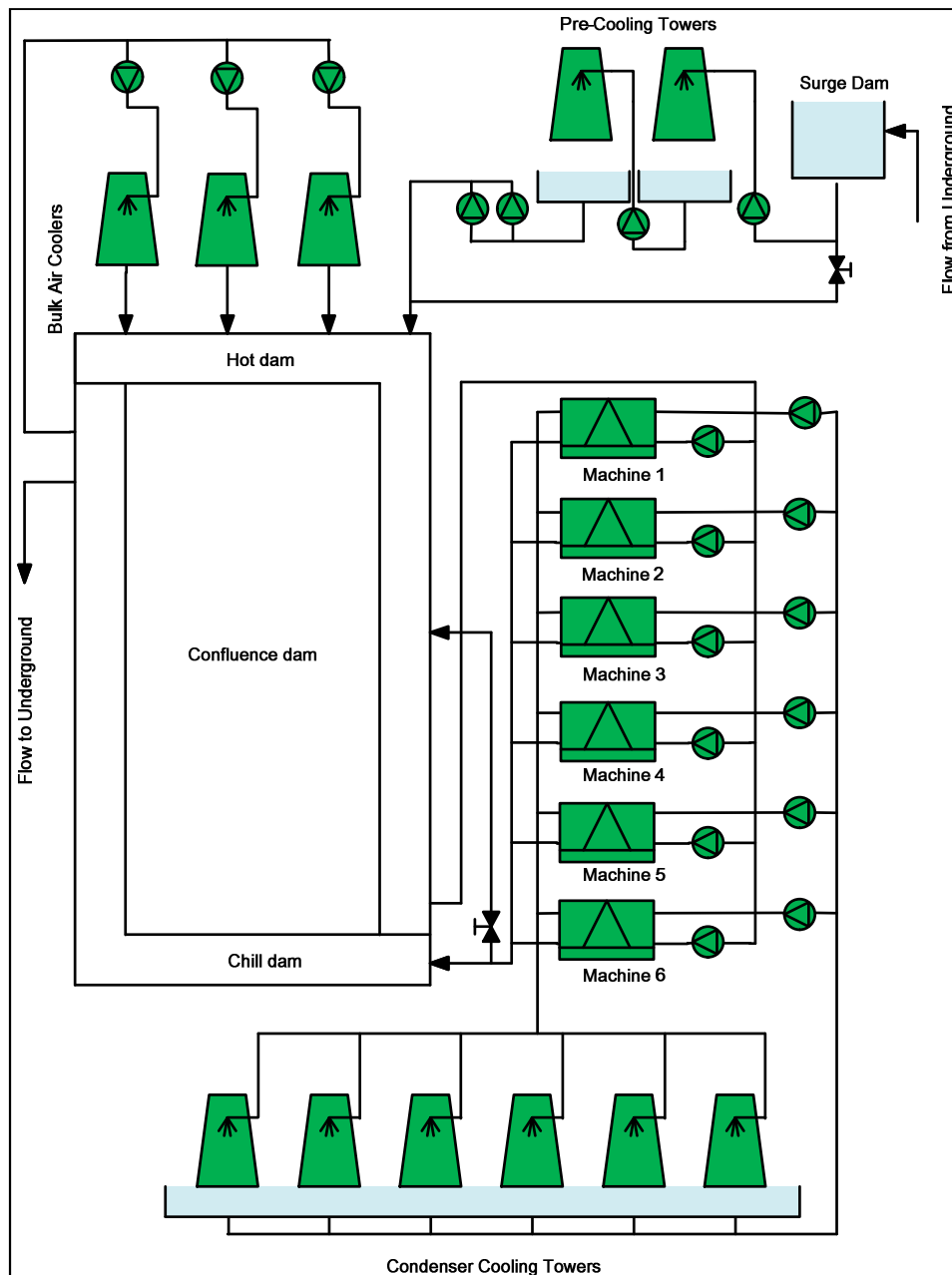


Figure 9: Mine A - cooling system layout before implementation

4.2.1.2 Water reticulation

20 MI/day of hot process water is pumped from underground into the surge dam at a temperature of approximately 26 °C during an average weekday. This dam is shown in *Figure 10* to the left and in the forefront of the picture. The water is pumped through the two pre-cooling towers in series and then transferred to the hot dam by means of transfer pumps from the second towers' sump. The pre-cooling towers are displayed to the right and to the background of *Figure 10*. The evaporator pumps pump water from the hot side of the

confluence dam through the refrigeration machines' evaporators, where the water is cooled to 3 °C, into the chill dam.



Figure 10: Mine A - surge dam and pre-cool towers

Water at 3 °C is sent underground at a rate of approximately 200 l/s. Each of the three BACs uses 250 l/s of chilled water at 3 °C, with a water outlet temperature of approximately 9 °C, which goes directly into the hot dam. The BACs are shown in the back of *Figure 11*.

In the middle of the hot and chill dam, there is a confluence dam where hot and cold water is allowed to mix. Chilled water from the chill dam is allowed to flow over into the confluence dam as well as hot water from the hot dam, depending on storage requirements. This is however a dynamic process and the confluence dam never reaches a steady state mixed temperature but rather cold water on the cold side and warm water on the warm side. There exists a blended region between the warm and cold sides, which can be controlled by changing the ratio of hot water to cold water entering the confluence dam. Excess chilled water not sent underground or used by the BACs flows over into the confluence dam. This

water flows to the warm region of the confluence dam contributing to additional pre-cooling of the warm water before it passes through the refrigeration machines.

Chilled water as well as warm water can be stored in the same dam whenever required. The precise functioning of the dam is not discussed in detail, as it falls beyond the scope of this document. The dam is shown in the forefront of *Figure 11*.

A back pass valve between the transfer pumps and the hot dam allows pre-cooled water to be circulated back into the surge dam. The hot and chill dams also have a back pass valve between them allowing chilled water to be passed back to the hot dam. This second back pass valve was part of a previous load shifting initiative implemented on the system. The control was no longer in operation due to the maintenance on the machines, for switching them on and off, making it unviable.



Figure 11: Mine A - hot, cold and confluence dam and BACs

4.2.1.3 Cooling system operational parameters

The following parameters were obtained via measurements and site verifications on the mine. The surface refrigeration machine operational parameters are displayed in *Table 2*.

Table 2: Mine A - refrigeration machine operational parameters

Description	Value
Number of fridge plants	6
Make	York
Compressor Type	Centrifugal
Refrigerant	R134a
Voltage [V]	6600
Cooling Capacity [kW]	6500
COP	5.5
Evaporator Outlet Temperature [°C]	3
Condenser Inlet Temperature [°C]	27
Evaporator Water Flow [kg/s]	250
Condenser Water Flow [kg/s]	450
Evaporator Pump Motor Rating [kW]	110
Number of Evaporator Pumps	6
Condenser Pump Motor Rating [kW]	160
Number of Condenser Pumps	6

The pre-cooling tower operational parameters are displayed in *Table 3*.

