

**AN INVESTIGATION INTO THE DSM AND ENERGY EFFICIENCY
POTENTIAL OF A MODULAR UNDERGROUND AIR COOLING UNIT
APPLIED IN THE SOUTH AFRICAN MINING INDUSTRY**

BY

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TITLE: AN INVESTIGATION INTO THE DSM AND ENERGY EFFICIENCY POTENTIAL OF A MODULAR UNDERGROUND AIR COOLING UNIT APPLIED IN THE SOUTH AFRICAN MINING INDUSTRY

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ABSTRACT

The South African mining industry is faced with the depletion of minerals and coal at current mining depths. This is forcing the industry to expand operations to greater depths where ample ore reserves are located. This expansion combined with the demand for increased production rates puts excessive strain on the ventilation and cooling systems of mines. The problem is that mining will eventually reach depths where current methods of ventilation and cooling will no longer be economical and practical to provide an acceptable working environment.

To solve this problem new technologies or alternative cooling methods are required. One such concept is called spot cooling. This entails cooling the air in a remote location rather than cooling the entire environment leading up to that location. Until now spot cooling was primarily done using chilled water but a significant limitation of this approach is that the cooling effectiveness is proportional to the chilled water temperature.

An alternative solution is the use of a localised refrigeration plant acting as a spot cooler. This is different from a standard underground refrigeration plant since it is modular with a smaller cooling capacity and is completely mobile so that it can be deployed at different locations as the need arises.

The purpose of this study is to develop such a prototype modular air cooling unit (ACU) to establish a suitable working environment and to evaluate the techno-economic impact if it is applied on a wide scale for deep mine expansion. Furthermore, from an electrical supply side the potential of the ACU to contribute to energy efficiency (EE) and demand side management (DSM) initiatives is also investigated.

A nominal 80kW prototype ACU was designed, manufactured and tested in both laboratory conditions and within an underground mining environment. A distinct advantage of the unit is that it can operate efficiently with high inlet water temperatures. This provides enhanced flexibility compared to existing technologies since it can utilise normal service water as heat sink as opposed to chilled water as a source of cooling.

A system simulation model was developed to investigate the deep mining techno-economic impact as well as the EE and DSM potential of the ACU compared to existing chilled water technology.

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Information and data from a case study mine were used to simulate different possible cooling configurations for deep mine expansion.

From the results two configurations emerged that show the best potential for deep mine application, namely an underground chiller plant combined with chilled water cooling cars (CWCs) or a hot underground dam with ACUs. Of these two configurations only the ACU configuration holds significant EE and DSM potential due to a reduction in the total power requirement and it is also by far the most cost effective. This could provide substantial benefits for both the mining industry and the South African electrical supply utility, Eskom.

From the results of this study it can be concluded that the ACU technology offers an energy efficient, cost effective and practical alternative to conventional cooling methods for deep mine expansion where the establishment of an acceptable working environment is a major concern.

TITEL: 'N ONDERSOEK NA DIE AANVRAAGBESTUUR EN ENERGIE BESPARINGSPOTENSIAAL
VAN 'N MODULÊRE ONDERGRONDSE LUGVERKOELINGSEENHEID VIR DIE
SUID-AFRIKAANSE MYNBEDRYF

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OPSOMMING

Die Suid Afrikaanse mynbedryf ondervind die probleem dat minerale en steenkool op huidige myndieptes uitgeput raak. Dit forseer die industrie om uit te brei na dieper areas waar ruim ertsreserwes steeds voorkom. Hierdie uitbreiding gekombineerd met 'n aanvraag na 'n verhoging in produksie plaas 'n addisionele las op die ventilasie- en verkoelingstelsels van myne. Die probleem is dat die mynbedryf dieptes gaan bereik waar huidige metodes van ventilasie en verkoeling nie langer ekonomies en prakties 'n geskikte werkomgewing kan skep nie.

Om hierdie probleem op te los is nuwe tegnologieë of alternatiewe verkoelingsmetodes nodig. Een so 'n konsep is lokale areaverkoeling. Dit behels die verkoeling van lug in 'n afgeleë area eerder as om die totale omgewing wat lei na die area te probeer verkoel. Tot op hede is lokale areaverkoeling hoofsaaklik gedoen met verkilde water. 'n Beperking van hierdie benadering is dat effektiewe verkoeling proporsioneel is aan die temperatuur van die verkilde water.

'n Alternatiewe oplossing is die gebruik van 'n lokale lugverkoelingseenheid in hierdie areas. Die eenheid verskil van standaard ondergrondse verkoelingsaanlegte hoofsaaklik omdat dit modulêr is met 'n kleiner verkoelingskapasiteit. Dit is verder ook ten volle mobiel vir gebruik in verskillende areas soos wat die behoefte ontstaan.

Die doel van hierdie studie is om 'n prototipe modulêre lugverkoelingseenheid te ontwikkel vir die skep van 'n geskikte werksomgewing en ook om die tegno-ekonomiese impak te evalueer wanneer dit op grootskaal geïmplementeer word vir diepmyne uitbreiding. Vanaf 'n elektriese voorsieningskant word die energiebesparings en aanvraagkanbestuur potensiaal van die eenheid ook ondersoek.

'n Nominaal 80kW eenheid is ontwerp, vervaardig en getoets in beide laboratorium kondisies en in 'n ondergrondse mynomgewing. 'n Unieke voordeel van die eenheid is die vermoë om selfs by hoë inlaat watertemperatuur 'n hoë werkverrigting te handhaaf. Dit ontsluit nuwe moontlikhede aangesien die eenheid normale dienswater as hitteput kan gebruik in teenstelling met konvensionele eenhede wat afhanklik is van verkilde water.

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'n Stelselsimulasiemodel is ontwikkel om die tegno-ekonomiese, energiebesparings en aanvraagkantbestuur potensiaal van die eenheid te vergelyk met bestaande verkilde water tegnologie. Inligting van 'n gevallestudie myn is gebruik om verskillende verkoelingskonfigurasies vir diepmyn uitbreiding te simuleer.

Uit die simulaties het twee hoofkonfigurasies die meeste potensiaal getoon vir diepmyn applikasie, naamlik 'n ondergrondse verkoelingsaanleg met verkilde water eenhede en die nuwe modulêre verkoelingseenhede gekoppel aan 'n warm ondergrondse dam. Van hierdie twee konfigurasies toon slegs die modulêre verkoelingseenheidkonfigurasie enige wesenlike energiebesparings en aanvraagkantbestuur potensiaal as gevolg van 'n verlaging in die totale drywingsinset. Dit is ook by verre die mees koste-effektiewe oplossing. Hierdie eienskappe hou voordeel in vir beide die mynbedryf en die Suid Afrikaanse elektrisiteitsvoorsiener, Eskom.

Uit die resultate van hierdie studie kan die gevolgtrekking gemaak word dat die modulêre verkoelingseenheidtegnologie 'n energie en koste-effektiewe alternatief bied vir konvensionele verkoelingsmetodes in diepmyn uitbreiding waar die vestiging van aanvaarbare werkstoestande uiters belangrik is.

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NOMENCLATURE

A	Area	m^2
c_p	Specific heat at constant pressure	$J/(kg \cdot K)$
d	Depth	km
D	Diameter	m
F_o	Fourier number	-
g	Gravitational acceleration	m/s^2
h	Enthalpy	J/kg
h	Convection heat transfer coefficient	$W/(m^2 \cdot K)$
i	Interest rate	%
k	Thermal conductivity	$W/(m \cdot K)$
\dot{m}	Mass flow rate	kg/s
n	Time	year
p	Pressure	Pa
P	Power input	W
Pr	Prandtl number	-
q, Q	Heat transfer	W
\dot{q}	Volume flow rate	m^3/s
Re	Reynolds number	-
T	Temperature	$^{\circ}C$
V	Velocity	m/s
w	Humidity ratio	kg_w/kg_a
X	Age factor	-
z	Height	m

GREEK SYMBOLS

Δ	Delta or difference	-
η	Efficiency	-
θ	Age	s
ρ	Density	kg/m^3
Σ	Sum of	-

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ABBREVIATIONS

ACU	Air cooling unit
COP	Coefficient of performance
CSIR	The Council for Scientific and Industrial Research
CWC	Chilled water cooling car
DSM	Demand side management
EE	Energy efficiency
EES	Engineering Equation Solver
ESCO	Energy Service Company
ESKOM	South African Electrical Supply Utility
ISO	International Standards Organisation
PBMR	Pebble Bed Modular Reactor
RAW	Return Airway
SPV	Simple present value
VAT	Value added tax
VRT	Virgin rock temperature

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CHAPTER 1

INTRODUCTION

CHAPTER 1

1. INTRODUCTION

1.1 The problem and its setting

Since 1998 there has been increased concern in the South African mining industry regarding underground working conditions. The effect of high temperatures combined with high humidity was identified as a major health hazard and a cause of reduced productivity. Research has shown that at wet bulb temperatures exceeding 27.4°C in stopes and development ends the risk of developing heat stroke and the rate of accidents increase dramatically. Furthermore, dry bulb temperatures should not exceed 37°C in working areas since the human body starts to absorb heat above this temperature.

A question that can be asked is why there was this sudden commotion about the working conditions in South African mines and increased research concerning the effects of temperature on the human body. The answer lies in the availability of ore bodies. A major problem facing the South African mining industry is the depletion of minerals and coal at easily reachable depths. This is forcing mining operations to migrate to greater depths where ample reserves of ore are still available. An example of this is the planned deepening of two Gold Fields mines, namely Kloof and Driefontein, at an estimated cost of R4.7-billion (Olivier, 2006).

Accompanying the migration to greater depths is the problem of the increase in virgin rock temperature (VRT). This is forcing the mining industry to install ventilation and cooling systems to reach their valuable ore reserves. The moment that ventilation is applied a secondary effect of increased depth starts to play a role, namely the auto compression (also sometimes erroneously referred to as the Joule-Thomson effect) of the air as it descends from surface to the working levels. This increases the total effective heat load on the ventilation and cooling system considerably.

For South African mines refrigeration to create an acceptable working environment becomes necessary at depths greater than about 1500m. This refrigeration is mainly distributed through the service water and bulk air coolers. The increased expansion combined with the demand for increased production rates puts excessive strain on the ventilation and cooling system of a mine. To such an extent that mining will eventually reach depths where current methods of ventilation and cooling will no longer be economical and practical to provide an acceptable working environment.

To solve this problem new technologies or alternative cooling methods are required for mining to be profitable at ever increasing depths. Various unconventional cooling system designs are currently being considered by the mining industry. The fundamental ideas behind many of them are not new but the application may not previously have been economical or practical.

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One such concept is called spot cooling, also known as localised air cooling. This entails cooling the air in a specific location where there is a high air temperature problem, rather than trying to cool the whole environment leading up to the problem area. Until now spot cooling was mainly done with so-called chilled water cooling cars (CWC). A significant limitation of these cooling cars is that their effectiveness is directly proportional to the supply temperature of the chilled water.

A possible solution for this problem is the use of a localised refrigeration plant acting as a spot cooler in remote locations. This unit should differ from a standard underground refrigeration plant based on size and capacity. Besides the requirement for such a unit to establishing a suitable working environment for the personnel underground, it must also be energy efficient (EE) to help reduce the electrical consumption of a mining operation and ideally also pose Demand Side Management (DSM) potential for industry.

1.2 Purpose of this study

The purpose of this study is to develop a prototype air cooling unit (ACU) to comply with the criteria listed above. In Chapter 3 the detail design of the ACU will be discussed, but in brief, the unit is based on a vapour compression cycle with the main features being its modularity and mobility. This first prototype is under evaluation on a developing mining level at a gold mine in South Africa.

Looking at the outcome of this study from a mining perspective, there is not only a need for accurate performance data on the ACU but also a thorough evaluation of the techno-economic impact of the ACU if applied on a wide scale for deep mine expansion. This includes the impact on the total cooling capacity of the mine and the capital investment required to do the additional cooling. Another important factor is the effect the ACU has on the amount of return water that must be pumped out of the mine which has a direct impact on the running cost of the mining operation.

Looking at the outcome of this study from an electricity supply side the ACU can possibly also benefit Eskom, the South African electrical supply utility, due to the DSM and EE potential of the unit.

The study will aim to reach the following main goals:

- Develop a spot cooler unit to contribute to solving the temperature problems in remote locations.
- Determine the techno-economic potential of the ACU when applied on a wide scale in a deep mine operation.
- Investigate the DSM and EE potential of the unit from an electricity supply side.

1.3 Method of investigation

To reach the goals set out in paragraph 1.2 the study will consist of the following:

- Based on operational and physical requirements a prototype ACU will be designed and manufactured.
- After completion the ACU will be subjected to laboratory tests to determine the performance of the unit for varying environmental conditions.