Table 3: Mine A - pre-cooling tower operational parameters

Description	Value
Number of Cooling Towers	2
Water Inlet Temperature [°C]	30
Water Outlet Temperature [°C]	24
Water Flow [kg/s]	360
Air Inlet Wet Bulb Temperature [°C]	22
Pump Motor Rating [kW]	70
Number of Pumps	2
Transfer Pump Motor Rating [kW]	40
Number of Pumps	2

The surface condenser cooling tower operational parameters are displayed in *Table 4*.

Table 4: Mine A - condenser cooling tower operational parameters

Description	Value
Number of Cooling Towers	6
Water Inlet Temperature [°C]	31
Water Outlet Temperature [°C]	27.5
Water Flow [kg/s]	450
Air Inlet Wet Bulb Temperature [°C]	22

The surface BAC operational parameters are displayed in *Table 5*.

Table 5: Mine A - BAC operational parameters

Description	Value
Number of BACs	3
Water Inlet Temperature [°C]	3
Water Outlet Temperature [°C]	9
Water Flow [kg/s]	250
Air Flow [kg/s]	250
Air Inlet Wet Bulb Temperature [°C]	18
Air Outlet Wet Bulb Temperature [°C]	7
Pump Motor Rating [kW]	75
Number of Pumps	3

4.2.2 Energy saving strategies

The following energy saving strategies were identified to ensure sustainable energy savings on the surface cooling system of Mine A.

4.2.2.1 Pre-cooling tower control

To balance the water flow between the pre-cooling towers' sump and the hot dam, the mine uses a back pass valve. This valve serves to re-circulate the excess pre-cooled water, not being transferred to the hot dam, back into the surge dam and reducing the temperature of the surge dam. This however, does not cause cooler water at the hot dam because decreasing the

surge dam temperature means reducing the inlet temperature of the pre-cooling tower; consequently leading to a reduction in the range of the tower instead of decreasing the outlet temperature of the cooling tower. This is due to the pre-cooling towers' potential cooling duty being governed by the ambient air conditions and can only cool the water to a certain temperature, thus in effect using unnecessary pump electrical energy to pump more water than required through the pre-cooling tower at maximum flow all of the time.

Variable speed drives (VSDs) will be used on the tower supply pumps' motors as well as the transfer pumps' motors to reduce and control the water flow. The VSDs will maintain a constant pre-cooling tower sump level with the two supply pumps as well as the desired hot dam level with the transfer pumps. The back pass valve will be decommissioned and closed allowing no pre-cooled water to pass back into the surge dam.

4.2.2.2 Evaporator flow control

Through utilising VSDs on the six evaporator pumps, the chill dam can be controlled at a desired user specified level. This will lead to a reduction in the flow between the minimum flow (before the machine will trip or to a maximum reduction of 40 %) and the design flow as the maximum. All the choking valves used to calibrate the system flow should be opened, as the flow is now controlled via variation of the pump speed instead of running at full speed only.

The minimum evaporator flow that will be allowed by the machine before it trips will automatically fall above the minimum ideal flow of 1 m/s, stated in Section 2.2, as this is the standard procedure by which the refrigeration machine manufacturers calibrate the machines. Care should however be taken in the event of altering these trip conditions on the machine PLC to avoid running into these unwanted low flow regions. These machines are typically designed for water velocities of 2 m/s; hence a reduction of 40 % in the water flow rate will not result in flow velocities of 1 m/s or less.

4.2.2.3 Condenser water flow control

The condenser water flow will be controlled to maintain a specified design cooling water temperature difference across the condenser by utilising VSDs on the condenser water

pumps. This allows the overall condenser water temperature to reduce, allowing for a reduction in compressor power of approximately 3 % per °C ⁽¹³⁾. This reduction in overall condenser temperature will be obtained by the reduction in water flow through the condenser cooling towers resulting in improved cooling tower utilisation. The choking valves used to calibrate the flow through the condenser should be opened as the flow will now be controlled by variation of the pump speed.

The condenser flow will automatically adapt to keep the temperature drop across the condenser constant as the climatic conditions change. The minimum flow will also be calibrated to remain inside the machine trip conditions, while the maximum flow will be set to the machine designed flow specifications.

4.2.2.4 Bulk air cooler water flow control

The water flow through the bulk air cooler (BAC) will be controlled using VSDs on the three BAC feed pumps. The average summer enthalpy was determined to be 70 kJ/kg while it was 25 kJ/kg during the winter. The BAC water flow will be controlled to run at design flow for any enthalpy measurement of 70 kJ/kg or higher (representing average summer conditions) and linearly reducing to no flow at 25 kJ/kg (representing winter ambient conditions) during which time the BACs are always switched off. This control of BAC water flow will result in a reduction of chilled water demand as the climatic enthalpy reduces and resulting in an energy saving.

4.2.3 Baseline simulation

An annual baseline simulation model was configured for the surface cooling system to be used as reference for the energy savings during the energy saving investigation. For these cooling systems, it is preferable to consider the annual values, as the saving potential will be a function of the climatic conditions which makes one full cycle per annum.

4.2.3.1 Simulation model

The baseline simulation boundaries are displayed in *Table 6*. The system operation was split into a summer and winter operation because the BACs are turned off for the duration of the

winter period due to the ambient air being suitable for underground mining. During the winter time a maximum of two refrigeration machines are required to supply the 200 l/s required for mining, whereas between four and six machines are required during summer depending on the cooling demand.

Table 6: Mine A - cooling system baseline simulation boundaries

Description	Sep - May	Jun - Aug
Hourly climate data	Klerksdorp	Klerksdorp
Average chilled water to underground flow [kg/s]	200	200
Chilled water supply temperature [°C]	3	3
Average evaporator flow [kg/s]	250 x 6	250 x 2
Fridge plants in operation	6	2
Condenser cooling towers in operation	6	6
Pre-cooling towers in operation	2	2
BACs in operation	3	0
Surge dam water temperature [°C]	26	26

The complete system was modelled using EfluxS as a real time dynamic integrated system simulator to simulate the cooling system for every hour of the year. The system layout as used in the simulation model is shown in *Figure 12*.

The six refrigeration machines were modelled as two machines, one representing the capacity of four machines and the other the capacity of two machines, to allow for the winter period during which only two machines are required.