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- The ACU will be equipped with measuring equipment and then be installed underground, to determine the in-situ performance of the unit over time.
- Based on the results of both the laboratory tests and underground testing, the ACU will be characterised for inclusion in a simulation routine. The characteristics of two different CWCs will also be included in the simulation. This is necessary to determine whether the ACU holds any benefit for the mining industry, compared to the conventional CWC method.
- A simplified mine system model will be developed to simulate the effect of both the ACU and CWC if implemented in different configurations for potential deep mining. The mine model will be based on information obtained for the case study mine where the prototype is being tested.
- From a mining perspective the running cost of the different configurations needs to be determined. Along with this the capital investment of the different options must be estimated to assess the present value for life cycle cost evaluations.
- The energy efficiency potential of the ACU configurations will be compared to the CWC configurations.
- The DSM potential of the ACU will be determined by investigating the potential peak demand reduction for the different configurations.

1.4 Statement of originality

The original contributions of this study are:

- The study will present a novel and practical ACU design with the distinct advantage of being able to operate efficiently with high inlet water temperatures. This results in the effective utilisation of normal service water as heat sink.
- The technical performance and techno-economic viability of the ACU will be evaluated and quantified in order to provide the mining industry with a practical solution to comply with legislation concerning suitable working environments.
- The study will quantify the contribution that the ACU technology may make to Eskom's DSM and EE initiatives to reduce the country's peak demand and electricity usage.

CHAPTER 2

LITERATURE SURVEY

CHAPTER 2

2. LITERATURE SURVEY

The aim of this chapter is two-fold. Firstly, to expand on the brief discussion of the problem and its setting in Chapter 1. Secondly, to support the statement of originality of this study. The literature surveyed will be divided into the following categories:

- Mining environment.
- Ventilation and cooling.
- Conventional cooling methods.
- Deep mining below 3000m.
- Cost of ventilation and cooling.
- Alternative cooling methods for deep mining.
- Spot cooling concept.
- Demand Side Management and Energy Efficiency.
- Mine simulation methods.

2.1. Mining environment

The working conditions in South African mines came under the spotlight due to mining operations migrating to greater depths where rich untouched ore reserves are located. The most important observation to emerge from the report of the Commission of Inquiry into Safety and Health in the Mining Industry (also known as the Leon Report) is that a safe working environment will remain an unreachable goal until the fundamentals of occupational health are more widely understood (Kielblock, 1998). According to the report, health at work is related to the environmental conditions in which work is done, and any separation of health effects from environmental conditions is inappropriate. The Leon Report released in the 1990's wasn't the first time that concerns were noted regarding the working conditions in the mining industry.

As early as the 1930's the combined effect of high temperatures and humidity was regarded as a major cause of reduced productivity but more importantly a health hazard (Sheer *et al.* 1984). Over the years research has shown that the risk of developing heat stroke and the rate of accidents increase dramatically at wet bulb temperatures exceeding 27.4°C in stopes and development ends (Leveritt, 1998; Marx *et al.*, 1998).

As stated in Chapter 1, accompanying the expansion to greater depths is the problem of the increase in VRT. The geothermal gradient is region dependent and varies between approximately 10°C/km and 22°C/km in South Africa (Marx, 1998). If for example the VRT is 50°C at a depth of 3000m, the air temperature inside any mine tunnel at this depth would also be 50°C, if not ventilated. According to Ramsden (1990), in 1961 the maximum wet bulb temperature for acceptable working conditions was considered to be between 30.6°C and 33.9°C. In 1990 acceptable temperatures were 3°C to 4°C

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lower. A major reason for the change in acceptable conditions was the realisation that the workforce is the mining industry's most valuable asset. For the mining industry to retain its staff, the continual improvement of working conditions is necessary.

Currently mines are prohibited by law from exposing workers to underground temperatures above $32.5^{\circ}\text{C}_{\text{wb}}/37^{\circ}\text{C}_{\text{db}}$ (South Africa, 2002). The only way the industry can comply with legislation is by installing ventilation and cooling systems. The higher the VRT, the greater the total heat load that the mine ventilation and cooling system must contend with, resulting in increased cooling capacities and costs. It is difficult for mines to adapt because historically rock breaking and hoisting criteria have dominated the mine design process with cooling and ventilation needs of lesser importance (Bluhm *et al.*, 2000).

Accompanying ventilation is a secondary effect of increased depth on the working environment. This is known as auto compression of the air as it descends from surface to the working levels. The reduction of gravitational potential energy of the downcast air is offset by an equivalent increase in the enthalpy, at a rate of roughly 9.8kJ/kg/km (Marx, 1998). This leads to a wet bulb temperature increase of about 4°C/km of depth. The total effective heat load on the ventilation and cooling system is increased considerably, although the relative contribution of the auto compression actually decreases with depth as the VRT effect becomes more dominant.

The problems explained above are not unique to South African mines. In 1985 Moser stated that there has been a growing demand for refrigeration in German mines mainly due to two reasons:

- As seams become exhausted the mining operation expands to greater depths resulting in higher temperatures.
- Stricter German legislation prohibits work in areas where the temperature is above 32°C .

The problem of increased VRT is even worse in Japan. Hiramatsu *et al.* (1980) described a scenario at the Toyoha mine. Due to VRT in the region of 85°C at levels 300m below surface it was extremely difficult to expand the mining operation. The only way to expand operations down to 450m below surface was by spraying chilled water on the rock surfaces and in the air. The problem with this technique was that only 50% of the cooling power of the chilled water was used to cool the air. A need was identified to improve the ventilation for more effective heat removal otherwise the mine would not be able to expand any further.

2.2. Ventilation and cooling

As stated earlier, the only way the mining industry could legally reach their ore reserves is by employing ventilation and cooling systems. This is easier said than done. Almost all of South Africa's gold mines fall in the category of VRT exceeding 35°C on some underground level. Above this temperature thermal control becomes a dominant factor in the planning of a mine ventilation system (Whillier, 1980). Sheer *et al.* (1986) stated that on average about 10 tons of air is circulated through a mine for each ton of rock that is broken. Employing air to remove heat from a deep mine is greatly hampered by the increase in temperature due to auto compression as the air goes down the mine.

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According to Whillier (1980) it has been shown that if the wet bulb temperature of the supply air to a level exceeds approx. 23 °C, refrigeration of the air on that level will be necessary to keep the wet bulb temperature in the work places below 28 °C. This implies that for South African mines refrigeration becomes necessary at depths greater than about 1500m.

In addition to air, water is also supplied to underground workings mainly for air cooling and hydro powered equipment. The temperature at which the water reaches the workings can dictate how hot the local environment becomes. Measurements in deep mines confirmed that service water, if not refrigerated, reaches the stope face at temperatures between 30 °C and 35 °C (Whillier, 1980). Thus there is a need to distribute as much refrigeration as possible through the service water and bulk air coolers.

Funnell *et al.* (2001) stated that when looking at the positional efficiency of cooling distribution with chilled service water at the workings the following apply:

- From a ventilation network perspective the positional efficiency could not be better since the cooling is in the actual workings.
- From a chilled water reticulation perspective the positional efficiency could not be worse since it is as far as it could be from the refrigeration source.

Ventilation and cooling requirements are often not anticipated or are perhaps overlooked at the design and planning stage of a shaft. Hence, cooling strategies have to be devised and refrigeration installations have to be engineered around the available shaft facilities. The engineering of installations under these circumstances is a costly exercise and the consequences can be quite significant. For example, it may rule out the possibility of employing an energy recovery device and force a mine to install a refrigeration unit underground (Van der Walt and De Kock, 1984).

In summary, it is clear that mines require refrigeration to create an acceptable working environment. This environment must comply with the legislative and health requirements of the people working underground. Ventilation and cooling installations were never developed to create comfortable conditions as this would be excessively expensive. The increased expansion combined with the demand for increased production rates puts excessive strain on the ventilation and cooling system of a mine.

2.3. Conventional cooling methods

Hegerman (1997) stated that the first mines were shallow and acceptable underground temperatures could be maintained by circulating air from the surface through the mine. The problem is that with increasing mining depth both the VRT and heat load increases but the cooling effect of air from surface decreases. This resulted in the introduction of refrigeration plants for cooling service water and ventilation air on surface before it is dispatched underground.

According to Ramsden (1990) in the 1970's mines developed a three phase policy for distributing cooling from refrigeration plants. In the first phase all service water that is used in normal mining

operations is cooled. This has the effect of making the service water a heat sink rather than a heat source. In the second phase the ventilation air is cooled in large bulk air coolers either underground or more commonly on surface. This phase ensures that the ventilation air has the maximum capacity to remove heat. Finally, in the third phase the ventilation air is cooled in much smaller quantities closer to the workings.

A simple description of a typical South African mine cooling system is given by Hegerman (1997). Heat in the workings is absorbed by cold water in heat exchangers or cold machine water flowing in the stopes or development ends. The warm water is returned by pumping to the cooling plant where the heat is rejected from the water to the atmosphere. The cold water or ice is then returned underground either directly or via energy recovery equipment for the cycle to start all over again.

Since about 1977 the majority of new refrigeration machines were installed on surface, with the result that by 1986 only half of the total industry's refrigeration capacity was located underground (Sheer *et al.*, 1986). Some of the surface installations are partly used to pre-cool the ventilation air on surface. This is done in an attempt to achieve satisfactory air temperatures throughout the system of underground tunnels and shafts leading to the stopes. The remainder of the refrigeration capacity is used for chilling water which is piped underground, to be used in air coolers and as service water for equipment.

After absorbing heat all of this water must be pumped back to surface and the high operating and capital costs of doing this constitute the disadvantage of surface installations. There is also a limit on the amount of cold water that can be circulated down the mine and back to surface. Several new plants installed in the late seventies have incorporated plate-type evaporators with the water on the outside of the plates being permitted to freeze. This enabled the water to be cooled to below 1 °C with a resulting increase in cooling effect in the mine of 6 to 12% for a given water circulation rate and given pump cost (Whillier, 1980).

According to Sheer *et al.* (1984) the advantage of using surface refrigeration installations is due to an increased efficiency compared to underground installations. The disadvantages of a surface plant include:

- Long insulated chilled water pipelines are necessary.
- High cost incurred from pumping water back to surface.
- The temperature of the down-going water rises due to the Joule-Thomson effect, unless mechanical work is taken from the flowing water.

Instead of refrigeration plants being installed on surface, they can also be installed underground. Although there are usually no serious problems with a heat rejection system located on surface, the same cannot be said of an underground heat rejection system or cooling tower. The latter invariably suffers from a shortage of air and thus condensing temperatures are much higher than on the surface, resulting in a considerable increase in the power consumption of the refrigeration machines for a given cooling capacity. This limitation is due to the temperatures of the up cast ventilation air into which the

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condenser heat is rejected. The high air temperatures result in increased volume flow rates required to satisfy the condenser capacity. There is however a limitation on the available flow rates mainly due to the fans that are available.

Van der Walt and De Kock (1984) stated that the water in an underground heat rejection system is usually contaminated with dust and blasting fumes, hence fouling, scaling and corrosion of pipes and condenser tubes can be a major problem that requires constant attention. It furthermore aggravates the problems associated with performance and operation of refrigeration machines underground.

In deeper mines the ventilation air is therefore cooled in bulk either on surface or underground on the main intake levels. In addition, when air passing through the workings becomes unacceptably hot, it must be re-cooled if it is to continue through to other work places. This re-cooling is done locally using smaller cooling coils. The hot air is blown across the heat exchanger while cold water is circulated through the tubes. In 1984 34% of all mine refrigeration was distributed through bulk cooling and 41% through cooling coils (Sheer *et al.* 1986). In 1990 31% of all cooling was distributed through chilled service water, 44% through large bulk air coolers and 25% through small air coolers positioned close to the face (Ramsden, 1990). According to Sheer *et al.* (1986) studies have shown that the performance of a cooling coil type heat exchanger can easily be reduced by as much as fifty percent due to a combination of moderate fouling and poor air distribution.

In 2001 Funnell *et al.* stated that for previous decades the design philosophy was that chilled service water should be the first stage in introducing cooling into a mining operation, followed by more formal air coolers. However, the efficiency of chilled service water as a carrier of cooling is greatly affected by thermal losses. The desired cooling effect can only be obtained if the water reaching the workings is cold. A factor influencing the implementation of chilled service water is that mines are becoming deeper and extend further resulting in only marginally improved conditions even with an efficient system due to the dominating effect of thermal losses.

According to Funnell *et al.* (2001) the minimum cooling that should be done in any mine is to cool water on surface in a cooling tower but the greatest cooling effect would be achieved by refrigerating the service water and supplying it to the workings at the coldest temperature possible. It is also stated that for deep mines where pumping costs are dominant the coldest possible water must be provided to the workings.

Ramsden (1990) felt that "... there is a basic need to review the cooling systems of the 1970's and early 1980's so as to develop more cost effective and appropriate cooling systems which will meet the social norms and legislation of the 21st century." The viability of future mining prospects may ultimately be decided by the capability of providing a safe and acceptable underground working environment in a cost effective manner (Marx, 1998).

2.4. Deep mining below 3000m

As stated in Chapter 1 South African gold-miner Gold Fields announced that they are planning to deepen two mining operations to become the world's deepest and second-deepest mines (Olivier, 2006). To do this Gold Fields will be looking to use new technologies in the endeavour.