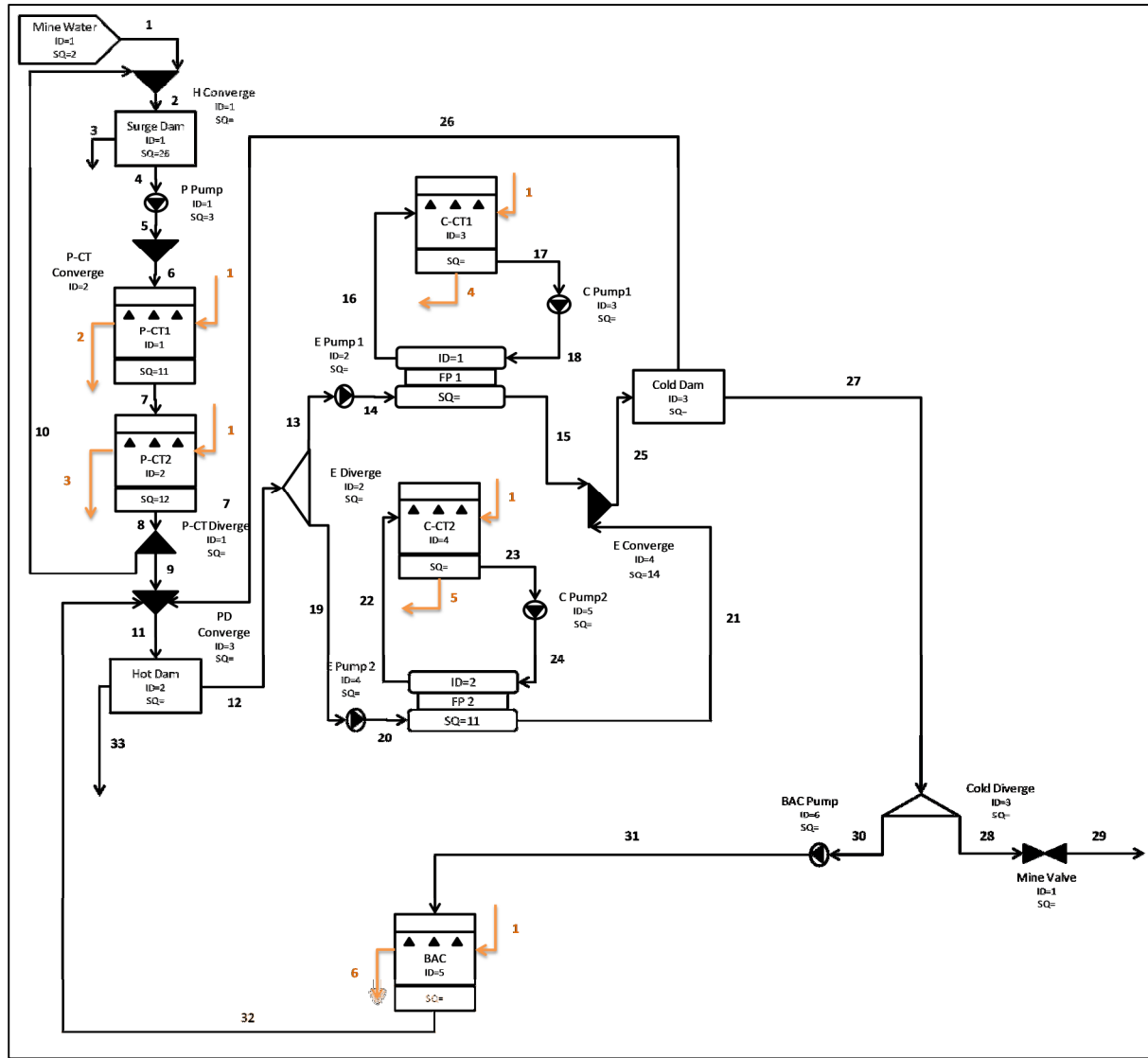


Figure 12: Mine A - simulation cooling system layout

4.2.3.2 Baseline simulation results

The daily average power usage profile of the simulated baseline is shown in *Figure 13*. There is a distinct reduction in power consumption during the winter months due to the BACs being switched off completely during this period of time. The BACs alone consume two thirds of the cooling capacity of the six refrigeration machines.

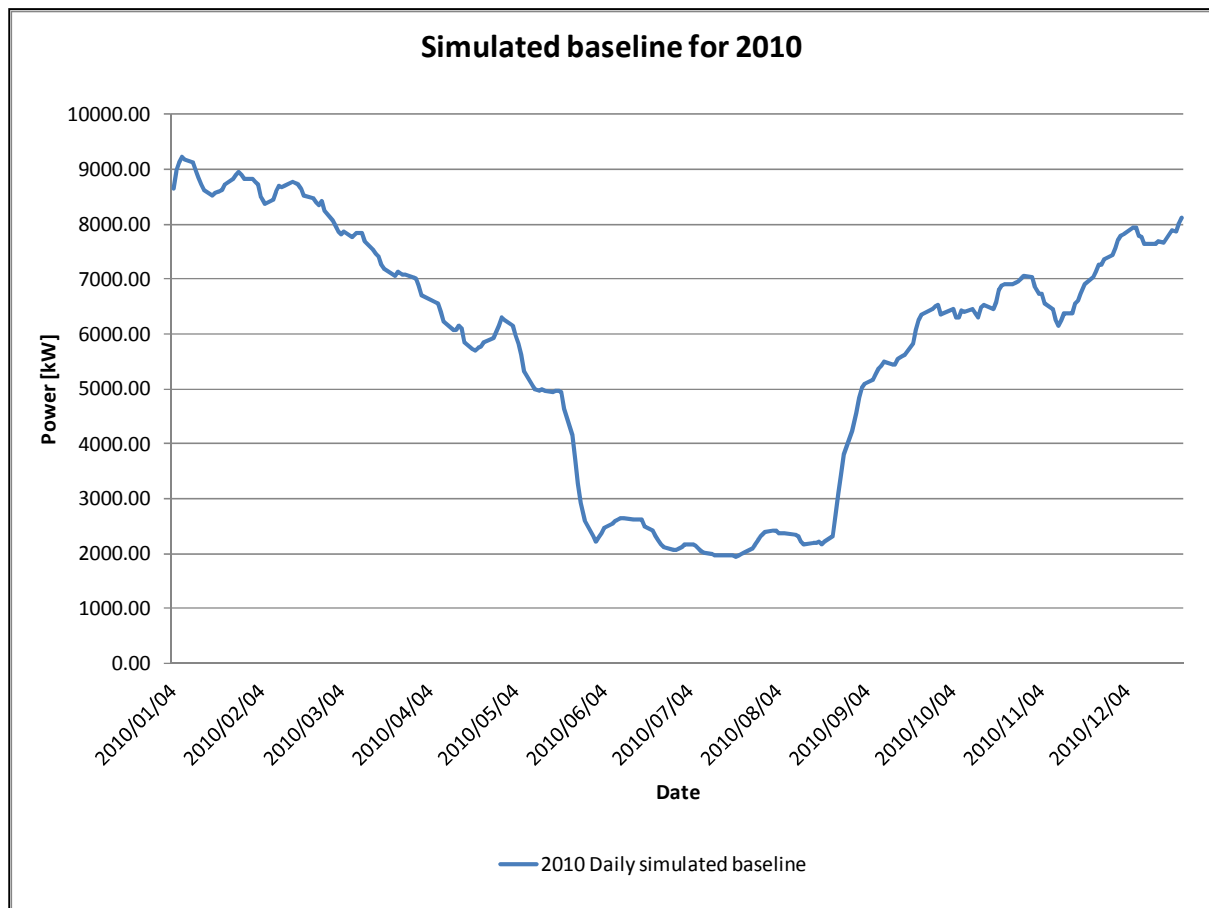


Figure 13: Mine A - simulated daily power consumption for 2010

4.2.4 Simulation verification

After the simulation model was developed, in accordance with the plant system operational data received from the mine, the results were verified by comparing it to the measured power consumption for 2010. The daily average power usage measured was compared to the simulated results as shown in *Figure 14*. From *Figure 14*, it can be seen that the simulated results follow the annual power profile closely in terms of the measured data. The simulated and measured results, as well as the percentage difference for each month, are displayed in *Table 7*. The significant reduction in power consumption at the end of the graph for the measured data is due to the mine shutting down partially during December.