This is not the first time that the use of new technologies was mentioned. As early as 1998 Marx *et al.* stated that if the current trend of mine expansion to greater depths continues, gold mining will eventually reach depths where current methods of ventilation and cooling will no longer be economical and practical to provide an acceptable working environment.

"How will cooling be distributed in deep mines in the 21st century with mining operations taking place at a mean rock breaking depth of 3500m, producing 200 ktons/month and requiring 100MW of cooling?" (Ramsden, 1990). To try and answer this question Ramsden sketched the following scenario: If the service water consumption is one ton of water per ton of rock mined then only about 5MW can be distributed through the service water thus 5% of the total cooling requirement. Since it is a deep mine, auto compression of the ventilation air will result in it being a heat source rather than a heat sink resulting in minimising the circulation of ventilation air. If it is assumed that all the ventilation air is cooled once in large bulk air coolers then the quantity of cooling distributed in this manner would be limited to a maximum of about 40MW. The remaining 55MW, or over half of the cooling required would have to be distributed through smaller air coolers. This clearly illustrates the future importance of smaller air coolers.

There have been developments in reducing heat loads for deep mines, including the backfilling of worked out areas, recirculation of the ventilation air, and insulation of intake haulages. Despite these developments, it is accepted that the deep mines of the 21st century will require vast quantities of refrigeration and ventilation (Bluhm *et al.*, 2000).

Ramsden (1990) felt that the major problem associated with large cooling installations of 100MW or more, is the mere size and practical problems associated with their operation. According to Ramsden it is clear that new technologies are required if mining is to continue to be profitable at ever increasing depths. The problem is that these new technologies will probably require a higher skilled workforce which will in turn demand agreeable working conditions as they will most likely be recruited from outside the mining industry. Eight years later Marx (1998) emphasised that the viability of ultra deep mining will be decided by the cost and availability of technical solutions for the provision and maintenance of a suitable underground working environment.

Mark Butterworth, CSIR: Miningtek research area manager, made a similar statement that mining at depths of 5000m will be feasible, provided that some technologies and equipment are developed (Creamer Media, 2001). Butterworth goes a step further by pointing out that another vital aspect needing recognition by the mining industry is the importance of planning, implementing and operating ventilation and cooling systems. The high capital cost of cooling equipment implies that an optimised phase-in strategy is as important as correctly sized equipment (Bluhm *et al.*, 2000).

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As shown in Figure 2.1 the relative increase in the cooling requirements exceeds the relative increase in depth, indicating that beyond a certain depth mining becomes uneconomical. The gradient of increase in cost to provide this cooling can be even steeper due to increasingly limited heat rejection capacity with depth.

According to Marx (1998) technologies to reduce the environmental control difficulties of mining at greater depth include:

- Simulation software which enables appropriate underground environmental assessment and designs, and considers new mining and environmental technologies.
- Novel and cost effective cooling technologies where advances in mechanical, electrical and hardware technologies provide less expensive cooling.
- An optimal balance is vital between refrigeration plants with a low coefficient of performance (COP) but good positional efficiency and plants with a high COP but poor positional efficiency. The ideal refrigeration plant should have both a high COP and good positional efficiency.
- Flexible ventilation and cooling systems is vital to the success of deep mining. Productivity can be enhanced and costs reduced through the provision of ventilation and cooling on an as-required basis.

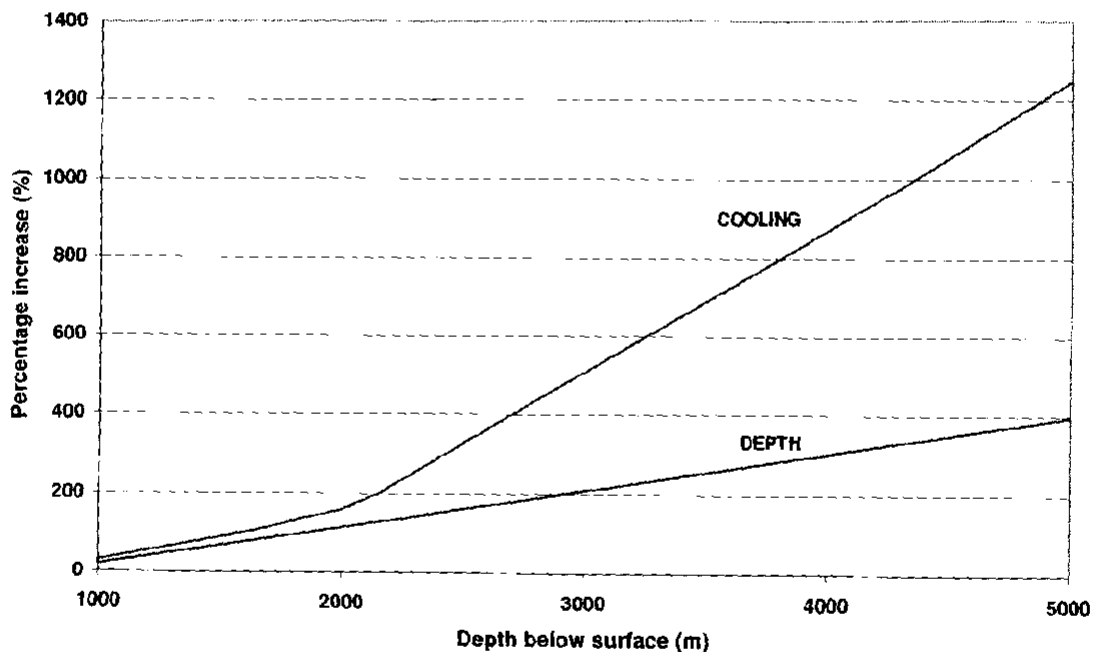


Figure 2.1: Cooling requirements with depth (Marx, 1998).

As is evident from the above references, there has been a lot of talk over the last few decades concerning solutions for the problems faced with deep mining. However, by 2006 there has been little progress in implementing practical solutions. A major factor that hampers this process is the costs associated with ventilation and cooling.

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2.5. Cost of ventilation and cooling

As early as 1984 Sheer *et al.* stated that for a deep mine the cost of environmental conditioning may contribute a quarter of the total mine working cost, rising to a third on very deep mines. Due to the extent of the difficulties caused by the harsh underground environment, the future profitability of the industry is at stake. Ramsden (1990) felt that with greater depths it is inevitable that mine cooling costs will increase at a startling rate outstripping general mining costs and the inflation rate.

As stated earlier, deep mines are operating at depths greater than 3000m resulting in the heat load per ton of rock mined being significantly higher than shallower mines. This leads to a greater installed cooling capacity requirement resulting in increased power consumption to run the cooling plant. Combined with this, increased pumping power required at greater depth significantly influence the power consumption. Therefore, both capital and operating costs increase significantly with depth influencing the total capital and working costs of ultra deep mines.

According to Hegerman (1997) air from surface bulk air cooling towers has little or no cooling power at a depth of 3400m. Therefore, increasing airflow through the mine will result in very little improvement in cooling. However, excessive power costs of main fans and additional refrigeration plants would be incurred.

To further emphasis this point Sheer *et al.* (1986) stated that the overall COP of a complete cooling system located on surface, including net water pumping power requirements, could decrease by 25% in going from 3000m to 4000m. This implies that the costs associated with this form of cooling system could well be unacceptably high for the next generation of very deep mines. The failure to introduce new cooling system designs will lead to the industry being forced to revert back to underground refrigeration machines on deep new shafts to reduce pumping costs.

Due to the increasing cost with depth Sheer *et al.* (2001) felt it a matter of high priority to develop a new generation of more cost-effective mine cooling systems, aiming to:

- Minimise the quantities of air and water circulated between surface and the ever more remote underground workings.
- Obtain better control over how the cooling is applied, resulting in better positional efficiency and reduced losses.

Sheer *et al.* (2001) determined the full cost implications of using several alternative types of cooling systems for mining depths down to 5000m below surface. The study showed that the most economical form of cooling system for great depths is a hybrid system consisting of a combination of underground and surface refrigeration machines. Because the maximum refrigeration capacity installed underground would be limited by the combined heat rejection capacities of the air and water streams returning to surface, additional refrigeration would generally need to be installed on surface. The comparative evaluations confirmed that for depths below about 3000m ice systems would be the least expensive option for the surface plant in spite of the high costs of making ice.

Bluhm *et al.* (2000) looked at the duties of ventilation and cooling on the Witwatersrand. The following overall air cooling duties were found on average:

- 135kW of air cooling per kt/month at mean depths of 3000m.
- 230kW of air cooling per kt/month at mean depths of 3500m.

The increase in air cooling requirements is due to the loss of efficiency with depth. The overall refrigeration requirements of water were found on average to be:

- 370kW of refrigeration per kt/month at mean depths of 3000m.
- 570kW of refrigeration per kt/month at mean depths of 3500m.

If a comparison is made for the same ore production, no refrigeration would be required at a depth of about 1800m, showing that the change in requirements between 3000m and 3500m of about 55% is very important. The anticipated change from 3500m to 4000m is predicted to be even greater to a point where the cost of refrigeration at such extreme depths will dictate the viability of mining operations. While short term needs are important to any mine, it will be necessary for management to identify and take note of long term requirements.

Marx *et al.* (1998) conducted a theoretical investigation showing that acceptable conditions can be achieved at an economically viable cost. However, realising greater efficiencies will require new developments and/or suitable modification of current cooling technologies. This can only be reached through a focused research and developmental effort concentrating on alternative cooling methods.

2.6. Alternative cooling methods for deep mining

The most important aspect of conventional cooling equipment is the fact that it is proven technology. Suppliers have extensive experience in the mining industry and are therefore in a position to select the best equipment for a particular application. The problem with this is that in a competitive market the suppliers of other types of refrigeration machines will find it difficult to penetrate (Marx *et al.*, 1998). Although it may be technically possible to mine at greater depth using existing technology, the drive to develop and implement new or improved technologies is spurred on by various factors including economic pressure and the importance of the health of the workforce.

Various alternative cooling system designs are currently under consideration by the gold mining industry for implementation. It must be noted that many of the fundamental ideas behind these systems are not novel but the application may have been uneconomical or impractical in the past (Sheer *et al.* 2001).

Ramsden (1990) believed cooling systems in the 21st century will use fluids which will have a much higher cooling capacity so that the size and cost of distributing the cooling can be significantly reduced. Ramsden predicted that many systems will use ice, because it has all the properties of water but in addition has a cooling capacity of four to five times greater.

Before examining alternative cooling systems, Bluhm *et al.* (2000) feels that any design procedure must select critical stages in the life of a mine for detailed analysis. This selection should be based on

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criteria including production rates, mining method, depth of mining, intake distances, number of stoping zones and degree of scatter. This is an iterative process resulting in the determination of heat loads and ventilation quantities, the sizing and positioning of air coolers and the selection of optimum refrigeration and cooling distribution systems.

A study of many different scenarios for extreme depths indicated that the optimum cooling strategy requires the use of primary bulk air cooling, secondary air cooling and tertiary air cooling, together with the use of chilled service water for distributing cooling directly to the working zones (Bluhm *et al.*, 2000). The size and position of air coolers are critical to produce a fully cost effective system. A proposed system consists of the following:

- Large primary bulk air coolers should be installed on transfer levels between the primary and secondary shafts, consisting of high efficiency spray chambers. Return water will be pumped to surface from this horizon. Below these depths, primary bulk air coolers and chilled service water are inadequate resulting in the need for secondary bulk air coolers and tertiary air coolers to cater for the heat load needs.
- The secondary bulk air coolers should be spray chambers permanently installed off main cross-cuts at locations where conditions are unacceptably hot. The proposed tactic is to maximise the duty of these installations and cool the air to as low as $18^{\circ}\text{C}_{\text{wb}}$.
- The tertiary air coolers should be transportable and strategically located to control working zone reject temperatures.

Thorp and Bluhm (1986) implemented small water-chilling units in conjunction with specially designed air coolers to solve heat problems encountered in a working area. The chilling units each had a 350kW cooling capacity. Each unit supplied its own air cooler with chilled water via a reticulation system. The air coolers were direct-contact units spraying the chilled water into the warm air stream, and collecting the water in a sump below the sprays. The water was pumped from the sump to a tank and then gravity fed to the evaporator. This resulted in the average face wet bulb temperature dropping from 32.3°C to 28.8°C .

Walters (1986) described the use of a relatively simple method for localised cooling of a development end. The unit consisted of a full-cone spray installed in the centre of a duct, spraying chilled water directly into the ventilation air stream. Cooling duties between 20kW and 50kW were measured, but the cooling was very depended on the water flow rate and temperature conditions. Marx *et al.* (1998) felt that the effect of auto-compression could be dealt with by coolers at mid-shaft levels and near the sub-shaft on working levels.

Del Castillo (1988) proposed an air cycle refrigeration system for use in deep mines to provide underground refrigeration without the use of water thus reducing pumping costs. A portion of the ventilation air is compressed on surface, then cooled and dried before being piped underground. The compressed air is expanded underground in a turbine driving a generator. The sub-zero exhaust air from the turbine is then mixed with the ventilation air to create a cool mixture. From the results of the

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investigation the author concluded that an air cycle system only becomes cost effective at mining depths greater than 4000m.