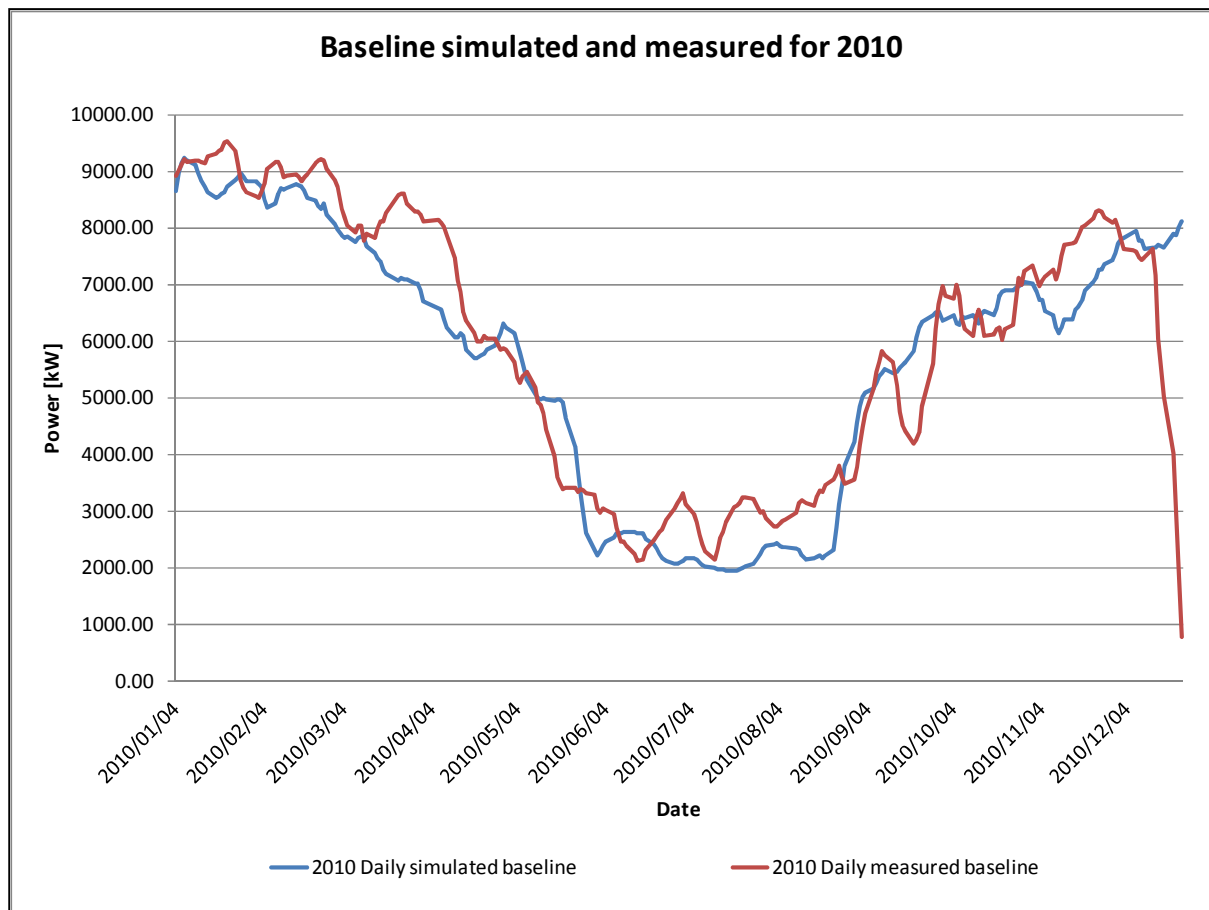


Figure 14: Mine A - simulated and measured daily power consumption for 2010

Table 7: Mine A - 2010 Simulated and measured monthly baseline

Month	2010 Simulated baseline [kW]	2010 Measured baseline [kW]	% Difference
January	8722.56	9201.05	5.20
February	8688.13	8840.05	1.72
March	7627.15	8247.93	7.53
April	6201.18	7092.52	12.57
May	5250.67	4639.84	-13.16
June	2392.79	2684.67	10.87
July	2033.99	2895.29	29.75
August	2318.09	3154.20	26.51
September	5473.23	4729.90	-15.72
October	6543.12	6413.73	-2.02
November	6718.27	7587.76	11.46
December	7829.21	5546.30	-41.16
Average	5801.54	5897.22	1.62

4.2.5 Energy savings simulation

The energy saving strategies identified were simulated in the following order:

1. Pre-cooling tower control
2. BAC water flow control
3. Evaporator flow control
4. Condenser flow control

The same model in EfluxS, developed to establish the baseline, was utilised for the energy savings simulations. However, the control for the four abovementioned electrical energy saving strategies was included one at a time to enable the determining of the resulting impact that each component has on the overall power usage of the entire system.

The following control boundaries were simulated for the separate electrical energy saving strategies:

1. The back pass valve returning pre-cooled water to the surge dam was decommissioned. The hot dam level and pre-cooling sump levels were maintained at 80 % using VSDs on the Pre-cooling tower supply pumps and hot dam transfer pumps.
2. The flow through the BAC was controlled using VSDs on the supply pumps. It was modelled to run at design flow for ambient enthalpies of 70 kJ/kg and above while decreasing the flow linearly to no flow at ambient air enthalpies of 25 kJ/kg and below.
3. Maintain a chill dam level of 80 % using VSDs on the refrigeration machine evaporator pumps. Limiting the minimum flow to 150 l/s to avoid tripping the refrigeration machine under minimum flow trip conditions and limiting the maximum flow to 250 l/s as the machine design flow condition.
4. The refrigeration machine condenser flow was controlled to maintain a temperature difference of 5 °C across the condenser. Limiting the flow minimum to 360 l/s to avoid tripping the refrigeration machine on low flow trip conditions and limiting the maximum flow to 450 l/s as the machine design flow condition.

4.2.6 Energy saving results

After comparing the simulated energy saving strategies' energy usage to the simulated baseline, it was found that the average potential yearly energy saving of the complete cooling system would be 1 773 kW. *Figure 15* shows the daily average baseline and energy optimised profiles. From this graph it is apparent that the saving potential decreases substantially during the winter period when the BACs are switched off. The largest savings are recorded during the autumn and spring times when the BAC flow is controlled by the ambient air enthalpy and gradually decreases and increases in conjunction with the climate. The average potential monthly electricity savings for the entire cooling system, over a twelve month period would be approximately 1.7 MW as displayed in *Table 8*.

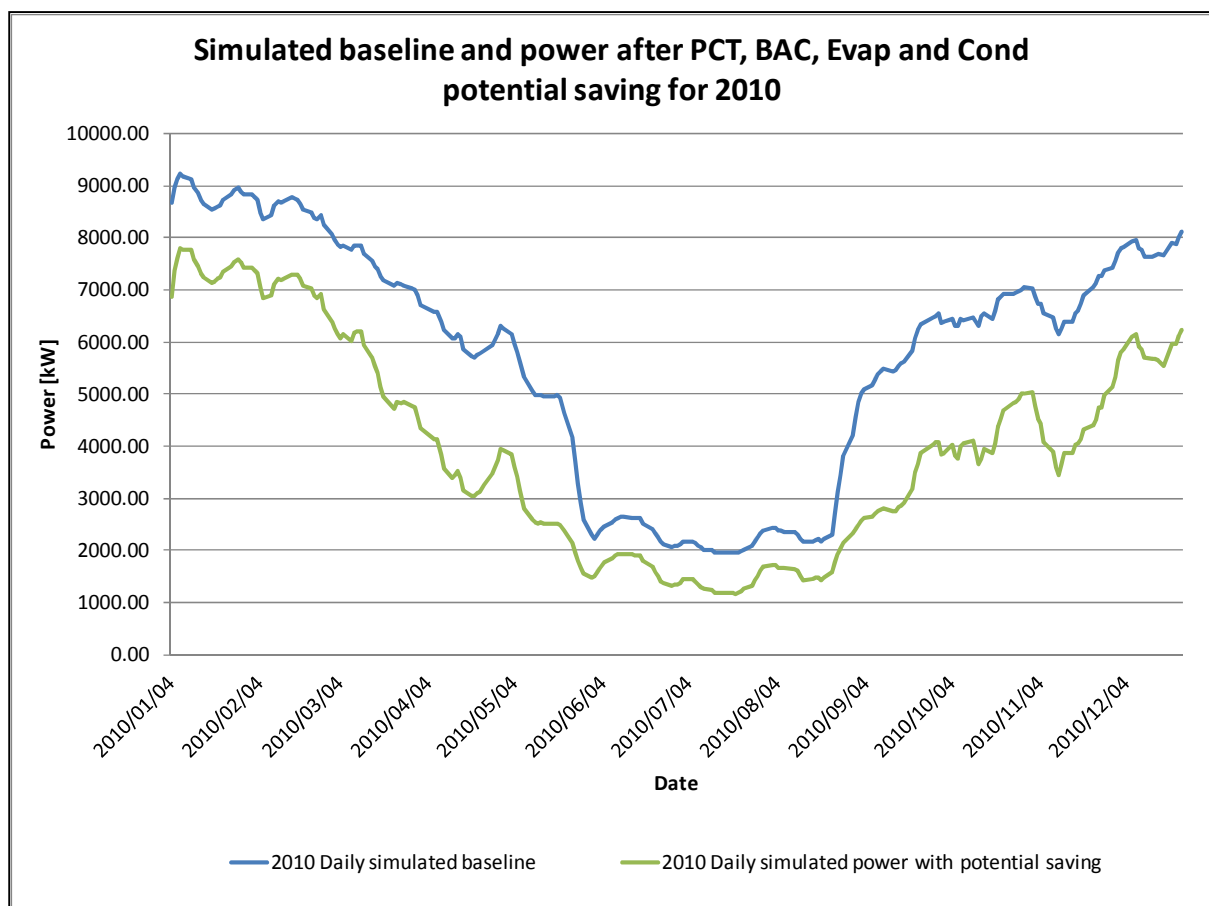


Figure 15: Mine A - daily simulated baseline and total potential saving profile for 2010

Table 8: Mine A - proposed energy savings for 2010

Month	2010 Simulated baseline [kW]	2010 Simulated power after proposed saving [kW]	Proposed Saving [kW]
January	8722.56	7193.23	1529.33
February	8688.13	7240.59	1447.54
March	7658.88	5785.92	1872.96
April	6201.18	3647.59	2553.60
May	5250.67	2911.14	2339.53
June	2392.79	1674.83	717.96
July	2033.99	1277.17	756.82
August	2318.09	1604.82	713.27
September	5473.23	2922.72	2550.52
October	6543.12	4088.95	2454.17
November	6718.27	4287.49	2430.78
December	7829.21	5871.75	1957.46
Average	5797.06	4024.30	1772.76

4.3 Case study 2: Mine B

4.3.1 System description

4.3.1.1 Layout

The mine cooling system consists of a refrigeration plant and eight pre-cooling towers situated in two separate banks of four towers per bank above the pre-cooling dam. In addition, there is one hot and one chill dam with an overflow from the chill dam into the pre-cooling dam. Three BACs, of which two supply the main shaft and one the rock shaft, comprise the complete cooling system. *Figure 16* shows the basic layout of the system.