Hegerman (1997) did a comparison between cooling systems to find the optimum system for Vaal Reefs No. 11 Shaft. The system selected took chilled water directly into cooling coils at the stopes after which it passes through spray chambers located in fresh intake airways. This allowed for high positional efficiency and increased heat absorption by the water before it left the workings. The water was returned to surface using water transformers, which were less than half the capital cost of three chamber pipe feeder systems and more efficient. On surface heat was rejected in pre-cooling towers and further cooled down to 0.5°C in ammonia water chillers. The reason for the ammonia water chillers was the cost compared to conventional R134a chillers. Three stages of energy recovery hydro hoisting took place after which the water reached the workings at 4.5°C. This application may not be viable for new mine shafts due to the law prohibiting the use of ammonia as working fluid. The concept could still work by replacing the ammonia with R134a but the cost effectiveness of the system will be affected.

Biffi and Bluhm (2001) investigated the heat loads and cooling requirements to enable mining operations to take place at depths up to 5000m. An important observation from the study was that for ultra-deep mines the heat energy absorbed in stopes is secondary in relation to the heat load of the intake tunnels. A suggested optimum approach is to cool the ventilation air to relatively low temperatures in coolers at stope entrances.

Den Boef (2001) simulated an underground mobile refrigeration plant while Stanton (2003) tested the unit in remote areas. The unit consisted of an off-the-shelf water-to-water chiller with a 500kW cooling capacity. The hot condenser water was circulated through a closed-loop to water-to-air coil cars situated in the return airway (RAW). At that point, return air was blown over the coils to extract heat from the 50°C condenser water before the water was circulated back to the chiller unit. On the other side of the chiller unit the chilled evaporator water was pumped through a closed-loop to CWCs. The hot supply air is blown over the coils to be cooled by the chilled water circulating through the tubes and back to the chiller unit. The main shortcomings of the unit include:

- The chiller plant wasn't really as mobile as the author originally claimed. Moving the unit would be a labour intensive operation.
- The cooling COP of the unit was relative low, between 2.5 and 3.3.
- Due to very narrow fin spacing on the water-to-air coils, the coils blocked quickly, resulting in a decrease in performance. To overcome this problem, solenoid controlled sprayers were installed to keep the outside surface of the coils wet. This had limited success as the sprayers blocked due to the quality of the water and the dust in the air. This resulted in more complicated maintenance.
- Due to the vast distances between the chiller unit and the condenser-water cars in the RAW, airlocks was a major problem mainly due to leaks at joints at the top of the raise. Air bleed valves were introduced to reduce the effect. A major problem with the piping in the raise is that if there was a problem no person would go to investigate due to the difficult climb up the raise.

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Kebonte and Biffi (2001) investigated the feasibility of using geothermal heat sinks at depth to maximise the positional efficiency of the system. The condenser heat is rejected to the rock thus avoiding the cost of surface heat rejection. Geothermal or ground-source heat pumps consist of a reversible vapour compression cycle linked to groundwater as source or sink. For their model Kebonte and Biffi assumed the ground piping would be buried in old excavations to eliminate the cost of drilling or trenching. Crushed rock would be used as a filling material to keep the thermal conductivity near to that of the rock. Calculations performed on an underground refrigeration plant model indicated that for a condenser duty of 4.7MW to be rejected to the rock, a ground heat exchanger of 209km long would be required if working with a 297mm inner diameter steel pipe. It was concluded that to achieve a given COP significant length of piping is required resulting in an unrealistic system due to pump sizes, cost of piping and space limitations.

Surface bulk air cooling systems are typically designed to cool ambient air to about 10°C and the effectiveness of this cold air is often further reduced when mixed with warm ambient air in the shaft before being downcast. It is generally considered that for comfort in man-conveying shafts the temperature should not be less than about 10°C dry bulb. In the absence of this constraint, the effectiveness of mine cooling systems using surface bulk air cooling can be enhanced by chilling the air to ultra cold temperatures of typically 2°C (Wilson *et al.*, 2003). As mining gets deeper it is inevitable that ultimately cooling can no longer be achieved from surface. There is a breakeven depth that can be identified and at that stage underground cooling should be considered. However the introduction of underground cooling is accompanied by an increase in both complexity and overall cooling system costs. Figure 2.2 illustrates an example of how the utilisation of ultra cold air may move the critical horizon for the introduction of underground cooling from a depth of 1600m to 1900m.

The thermodynamic penalties related to operating a colder refrigeration plant are mainly an increase in compressor power. Figure 2.2 illustrates that the ultra cold downcast air may move the underground cooling horizon deeper and this option can remain attractive down to depths of about 2000m. Below this point underground cooling will be inevitable. For a given underground cooling effect and depth, as the downcast air is made colder the result is that less overall ventilation is required resulting in a reduction in main fan power. However in order to make the downcast air colder, refrigeration running cost increases and refrigeration equipment becomes more expensive.

An area in the mining industry that needs a lot of attention is the cooling of remote underground locations where the effect of heat on the workforce is more severe than in general underground areas. In the next section a technique for cooling these areas will be discussed.

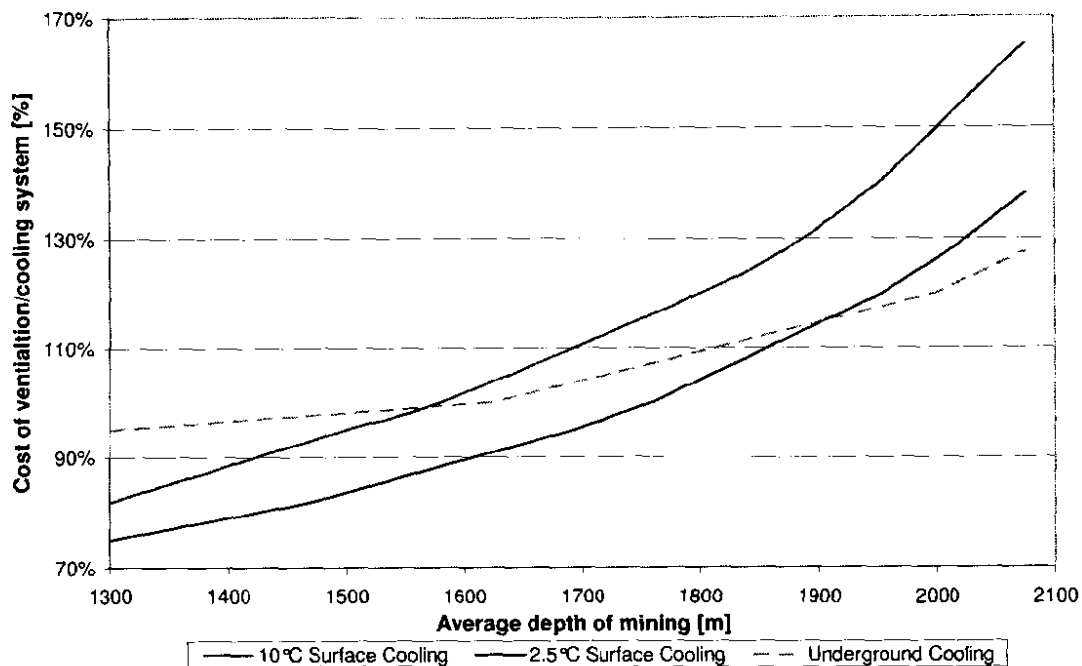


Figure 2.2: Cost of ventilation and cooling systems (Wilson *et al.*, 2003).

2.7. Spot cooling concept

Ramsden (1990) stated that during the 1980s underground air coolers have fallen from favour because of high maintenance requirements, poor performance due to flow control problems, and generally high operating costs. Ramsden felt strongly that to ensure the availability of these coolers for the new generation of deep mines in the 21st century, development must start well in advance. Due to mechanisation and advances in mining methods, size and weight restraints may not be such limiting factors for new mines.

In an article regarding an underground chilled water refrigeration plant being installed at Driefontein mine, Cromberge (2004) quoted John Kidd, York International (SA) engineering manager, saying "Bulk air-coolers placed on the surface of a deep mine will therefore be insufficient to cool the whole mine." This is mainly due to the auto-compression effect with increased depth. John Kidd also noted that spot cooling near the working area is a potential answer to much of the problems.

Sheer *et al.* (1984) felt it impractical to consider providing individual micro-climate systems for the hundreds of thousands of men working underground and believed there is little alternative but to condition all working areas. The concept of spot cooling can be seen as an inter-level cooling method between the impractical micro-climate systems and cooling the whole area.

Spot cooling, also known as localised air cooling, entails cooling the air in a specific location where there is a high air temperature problem, rather than trying to cool the whole environment leading up to

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the problem area. These installations usually consist of a smaller capacity cooling unit in the problem area. Figure 2.3 illustrates the installation of a spot cooling unit in a remote location to supply cold air to a problem area, like the working front. The unit extracts energy from the hot air that is blown over a heat exchanger coil, transferring the energy from the air to chilled water flowing inside the coil.

Until now spot cooling was mainly done with so-called chilled water cooling cars (CWCs) of which an example is shown in

Figure 2.4. A significant limitation of these CWCs is that their effectiveness is directly proportional to the supply temperature of the chilled water. Due to the vast distances the chilled service water travels from the surface cooling plant to remote underground areas, the water arrives at a temperature insufficient for cooling, resulting in a drastically reduced CWC performance. Sheer *et al.* (1986) stated that studies have shown that the performance of a cooling coil type heat exchanger easily reduces by fifty percent due to a combination of moderate fouling and poor air distribution.

These units are also only effective if a mine is equipped with a chilled service water system. For mines supplying regular un-cooled service water to equipment there is no advantage gained by installing these units. As stated in Chapter 1 a possible solution for this problem is the use of a localised refrigeration plant acting as a spot cooler in remote locations. However, this unit should differ from a conventional underground refrigeration plant based on size and capacity. Energy efficiency is a further requirement for such a localised refrigeration plant to help reduce the electrical consumption of a mining operation.

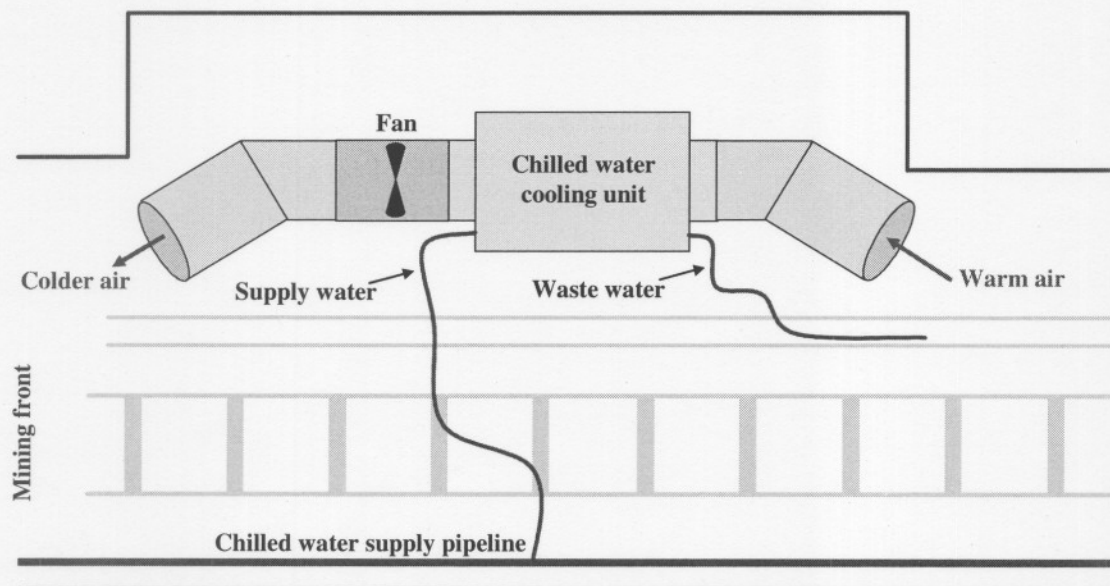


Figure 2.3: Spot cooling concept.

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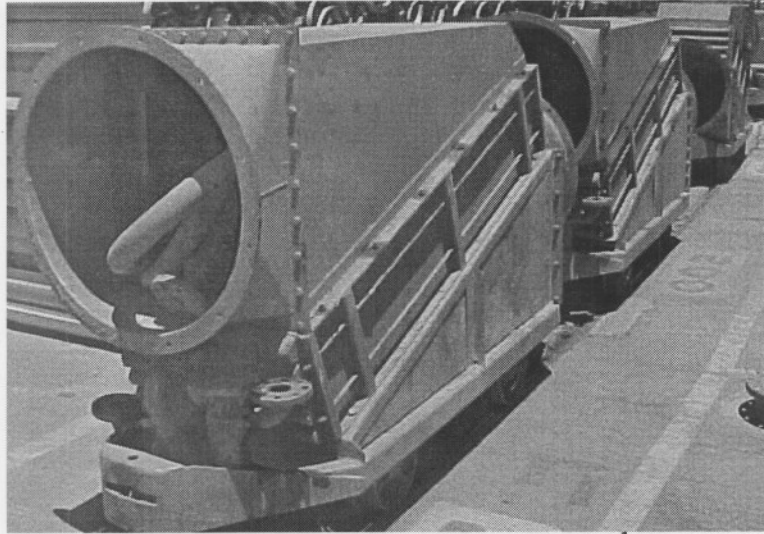


Figure 2.4: Chilled water cooling car¹.