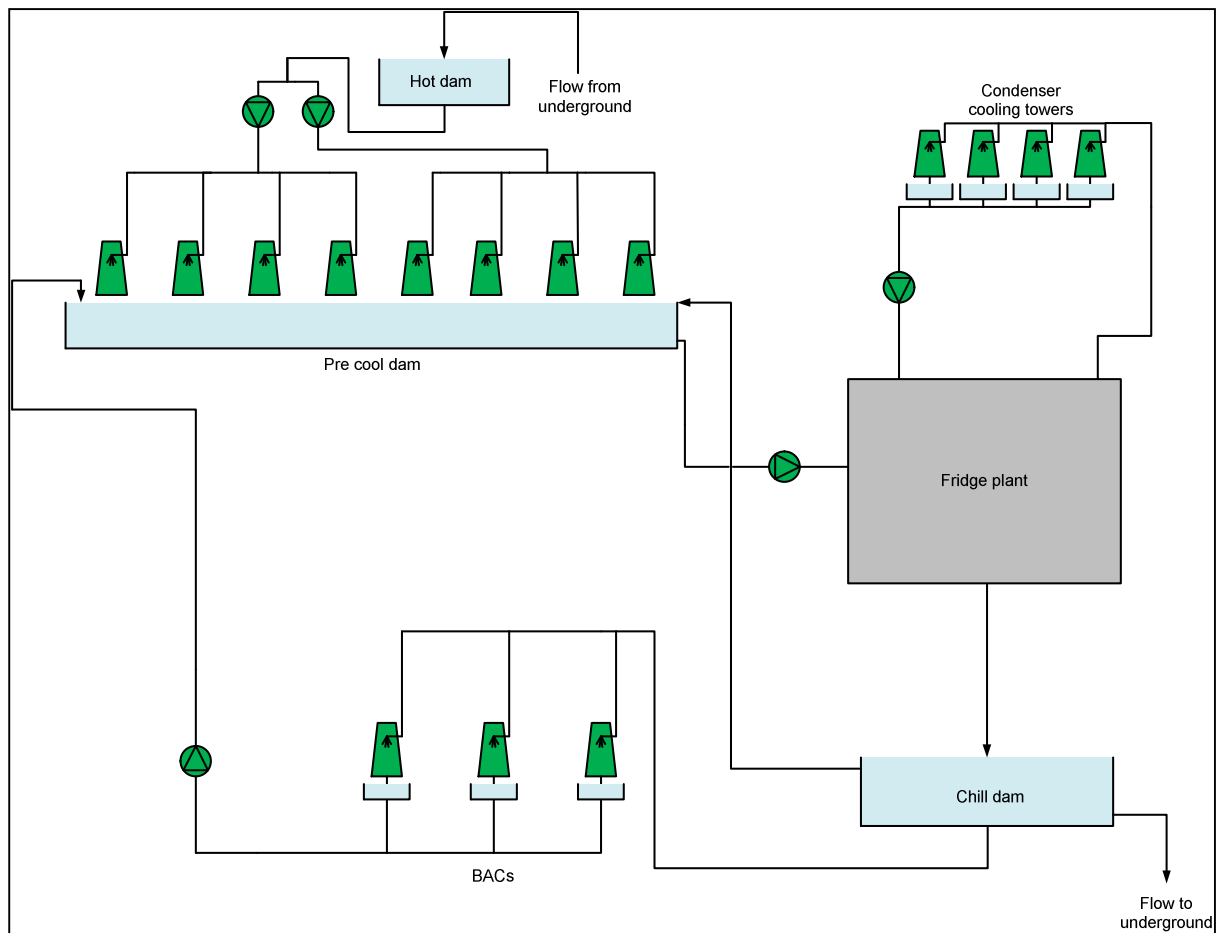


Figure 16: Mine B - cooling system layout

4.3.1.2 Water reticulation

Warm process water is pumped from underground at approximately 30 °C into the hot water dam. The warm water is pumped through the pre-cooling towers at an average rate of 300 l/s and into the pre-cool dam. From the pre-cool dam it is transferred through the refrigeration plant, which cools the water to 5 °C, into the chill dam. Water from the chill dam is sent underground at a rate of 250 l/s and to the BACs at a rate of 150 l/s. Return water from the BACs is pumped into the pre-cool dam. The existing pre-cooling towers are shown in *Figure 17* and *Figure 18*.



Figure 17: Mine B - existing pre-cooling towers (1)



Figure 18: Mine B - existing pre-cooling towers (2)

4.3.1.3 Pre-cooling tower operational parameters

The following pre-cooling tower parameters, displayed in *Table 9*, were measured on site.

Table 9: Mine B - pre-cooling tower operational parameters

Description	Surface
Number of Cooling Towers	8
Water Inlet Temperature [°C]	30
Water Outlet Temperature [°C]	26
Water Flow [kg/s]	300
Air Inlet Wet Bulb Temperature [°C]	17
Pump Motor Rating [kW]	75
Number of Pumps	2

4.3.1.4 Refrigeration machine operational parameters

The refrigeration plant operational parameters that were measured on site are displayed in *Table 10*.

Table 10: Mine B - refrigeration plant operational parameters

Description	Surface
Number of refrigeration machines	4
Installed electrical capacity per machine [kW]	1250
Water Inlet Temperature [°C]	23
Water Outlet Temperature [°C]	5
Water Flow [kg/s]	300
COP	6

4.3.2 Energy saving strategy

A single energy saving strategy was identified for application to this system. The identified strategy was to improve the pre-cooling tower efficiency. The most effective way to achieve this was to replace the current units.

4.3.3 Baseline simulation

An annual baseline simulation model was configured for the pre-cooling tower to be used as reference for the electrical energy savings during the energy savings investigation. It is preferable to consider the annual values as the saving potential will be a function of the climatic conditions which makes one full cycle per annum.

4.3.3.1 Simulation model

The simulation boundaries for the baseline are displayed in *Table 11*. The system was modelled with one tower and one pump representing all eight towers with the two supply pumps.

Table 11: Mine B - pre-cooling tower baseline simulation boundaries

Description	Jan - Dec
Hourly climate data	Klerksdorp
Average water flow through pre-cooling towers [l/s]	300
Hot water dam temperature [°C]	30

The pre-cooling tower baseline was simulated using EfluxS, as real time dynamic integrated simulator, simulating the pre-cool outlet temperature for each hour of one year. This was achieved by using the measured operational parameters displayed in *Table 9* in conjunction with the simulation boundaries displayed in *Table 11*. The system layout that was used is presented in *Figure 19*.