During the literature survey it became evident that the concept of a portable refrigeration unit was firstly developed during the 1960's. Chapple and Siegel (1967) filed a patent for a portable mine cooling unit. The unit is divided into two halves. In the top half the air passageway is located with all the refrigeration components located in the bottom half to improve the stability of the unit. The unit has a built-in high velocity fan with the shortcoming that the blades can be damaged by debris. The designers placed the evaporator coil before the fan to protect it from dust and debris. To stop any debris from continuing on to the fan the fins on the coil are spaced close together. The problem with this is that due to the narrow fin spacing the coil face area will block easily therefore requiring regular cleaning. The layout of the unit is such that cleaning would be a complicated job. The coil can only be accessed if the side panelling of the unit is removed. The design on the shell-and-tube condenser consists of refrigerant entering the water cooled condenser at the top and passing through baffle chambers to a lower collecting sump at the bottom of the condenser, from where it flows to the expansion valve. No evidence could be found in the open literature that the unit patented by Chapple and Siegel was ever implemented in the mining industry.

In 1975 McDonald and Caffall also filed a patent for a portable air conditioner unit for use in mines and other similar restricted areas. The unit is completely self-contained and the designers claimed it to be easily movable in working areas and passageways. The drawbacks of the design include:

- There is a filter installed on the air side for dust removal which has practical implications. Filters require regular cleaning which underground personnel rarely will do, resulting in them usually removing the filter the first time it clogs up and never replace it again. Similar to the unit of Chapple and Siegel (1967) the side panelling must be removed to clean the coil.

¹ <http://www.manos.co.za>

- The unit has a built-in fan which is operated with a belt drive between the air passage way and an enclosure for all electrical components. The interface between the two chambers has seals where the belt moves through. These seals may become a problem letting dust and moist air through due to wear and tear. Furthermore, the mining industry would rather install its own type of fan externally to the unit for maintenance purposes and standardised equipment used.
- The refrigeration components excluding the evaporator are located in an air tight chamber situated in the top half of the unit, with the air passageway located in the bottom half. This can make the unit top heavy during transportation and rigging increasing the chances for an accident resulting in damage to the equipment.

As with the patent of Chapple and Siegel (1967) no evidence could be found if the technology was ever implemented.

2.8. Demand Side Management and Energy Efficiency

The term Demand Side Management (DSM) originated in the United States during the early 1980s. The term DSM was later adopted in the United Kingdom, Europe and Australia. DSM is the process whereby an electricity supplier influences the way electricity is used by a customer. DSM means the planning, implementation, and monitoring of end-user activities to encourage consumers to modify patterns of electricity usage, especially the time of use and level of electricity demand (Eskom, 2006a).

Electricity demand is the amount of electricity required by all electrical equipment to operate simultaneously in a building, an area or a city. The prime objective of DSM is to manage the demand effectively, resulting in reduced amounts of electricity consumption during peak times. If electricity is managed correctly, the demand will be more consistent helping electricity suppliers to meet the requirements of all of its consumers. If implemented correctly, the benefit of DSM is efficient use of electricity without influencing the production and satisfaction levels of a customer. This will result in significant cost savings for both the provider and consumer.

DSM has the following benefits if implemented correctly (Eskom, 2006a):

- Reducing demand during peak times.
- Delaying the requirement for infrastructure capital investment.
- Keep the cost of electricity down.
- A range of electricity efficient options available for customers to implement and benefit financially.
- Supporting the macro-economic development of economies through improved productivity.

In terms of DSM implementation in South Africa, the concept is still relatively new with the first DSM plan only produced in 1994. For the last decade Eskom has funded many DSM initiatives and promoted the efficient use of electricity.

Along with DSM, energy efficiency (EE) is also vital to limit the impact of energy use on the environment. This includes reducing greenhouse gas emissions, supporting sustainable energy resources usage and reducing energy costs.

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EE has the following benefits if implemented correctly (Eskom, 2006a):

- Saving electricity means saving money.
- Preservation of non-renewable resources.
- Conserving the environment by reducing emissions and water consumption at power stations.

In South Africa most people use electricity during peak times resulting in excessive strain on the country's electricity resources. To cope with this Eskom needs to generate more electricity for these short peak periods. With the current supply capacity stretched to its limits, Eskom is forced to build more power stations to cope with the peak periods after which the stations will operate below capacity for large parts of the day. The problem is that consumers will inevitably have to bear the cost. By using electricity efficiently consumers limit the need for large new power stations. Due to the fast expansion of the electricity supply network in South Africa to new users, the installation of more power plants won't be averted but it gives the chance for new power generation technologies, like the Pebble Bed Modular Reactor (PBMR), to be implemented instead of conventional coal fired stations.

The main result is that the environment will benefit from the efficient use of electricity. For every kilowatt-hour of electricity saved about one less kilogram of carbon dioxide generated by conventional power station is released into the atmosphere. The South African Government recognised the importance of electricity efficiency by promoting the efficient use of electricity in all sectors. Due to the great importance of EE to everyone in South Africa, Eskom committed itself to the efficient use of electricity by giving DSM high priority.

In the last few years a large number of DSM projects were launched in the mining industry to try and reduce the daily peak demand between 18:00 – 20:00. Energy Service Companies (ESCOs), along with their respective sub-contractors, successfully implemented projects focusing on:

- Reducing the peak demand contribution of water heating installations at hostels using various innovative technologies such as the in-line heating methodology.
- Pump scheduling on the pumping system used to return water from underground dams to surface. This is done using advanced optimisation software.

In the mining industry the importance of energy savings has been evident for the last few decades.

According to Van der Walt and de Kock (1984) the prime objective of engineering developments in the late 1970's was to reduce:

- The amount of water to be circulated between an underground reservoir and various users thus reducing the energy consumption associated with pumping.
- The amount of water to be circulated between a refrigeration installation and the underground reservoir. On a deep mine the energy consumption associated with pumping water to the surface is typically between two and three times the energy consumption of the associated surface installation.

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- The overall energy consumption associated with cooling. The main users include the high pressure water pumps for returning water to surface, the refrigeration installations and the fans and pumps associated with stope air coolers.

From an energy savings viewpoint the main advantage of cooling air on surface is the saving in electrical power associated with pumping the used cooling water back to surface. The main concern of this approach is that significant costs and losses associated with a malfunction or inadequate design exists due to:

- Loss of production when the wet bulb temperature rises above critical limits and production has to stop.
- Loss in productivity when the wet bulb temperature rises above 28 °C. As stated earlier, the thermal environment has a direct effect on the physical and mental performance of workers.

In 1994 Delpont and Lane felt it a matter of high priority to implement a DSM program targeted at deep level mining because it offers great opportunities to change the demand profile of South Africa. Electricity satisfies more than 95% of the mining industry's energy requirements. Delpont and Lane made the following statement which is very much in line with what is currently happening in the mining industry: "The utility companies have to realise that the mining industry probably has the biggest load management potential in South Africa and that expertise already exists. Co-operation and collaboration between mine personnel, utility planners and researchers is of the utmost importance for the development of realistic load management techniques for the mining sector."

Mills and Rosenfeld (1996) stated that many non-energy benefits may play a critical role in consumer perceptions of energy-efficiency technologies. The authors also felt that greater recognition of non-energy benefits and efforts to assess them properly will help the marketing of energy-efficiency technologies.

Funnell *et al.* (2001) felt that there is a strong motivation to consider alternatives to free discharge water such as the use of in-stope air coolers. The authors state that an optimisation of cooling resources can be achieved by cycling in-stope air coolers for use only during periods of demand. In this way the overall chilled water quantity can be greatly reduced.

Den Boef (2003) investigated the DSM potential of underground services by using three case study mines. For each mine the DSM potential was calculated using simulation, calibration, verification and optimisation. The study only focused on DSM strategies for the bulk air coolers and waste water pumping. A strategy was implemented at a mine for a trial period of 3 months. The implementation had mixed success due to the mine opting for the control to be done manually by a mine representative. The implementation results together with the case study results were used to estimate the national mining sector DSM potential. It was concluded that through DSM and load shifting actions alone Eskom won't avert the electricity crisis it's facing. The author proposed that Eskom amend the tariff structure to reflect the true cost of electricity, but more important, provide incentives for DSM and load shifting.

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Swart (2003) developed a mathematical model to minimise the electricity cost of a deep mine by optimising the use of the refrigeration plants and ventilation fans. The model was evaluated by using information from a case study mine. The model was firstly used to determine the savings potential by sustaining the environmental condition at $25.5^{\circ}\text{C}_{\text{wb}}$. This resulted in limited savings. The author then used an approach of controlling the air cooling power supplied to the workers at $300\text{W}/\text{m}^2$. This led to a cost saving of 70% compared to the wet bulb approach. However, to realise these savings capital investment is necessary in the form of control instrumentation and variable speed drives for the fans. Currently mines are hesitant to install variable speed drives due to operational and maintenance issues. The model was not implemented at the mine and left for a future study along with the possibilities to use the model for automatic real time energy management.

Prinsloo (2004) did an energy optimisation by means of load shifting on the pumping system of a gold mine. A real time energy management system was installed on the mine and the pumps were controlled according to certain constraints. The project formed part of Eskom's DSM initiative. The average load shift obtained for a five month period (June – November 2004) was 3.66MW resulting in an average monthly cost saving for the mine of kR60.

Webber-Youngman (2005) developed an integrated approach for optimising hot mines, concentrating on ventilation, air cooling and pumping. In essence this study is the same work as conducted by Swart (2003), with the only part added the optimisation of the clear water pumping. The study includes an extensive discussion on variable speed drives and contaminants but it was never implemented in the model.

2.9. Mine simulation methods

In the literature there are ample references to the simulation of mine tunnel ventilation as well as commercial software available for this task. In this section only a few of the references will be listed.

Wu and Topuz (1998) developed two models for the optimisation of ventilation control device locations and sizes. The first is a mixed integer model for controlled flow networks and the second is a nonlinear non convex model for semi-controlled flow networks.

Ramsden *et al.* (2001) developed guidelines for ventilating and cooling ultra-deep mines by developing a model mine for mining at depths from 4000m to 5000m. Various cooling and ventilation strategies were studied by implementing an iterative procedure consisting of:

- A thermodynamic simulator to determine airflow and temperature distributions throughout the mine.
- A spreadsheet program for determining water flow requirements for the air coolers. Based on the water flow the reticulation system was sized and water temperature increases determined.
- A refrigeration simulator to determine the size of the underground refrigeration plant.
- Determining the capital and operating costs for the ventilation system, refrigeration plant and chilled water reticulation system.

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Twort *et al.* (2002) developed an approach to the energy analysis of mine cooling systems using the concept of thermal exergy analysis. The reason behind the development of the new approach was mainly because mechanical mine cooling systems are often adopted which can be expensive in both the associated capital and running costs. To optimise the costs the authors felt it essential to provide the mine operator with a method to determine the most cost effective and efficient mine cooling system. The exergy approach included models of various cooling systems as well as chilled water distribution networks.

Hu *et al.* (2003) developed a nonlinear model to simulate air flow in coal mine ventilation networks. The model can also be used for water distribution networks. Lowndes *et al.* (2004) identified a need to improve the efficiency in the design and operation of auxiliary ventilation systems to maintain an adequate underground environment. A climate prediction model was developed to predict the psychrometric and thermodynamic conditions within long rapid development single entry tunnels taking into account the mass and heat transfer.

Lowndes *et al.* (2006) conducted a sensitivity analysis on the model developed earlier (Lowndes *et al.*, 2004) by using measured climatic survey data from a case study mine. The purpose of the analysis was to evaluate the impact of the variation in the range of specific input parameters.

There is a commercial product currently on the market called VUMA developed for detail mine ventilation, cooling and environmental control (VUMA, 2006). This is a suite of software for the analysis of mine networks, systems and components. Vuma-network can be used for the simulation of steady-state environmental conditions encountered underground. It solves airflow, pollutant and temperature distributions throughout a ventilation circuit or cooling network.

2.10. Summary

The two main conclusions that can be made from this survey are:

- Alternative cooling methods are getting a lot of attention from both a research and industry perspective. However, limited success was gained until now through most of the attempts. This simply emphasizes the need for the proposed spot-cooler development for deep mine applications.
- DSM is being implemented on a wide scale in South African mines with varying success. The implementation of EE projects are starting to get more attention due to the electricity concerns of the mining industry with the migration to greater depths. This implies that the proposed spot-cooler may strengthen current and future DSM and EE initiatives, especially in deep mines.

In the next chapter the design and characterisation of the ACU will be discussed in more detail.