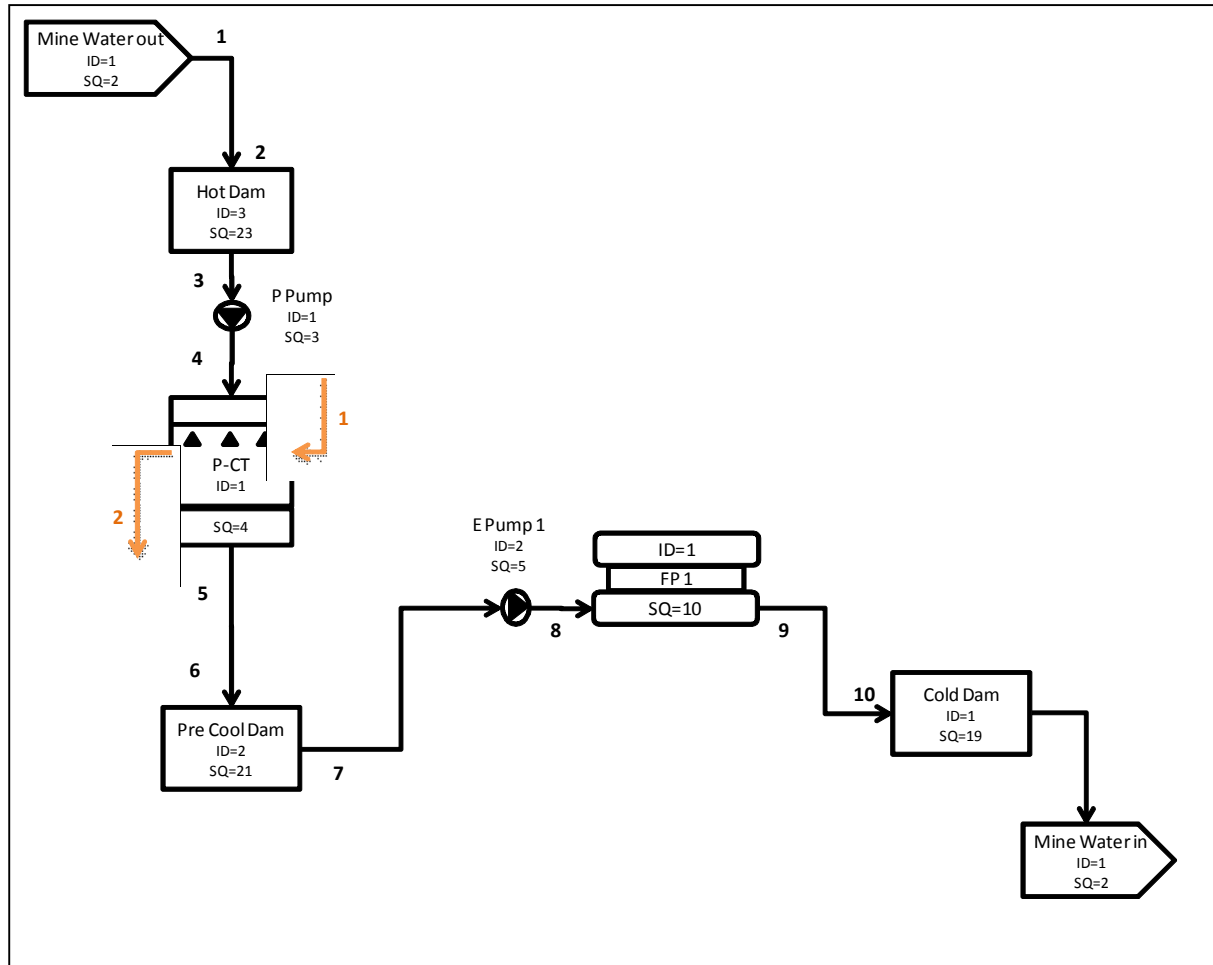


Figure 19: Mine B - simulation pre-cooling tower layout

4.3.3.2 Baseline simulation results

The daily average pre-cooling tower cooling results shown as tower water outlet temperature, as obtained by using the simulation model, is shown in *Figure 20*. The ambient enthalpy is also illustrated in the graph for comparison. From *Figure 20* it is apparent that the outlet temperature poorly reflects the change in ambient enthalpy. This is due to the poor physical condition of the tower and the minimal percentage of cooling duty it performs.

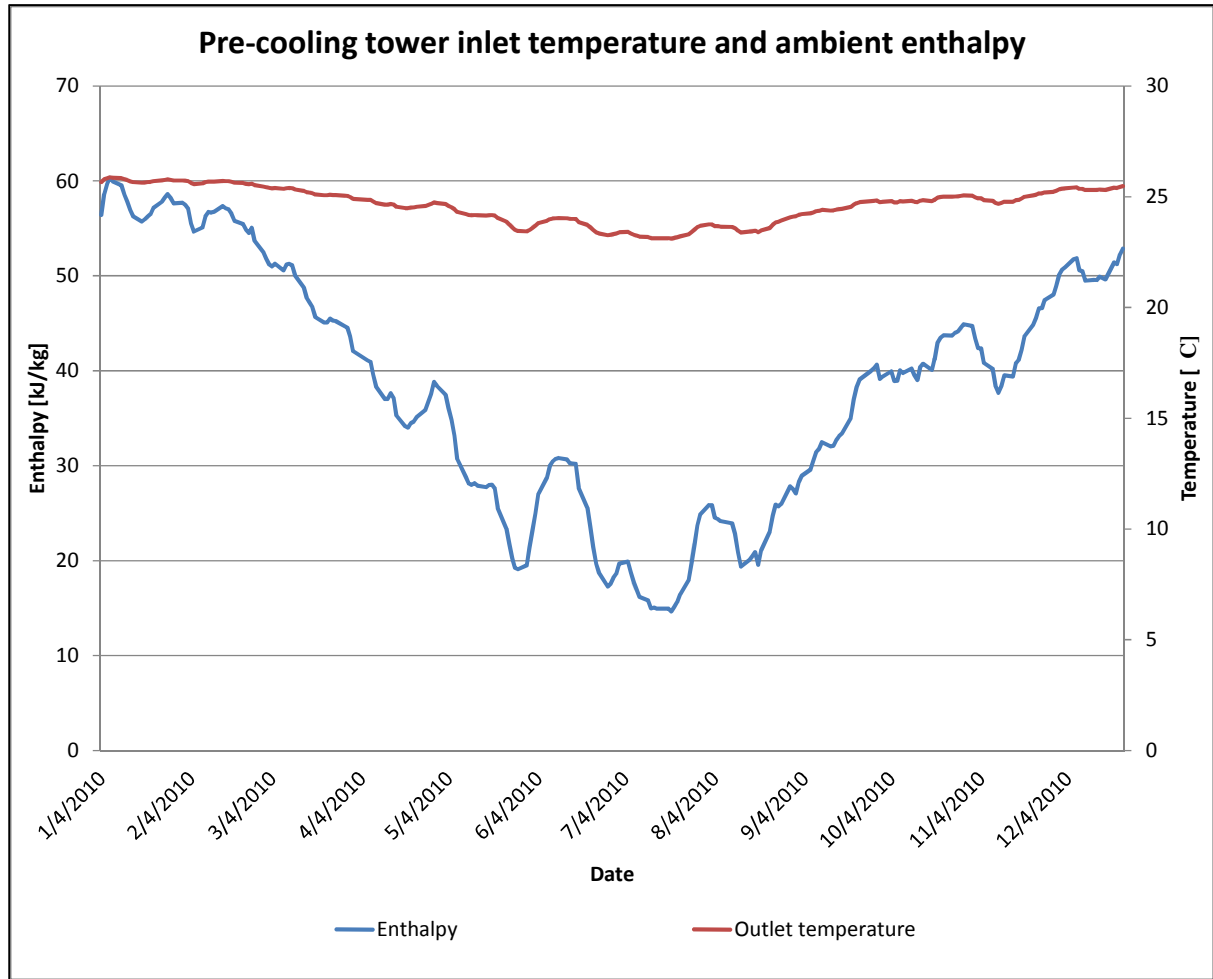


Figure 20: Mine B - pre-cooling tower performance baseline

4.3.4 Energy saving simulation and results

The same simulation model was used, but in this case the model was adjusted to achieve the pre-cooling tower design specifications displayed in *Table 12*. *Figure 21* shows the newly suggested outlet temperature along with the ambient enthalpy superimposed as it was obtained from the simulation model. Using the refrigeration plant operational parameters displayed in *Table 10*, the simulation model was able to calculate the saving potential in the case of the pre-cooling towers operating at design conditions. The simulated monthly average savings are displayed in *Table 13* with an average saving of 1.28 MW over an annual period.

Table 12: Mine B - pre-cooling tower design specification

Description	Surface
Water Inlet Temperature [°C]	30
Water Outlet Temperature [°C]	19.7
Water Flow [kg/s]	300
Air Inlet Wet Bulb Temperature [°C]	17

From *Figure 21* it can be seen that the pre-cooling tower outlet temperature now follows the ambient enthalpy.

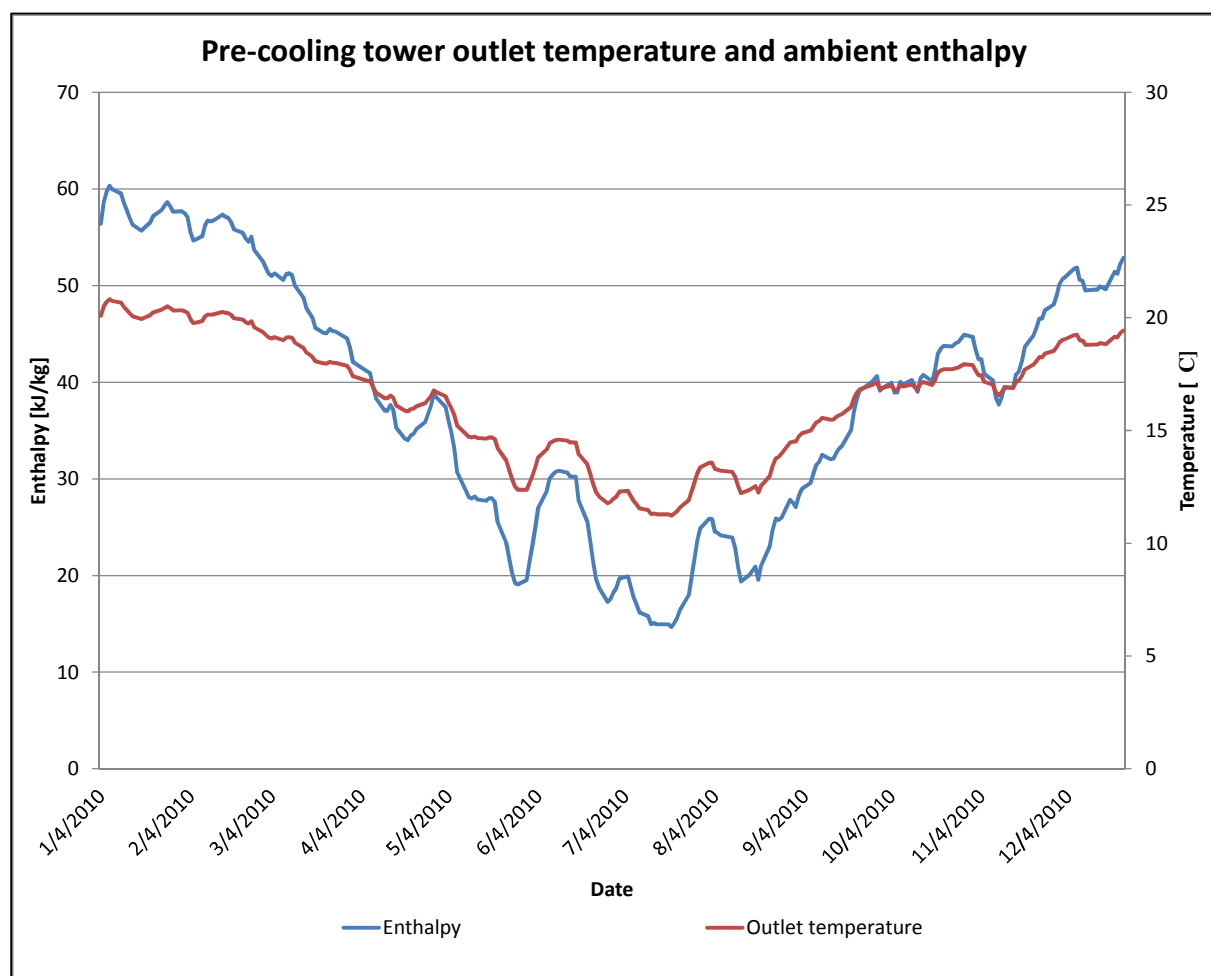


Figure 21: Mine B - optimised pre-cooling outlet temperature

Table 13: Mine B - proposed energy savings

Month	Proposed Saving [kW]
January	1001.49
February	999.05
March	1088.99
April	1308.74
May	1522.51
June	1456.02
July	1601.39
August	1473.43
September	1460.16
October	1244.15
November	1219.97
December	1064.03
Average	1287.29

4.3.5 Simulation verification

The simulation model was corroborated by verifying the savings suggested by the simulation model. This was achieved using first principle thermodynamic equations, presented in Section 4.3.5.1 ⁽²⁸⁾ and was subsequently compared to the suggested saving obtained from the simulation model.

The present operational outlet temperature measured as displayed in *Table 9* was compared to that of the suggested design outlet temperature displayed in *Table 12*. The difference between these two temperatures would reflect the additional cooling that can be achieved in the event of the pre-cooling tower operating at design specifications. The additional cooling was calculated using *Equation 14*. The difference between the existing pre-cooling towers' performance and the design performance was then converted to the required electrical energy to perform the cooling by using *Equation 13*. This represents the electrical saving achievable on the refrigeration system when obtaining design performance from the pre-cooling towers.

4.3.5.1 Equations⁽²⁸⁾

$$COP = \frac{\dot{Q}_L}{P_{wr}} \quad \text{Equation 13}$$

$$\dot{Q}_L = \dot{m} \cdot C_p(T_i - T_o) \quad \text{Equation 14}$$

4.3.5.2 Calculation

$$\dot{Q}_L = \dot{m} \cdot C_p(T_{out\,operational} - T_{out\,design}) = (300)(4.18)(26 - 19.7) = 7900.2\,kW$$

$$P_{wr\,saving} = \frac{\dot{Q}_L}{COP} = \frac{7900.2}{6} = 1316.7\,kW$$

4.3.6 Comparison between calculated and simulated saving

The saving calculated in Section 4.3.5.2 was compared to the saving from the simulation model in Section 4.3.4 with a difference of 2.2 % relative to the calculated saving to the conservative side. This deviation from the simulated results could be due to average temperature and flow assumptions that were made to simplify the calculation in 4.3.5.2. This can be considered to be acceptable and verifies that the simulation yields a conservative result with only a small discrepancy. An electrical energy saving of 25 % can be expected on the refrigeration system.

The simulation verification was successful and the results obtained by simulation could be assumed to be an accurate reflection of the actual system performance.

4.4 Conclusion

In both case studies the simulation model provided satisfactory results with an acceptable, immaterial discrepancy. In both cases the discrepancy was on the conservative side. First principle calculations were only used for case study two due to it being less complex and dealt with fewer variables than that of case study one where first principle calculations would be impractical.