CHAPTER 3

AIR COOLING UNIT DESIGN & CHARACTERISATION

CHAPTER 3

3. AIR COOLING UNIT DESIGN & CHARACTERISATION

The focus of this chapter is to describe the design and characterisation of the first prototype Air Cooling Unit (ACU). The chapter will begin with an explanation of the modified spot cooling concept, followed by an outline of the most important design specifications of the ACU. This results in a brief description of the detailed design leading to the laboratory testing of the first prototype. Based on the tests the ACU characterisation will be discussed. To conclude this chapter the characterisation of two different CWCs will be done for implementation in the mine simulation model along with the ACU model.

3.1. The modified spot cooling concept

In paragraph 2.7 the concept of spot cooling, or localised air cooling, was explained with the help of Figure 2.3. It was also stated that a possible solution for the problems encountered using CWCs is the use of a localised refrigeration plant acting as a spot cooler in remote locations. Figure 3.1 shows an example of a 250kW machine in a gold mine producing chilled water for air cooling in a closed circuit to cater for hot spots.

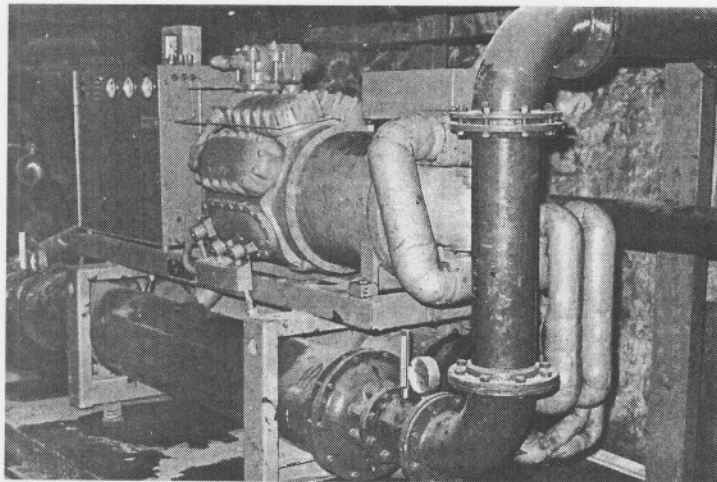


Figure 3.1: A 250kW spot cooler.¹

According to Burrows (1989) the capacity of scattered underground refrigeration plants range from 250kW to 1.75MW with some units even providing as much as 3.5MW. The advantages of the scattered system include:

- Low capital cost.
- Good positional efficiency.

¹ (Burrows, 1989)

The disadvantages include:

- Lack of flexibility. To move an installed plant to a new location is very costly.
- Difficulty to maintain due to limited accessibility.
- Heat rejection is often limited by the lack of return air.

According to Burrows (1989) there was a tendency to move away from scattered plants except in the cases where very low capacities were involved.

The new ACU concept proposed in this thesis is illustrated in Figure 3.2. It shows an ACU used in a remote location to supply cold air to a problem area.

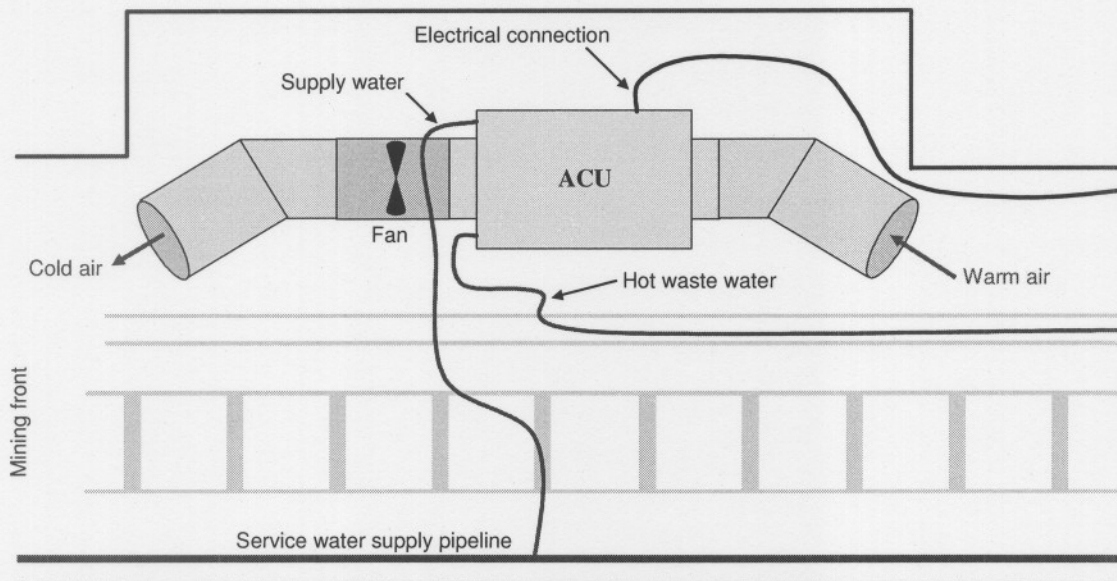


Figure 3.2: Modified spot cooling concept.

The main difference between the conventional spot cooling concept explained in Chapter 2, and the new ACU spot cooling concept shown here is the quantities and temperature of the water required as energy transport medium. For the conventional concept a supply of chilled water between 10°C and 15°C is a necessity for the CWCs to operate effectively, by using the water as a transfer medium to extract energy from the warm air. Similar to this the ACU concept, employing a refrigeration unit, also needs a water supply as transfer medium. However, this water does not have to be chilled and can be regular service water. This service water can either be supplied from surface or from an underground storage dam.

Because it is a vapour compression cycle operating at a condensing temperature in the region of 50°C and employs a very effective condenser coil, the inlet water temperature can be as high as 40°C. The water temperature will however have an effect on the mass flow rate through the condenser which will increase accordingly with an increase in inlet water temperature. The vapour compression cycle employed is shown in Figure 3.3. The energy absorbed through an air-to-refrigerant evaporator is

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rejected via a refrigerant-to-water condenser into the waste water return of the mine level. The resultant effect is that cold air is supplied by the unit while warm water is dumped in the waste water return and pumped away. The main purpose of a spot cooler is therefore to move energy from an area where there is a heat problem to a section where this additional heat load can be managed more easily.

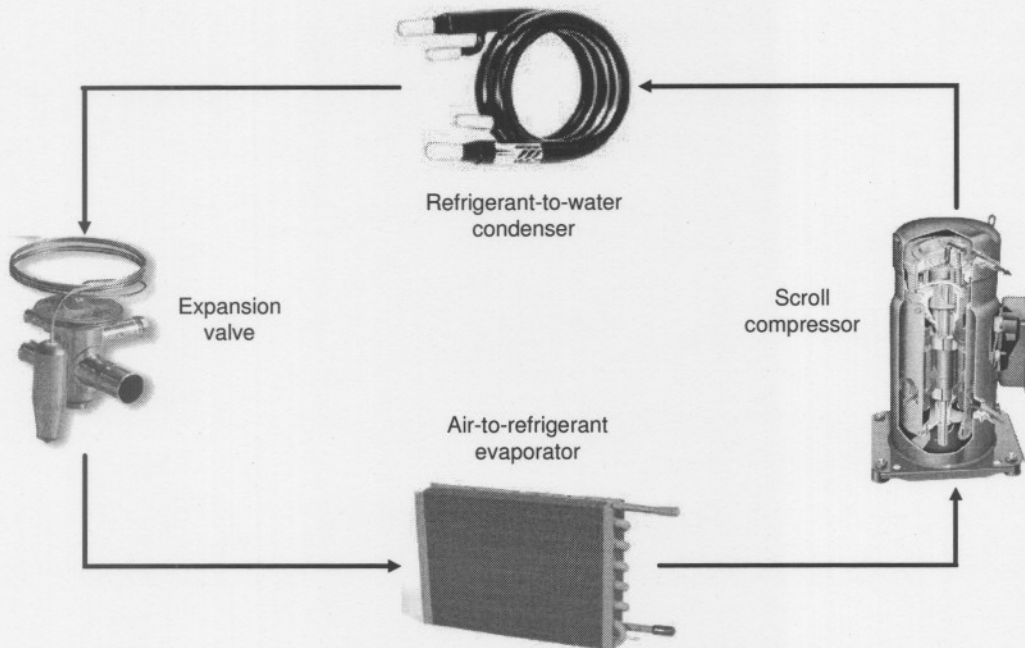


Figure 3.3: Refrigeration cycle of the ACU.

3.2. Basic concept of the ACU

The concept of the ACU is based on existing heat pump technology for the heating of sanitary hot water in the commercial building sector. To understand the advantages of the heat pump technology for the mining industry, it is necessary to first explain what a heat pump is. A heat pump is a mechanical device that uses a refrigerant, like R-134a, to transfer heat from a low temperature source to a high temperature sink via a vapour compression cycle that requires power input from a compressor. This was illustrated in Figure 3.3.

For the heating of sanitary hot water in the commercial building sector the emphasis of the heat pump falls on heating, with cooling as a by-product. As shown in Figure 3.4, typically for each 1kW of electrical input at the compressor, approximately 2kW of energy is absorbed from the ambient air, resulting in approximately 3kW of energy transferred to the water for heating. This relates to a coefficient of performance (COP) of approximately two for cooling and three for heating i.e. $COP_{cooling} \approx 2$ and a $COP_{heating} \approx 3$.

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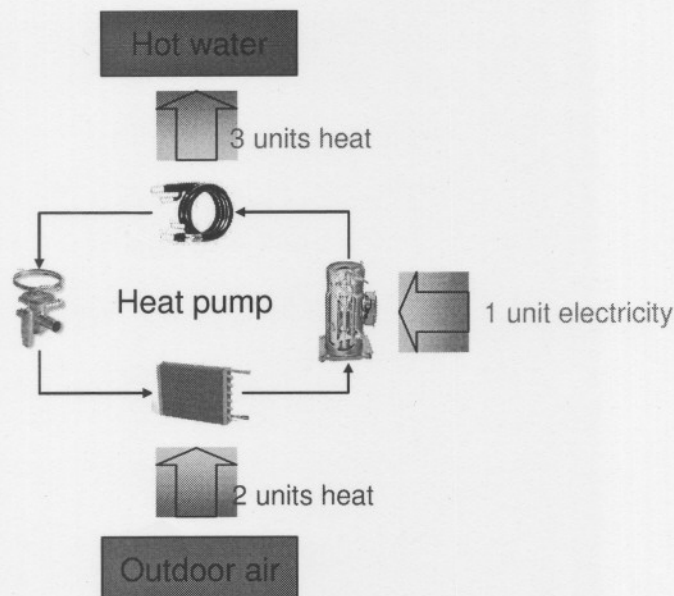


Figure 3.4: Basic concept of a heat pump.

Over the past few years a specialised and innovative heat pump called the 'in-line heat pump', making use of advanced technology components and methodologies, was developed (Rousseau and Greyvenstein, 2000). A major feature of the in-line heat pump technology is that high outlet water temperatures ($60^{\circ}\text{C} - 65^{\circ}\text{C}$) can be obtained for a wide range of inlet water temperatures ($15^{\circ}\text{C} - 50^{\circ}\text{C}$) and at relatively low condensing temperatures ($< 55^{\circ}\text{C}$). This is not typical for conventional heat pumps and large water chilling plants where the required outlet water temperature is usually much lower.

This ability of the ACU to operate with high outlet water temperatures is ideal for underground air cooling in the mining industry since it means that a large temperature difference can be obtained on the water side. This result in very low water mass flow rates needed to transport the energy extracted from the air away from the area being cooled.

Looking back at the explanation of a heat pump it must be noted that to achieve 2kW of air cooling approximately 3kW of energy must be transferred via the heated water. It is important to realise that the water needs to be pumped away from the area where the cooling takes place, otherwise more heat will be added to the air than is being removed for cooling due to the input from the compressor. However, due to the the high heat capacity of water compared to that of air, it functions as a good energy transport medium. Most mines in South Africa have large quantities of waste water at temperatures of around $25\text{-}35^{\circ}\text{C}$ flowing to underground dams. By adding to it a relatively low flow rate of water at a temperature of approximately 45°C should not significantly contribute to an increase in the average temperature of the waste water return system. However, the full impact of this will be analysed in detail in the subsequent chapters.

3.3. Design specifications of the ACU

Before development of the prototype could commence it was necessary to identify the critical design specifications for effective underground application of the proposed ACU. During the design phase it was necessary to take note of a statement made by Van der Walt and De Kock (1984). They stated that in the past, stope cooling systems required a considerable amount of attention by personnel. It was felt that should production personnel be required to assume responsibility for their own local environmental requirements, this could interfere with their production activities resulting in lower productivity or production losses. This implies that for the ACU to be successful it should be a stand-alone unit requiring minimal operating skills and maintenance. In order to identify the critical design specifications three categories were used namely: operating conditions, physical specifications and performance specifications.

3.3.1. Operating conditions

- Remote locations - The main area of application for the unit will be remote underground locations where the current cooling systems are ineffective.
- Air temperatures - The unit will operate mainly in areas where the wet bulb temperature is 31.5°C and above.
- Water temperatures - The service water temperature in these locations may be as high as 35°C.
- Electricity - The unit must operate on 525V, three phase power supply, as this is the standard used in underground operations in South Africa.

3.3.2. Physical specifications

- Modularity - The design of the unit should be such that an ACU can be used as a standalone unit or connected in series/parallel with other ACUs to increase the total installed cooling capacity.
- Mobility - The unit should be mobile and comply with the transport infrastructure of a mine. Furthermore, the unit should be easily transported to progress along with the mine expansion.
- Size - The unit must fit inside the shaft cage for convenient transport from surface to the underground levels. The size should furthermore be such that the unit can easily be placed in an area blasted into the rock wall next to the rail, also known as a cubby.
- Weight - The total weight of the unit should be within the limits for underground rigging from the rail to the installation position.
- Robustness - It must be robust to withstand the rugged handling and blasting conditions to which equipment is subjected in a mining environment.
- Connections - The connection of a fan, electrical supply and water should be convenient to comply with the statement made by Van der Walt and De Kock (1984).

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- Operation - The unit should be straightforward to operate by qualified mining personnel. Complex operating procedures resulting in the necessity to train personnel should be avoided as far as possible.
- Maintenance – The underground maintenance of the unit should be low-key. On the air side the coil should be easily accessible for cleaning. The coil should be designed in such a way to minimise fouling.

3.3.3. Performance specifications

- Cooling capacity - The cooling capacity should be such that the unit complies with the physical specifications of being modular and mobile. Based on preliminary calculations the cooling capacity was set at 80kW nominal.
- Air temperature drop - The air outlet wet bulb temperature should be between 5°C and 10°C lower than the inlet wet bulb temperature based on the air volume flow rate through the unit.
- Water mass flow rate - The unit must use as little water as possible to reach the desired cooling capacity.
- Refrigerant - Only refrigerants approved by the mining industry may be used.

3.4. Design of the ACU

The design of the prototype ACU was done by the author and is described in more detail in Appendix A. In this section only a brief description of the design will be given. The design was divided into two main headings, namely the mechanical design and the refrigeration plant design.

3.4.1. Mechanical design

To make the ACU modular and mobile it was decided to equip the ACU with a rolling stock undercarriage fixed to the cabinet housing the refrigeration components. This makes transportation of the unit convenient for mining personnel. The physical layout of the ACU was divided along the length into two compartments as shown in Figure 3.5. The one compartment contains all the refrigeration components excluding the evaporator, and in the second compartment only the evaporator coil is installed. Due to the dust content underground it is advisable to keep the compressor and electrical components in a semi dustproof section away from constant air movement to extend the life expectancy of the components. From a maintenance perspective cleaning the evaporator coil also becomes relatively easy without being concerned that the compressor or electrical components will be damaged.

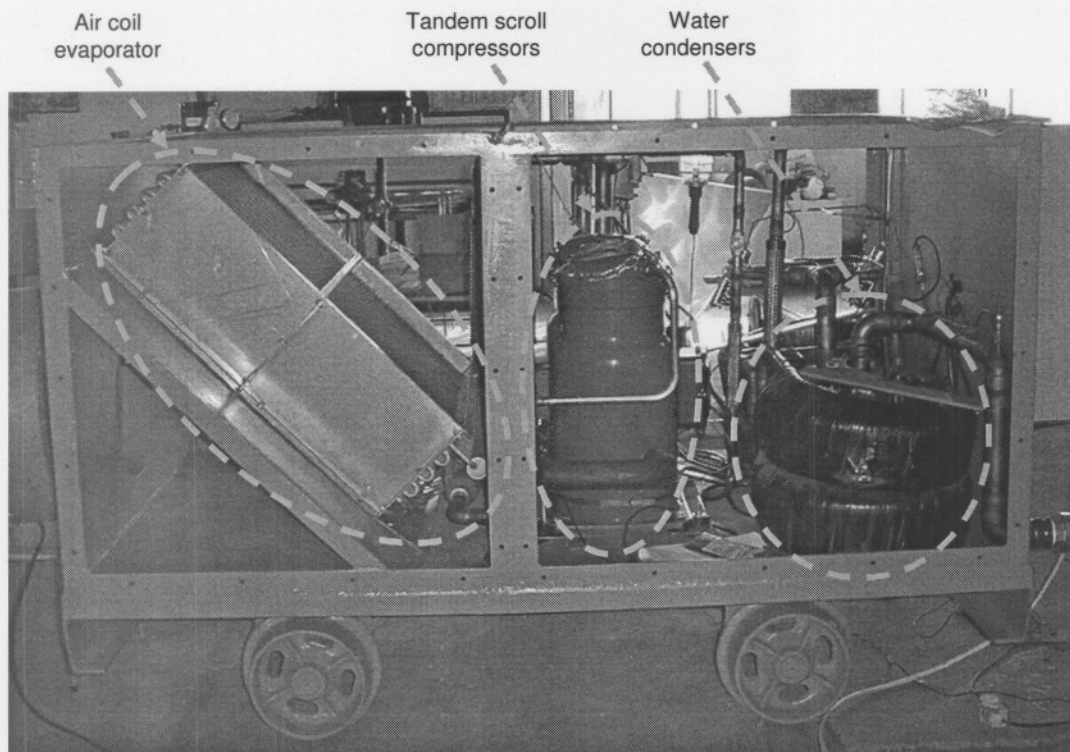


Figure 3.5: Internal layout of the prototype ACU.

The evaporator compartment has one air inlet and one air outlet as can be seen from Figure 3.6. This is for the unit to operate in two possible configurations. The first configuration consists of the ACU being connected to the fresh air ducting installed in the haulage. The fresh air is then cooled through the evaporator and blown directly into the area being cooled or ducted back into the fresh air supply sent deeper into the mine. The second configuration consists of the ACU being placed in an area needing direct cooling. Warm surrounding air is sucked through the top opening over the evaporator and then blown directly via a fan back into the environment.

Figure 3.6 also shows the ACU when all the side panels have been installed. The water inlet and outlet are also located on the same side as the compartment housing the water condensers.

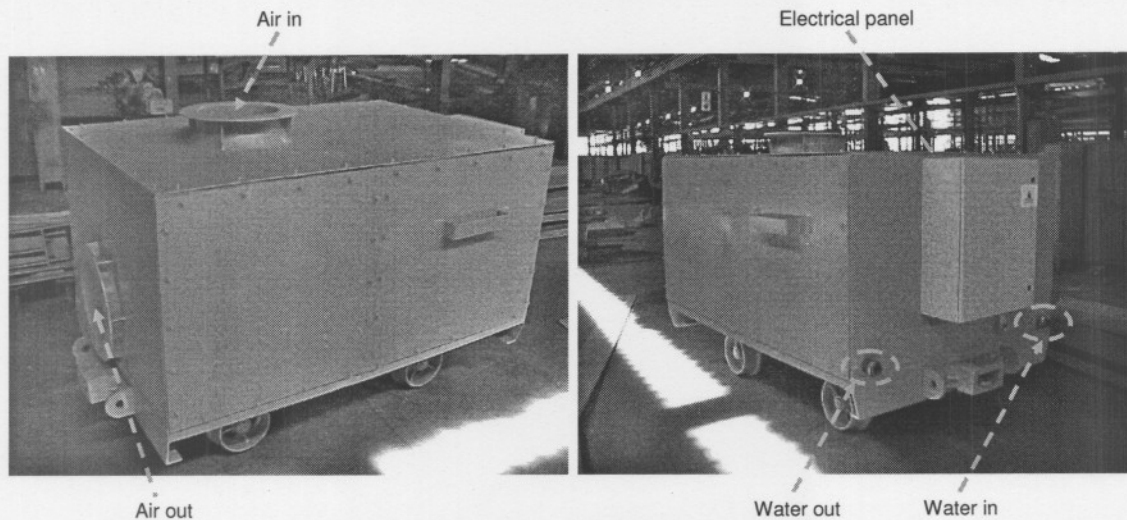


Figure 3.6: External view of the prototype ACU.

3.4.2. Refrigeration plant design

To comply with mine regulations regarding the use of refrigerants, the design was done for working fluid R134a. The main components of the refrigeration cycle discussed here are shown in Figure 3.3.

3.4.2.1. Compressor

In order to make the ACU rugged but at the same time maximising the efficiency, it was decided to implement scroll compressor technology. Due to the low capacity of single scroll compressors the design had to cater for a tandem compressor configuration. The Maneurope Performer SZ370 was selected and its performance at $15^{\circ}\text{C}_{\text{evap}} / 50^{\circ}\text{C}_{\text{cond}}$ ARI conditions are listed in Table 3-1.

Table 3-1: Specifications of the tandem scroll compressor.

Description	Value
Cooling capacity	84.5kW
Power consumption	16.8kW
$\text{COP}_{\text{cooling}}$	5.03

3.4.2.2. Evaporator

To cool the air it was decided to opt for an air-to-refrigerant finned-tube heat exchanger. This increases the positional efficiency of the ACU and reduces any losses involved with using chilled water as an intermediate and piping it to a CWC. The evaporator was designed for an evaporating temperature of 15°C due to the nearly constant high wet-bulb temperature conditions encountered underground. For this reason an expansion valve with a maximum operating pressure was installed to ensure that the evaporating pressure doesn't go above this critical point and damage the compressor.

Due to the potential dust content that may result in blockage of the airside, the coil was designed with only 6 fins per inch. This should also allow more convenient cleaning of the coil on the air-side.

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Slightly higher than normal air on-coil velocities were also specified in an effort to reduce the potential for dust and grime to accumulate between the fins.

Based on a compressor cooling capacity of about 84.5kW the performance of the designed coil is listed in Table 3-2. More detail on the coil simulation and design is given in Appendix A.

Table 3-2: Evaporator coil performance.

Description	Value
Total duty	84.69kW
Sensible duty	21.74kW
Air volume flow rate	2.53m ³ /s
Air face velocity	3.16m/s
Inlet dry bulb temperature	34.0°C
Inlet wet bulb temperature	32.5°C
Outlet dry bulb temperature	26.8°C
Outlet wet bulb temperature	26.7°C
Air pressure drop	147Pa
Refrigerant pressure drop	4.63kPa

3.4.2.3. Condenser

Based on the compressor cooling capacity of 84.5kW and a power input from the compressor of 16.8kW the condenser was sized to transfer at least 101.3kW from the refrigerant to the water. In order to obtain high water outlet temperatures at relatively low condensing temperature a fluted tube-in-tube condenser is employed. Such a condenser has a fluted inner tube with a smooth outer tube. The refrigerant flows in the twisted annulus through small channels formed due to the shape of the fluted tube, and the water flows in the inner tube, in a counter flow configuration. A more detailed explanation on fluted tubes is given in Appendix A.

As stated earlier the compressor selection was done at a condensing temperature of 50°C. The reason for this is that the water can be heated to a higher outlet temperature resulting in less water circulated through the ACU which in itself has an effect on the total pumping required in the mine. The water outlet temperature is controlled by a water regulating valve sensing the pressure on the refrigerant side. To obtain the required 101.3kW it was necessary to install four BTSSC-84 fluted tube condensers in parallel, each with a capacity of about 28kW.

3.5. Experimental evaluation

3.5.1. Test facility

In order to test the ACU in the laboratory it was necessary to create an environment with similar temperature conditions as those found underground, i.e. wet bulb temperatures in the region of 31.5°C. Working on the assumption that the evaporator coil would be completely wet once it is

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operational in humid underground conditions, it was decided to employ a water sprayer system to reach the required temperature range. Hot water was firstly cycled through a water-to-air coil to increase the dry bulb temperature of the supply air. Next warm water was sprayed into the heated ambient air flowing towards the evaporator to increase the wet bulb temperature of the air. This is illustrated in Figure 3.7. The air volume flow rate through the unit was varied using a damper on the outlet side of the ACU.

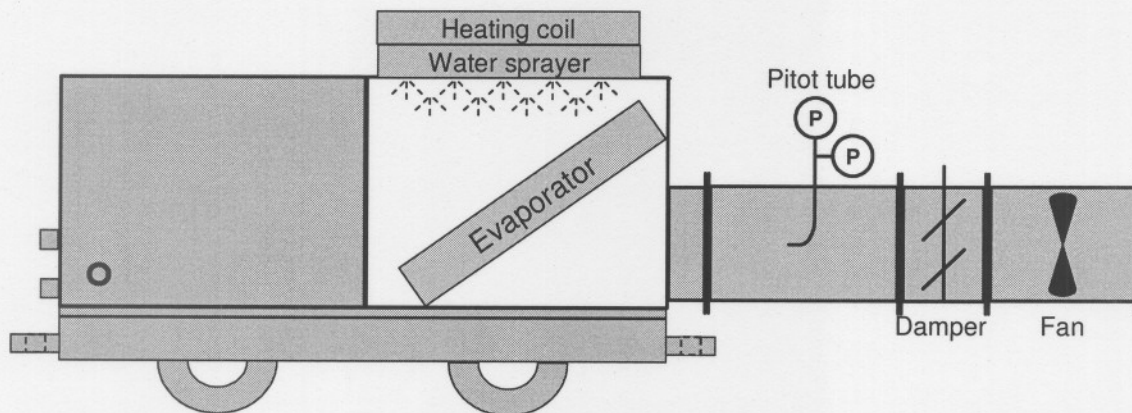


Figure 3.7: Air side test set-up.

Due to the ACU operating on a 525V electrical supply a step-up transformer was installed to transform a standard commercial supply of 380V into the required 525V. It was furthermore important to be able to vary the water temperature supplied to the condenser coils to fully characterise the ACU. This was done by employing an existing heat pump test facility equipped with a dual function heat pump and mixing valves as illustrated in Figure 3.8. The test matrix will be discussed in paragraph 3.5.3.

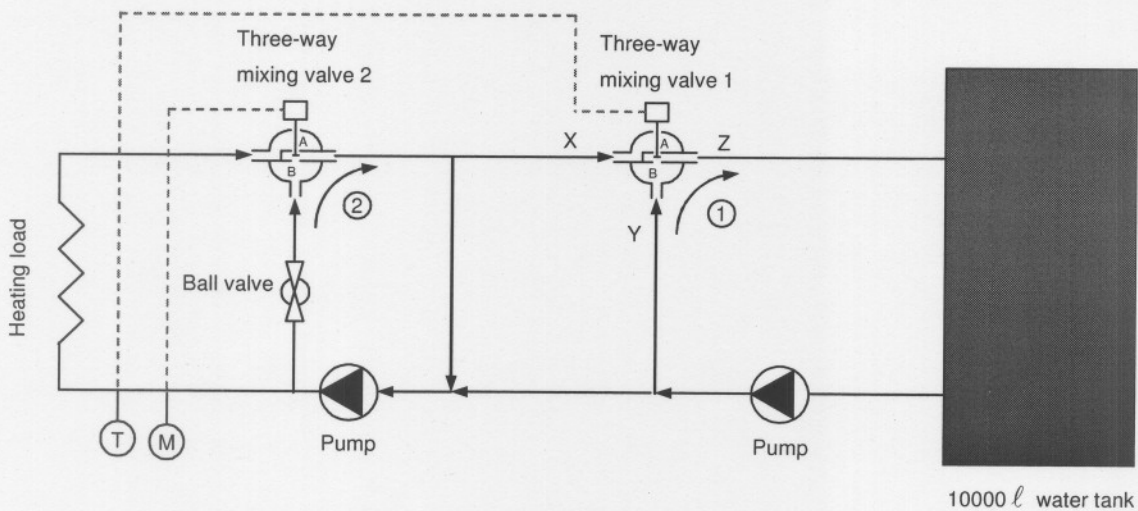


Figure 3.8: Water loop supplying the condensers.

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3.5.2. Measuring equipment

In order to fully characterise the performance of the ACU it was necessary to install measuring equipment at critical points in the cycle. This equipment was used for both the laboratory tests and underground testing. The position of the equipment shown in Figure 3.9 can be described as follows:

Compressor:

- The power input to the compressor is determined by measuring the electrical current and voltage of the three different phases and converting it into kW using a power meter.
- On the refrigerant side the condensing and evaporating pressures are measured to determine the respective condensing and evaporating temperatures. This is used as an indication whether the unit is operating close to the desired conditions stated in Section 3.4.2. This is also used for comparisons with the data sheets of the compressor.

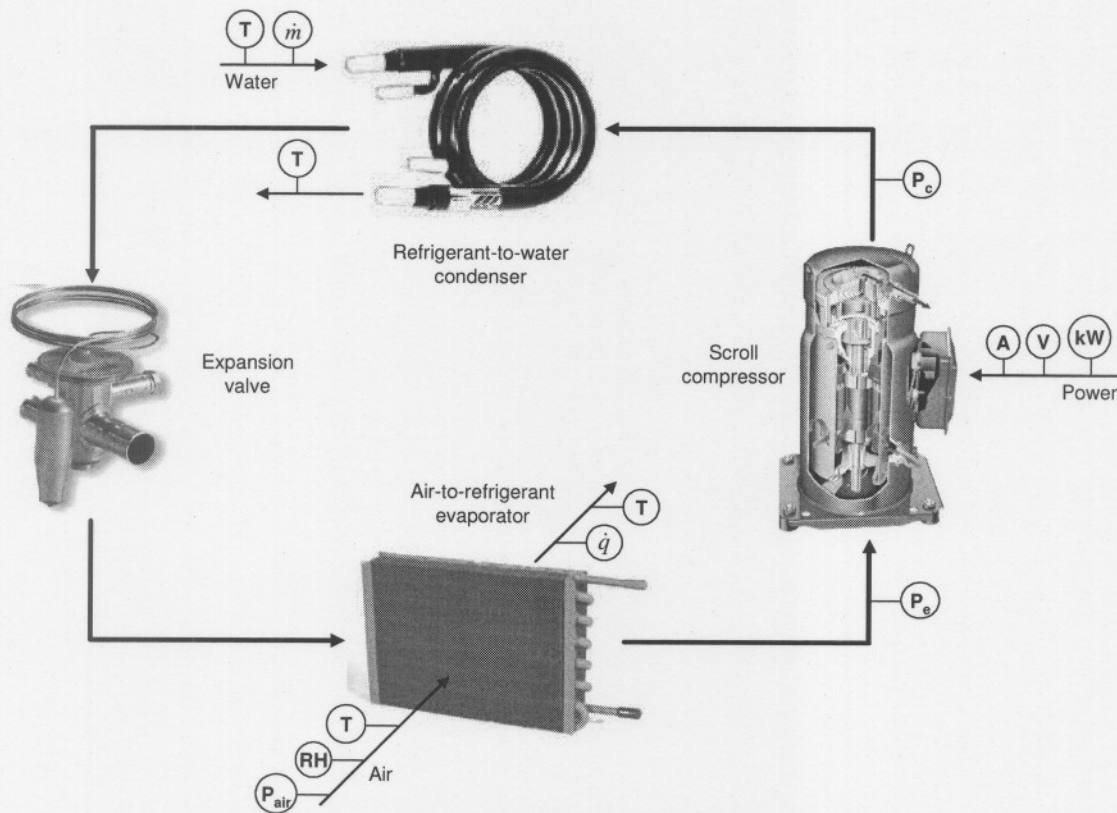


Figure 3.9: Instrumentation positioning.

Refrigerant-to-water condenser:

- Both the inlet and outlet water temperatures are measured.
- The water mass flow rate through the condenser is also measured to calculate the energy transferred from the refrigerant to the water as follows:

$$q_{water} = \dot{m} \cdot c_p \cdot \Delta T. \quad (3-1)$$

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Air-to-refrigerant evaporator:

- On the inlet side the air dry bulb temperature, relative humidity and atmospheric pressure are measured. From this the air inlet enthalpy can be obtained.
- As shown in Figure 3.7 the air side volume flow rate was determined by using a pitot tube connected to pressure transmitters in the discharge ducting.
- From Table 3-2 it is evident that based on the evaporator being sufficiently oversized the outlet air humidity approaches 100%. Based on this, an assumption is made that the air exiting the unit will be 100% humid. The enthalpy of the outlet air can now easily be determined.
- The energy transferred from the air to the refrigerant can then be calculated from:

$$q_{air} = \rho \cdot \dot{q} \cdot \Delta h. \quad (3-2)$$

The data loggers used to gather all of the measurements are battery operated for convenient underground application. The information gathered by the loggers can be downloaded to a computer for processing.

To validate the accuracy of the measurements it was necessary to conduct an uncertainty analysis on the measuring equipment. The detailed uncertainty analysis is described in Appendix B. In summary, the average uncertainty in determining the heat transfer on the condenser side was found to be 1.35%, and on the evaporator side it is 4.13%.

3.5.3. Test matrix

To properly evaluate the performance of the ACU over varying conditions a test matrix had to be generated. During the discussion on the modified spot cooling concept in section 3.1 it was stated that the unit is not dependant on chilled water and can operate using inlet water temperatures even as high as 40°C. In practice a typical maximum wet bulb temperature of 33°C is found in areas where personnel work, and the water temperature can only approach the wet bulb temperature at best. Working with a delta approach temperature of 3°C the maximum water supply temperature for the test phase was limited to 30°C.

On the air side the maximum wet bulb inlet condition was set at 33°C. Because the ACU can also be used in areas with lower wet bulb conditions the minimum wet bulb temperature was set at 23°C to determine the performance of the unit under these conditions.

To determine the effect of the air volume flow rate on the performance it was varied using a damper as shown in Figure 3.7. For testing purposes the ACU was equipped with a 4.0kW mine ventilation fan of which the volume flow rate was varied between 1.9m³/s and 3.1m³/s. A summary of the test conditions is given in Table 3-3.

Table 3-3: ACU test matrix.

Condition	Range
Water inlet temperature	$14\text{ }^{\circ}\text{C} < T_{\text{water, in}} < 30\text{ }^{\circ}\text{C}$
Air inlet temperature	$25\text{ }^{\circ}\text{C} < T_{\text{db, in}} < 45\text{ }^{\circ}\text{C}$ $23\text{ }^{\circ}\text{C} < T_{\text{wb, in}} < 33\text{ }^{\circ}\text{C}$
Air volume flow rate	$1.9\text{m}^3/\text{s} < \dot{q} < 3.1\text{m}^3/\text{s}$

3.5.4. Test results

The most important results of the laboratory tests are shown in Figure 3.10 to Figure 3.12. Each data point is the average of a 10 minute sampling interval. The data points on each graph were divided into three groups based on air mass flow rate regions. Figure 3.10 shows the air outlet wet bulb temperature given a certain inlet wet bulb temperature. At low inlet wet bulb temperatures and low volume flow rates a ΔT of about $10\text{ }^{\circ}\text{C}$ is obtained. At higher volume flow rates accompanied by high inlet wet bulb temperatures a ΔT of about $5\text{ }^{\circ}\text{C}$ is obtained. The average change in wet bulb conditions over the test range is $7.4\text{ }^{\circ}\text{C}$.

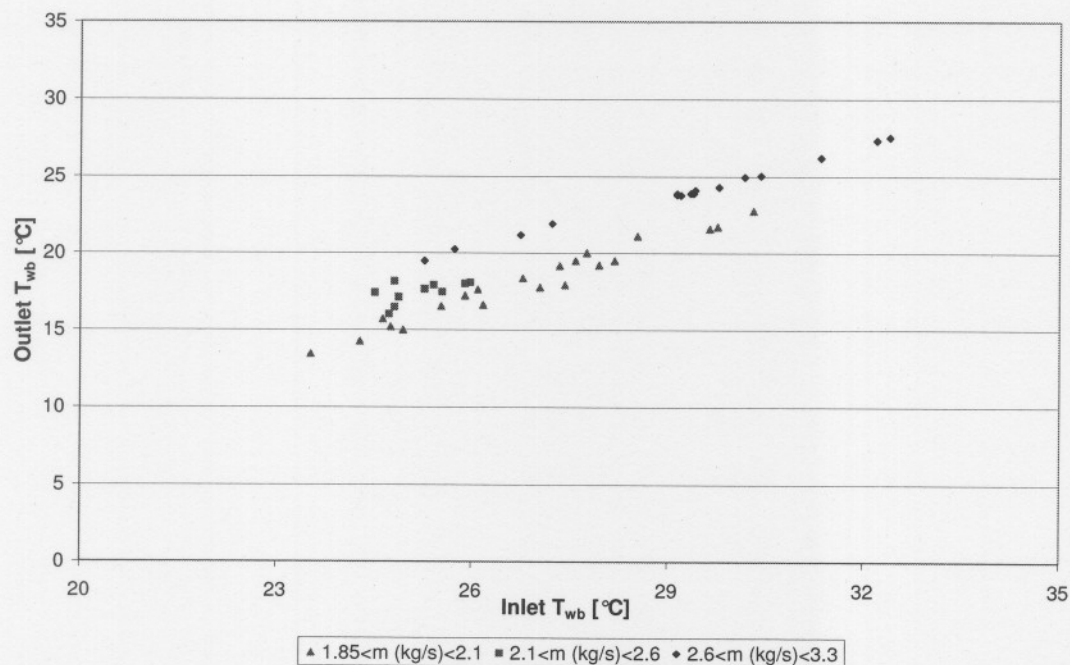


Figure 3.10: Change in wet bulb temperature over coil.

What is evident from the graph is that the ACU can cater for the two main mine ventilation philosophies. The first philosophy is to supply a working area with a high volume flow rate thus reducing the temperature in the area by only a few degrees. The high velocity of the air provides the cooling effect. The other philosophy consists of supplying a working area with cold air but at a lower

volume flow rate. The low temperature gives the cooling effect. In both instances approximately the same amount of cooling is done.

Figure 3.11 illustrates the mass flow rate for different inlet water temperatures. As stated earlier the ACU is equipped with a water regulating valve controlling the outlet water temperature between 45 °C and 50 °C by varying the mass flow rate. As can be seen from the figure, during the tests the unit used between 0.667 l/s and 1.195 l/s. The potential economic impact of the unit, to be discussed in the chapters to follow, is closely linked to the amount of water the ACU uses compared to other cooling methods like CWCs.

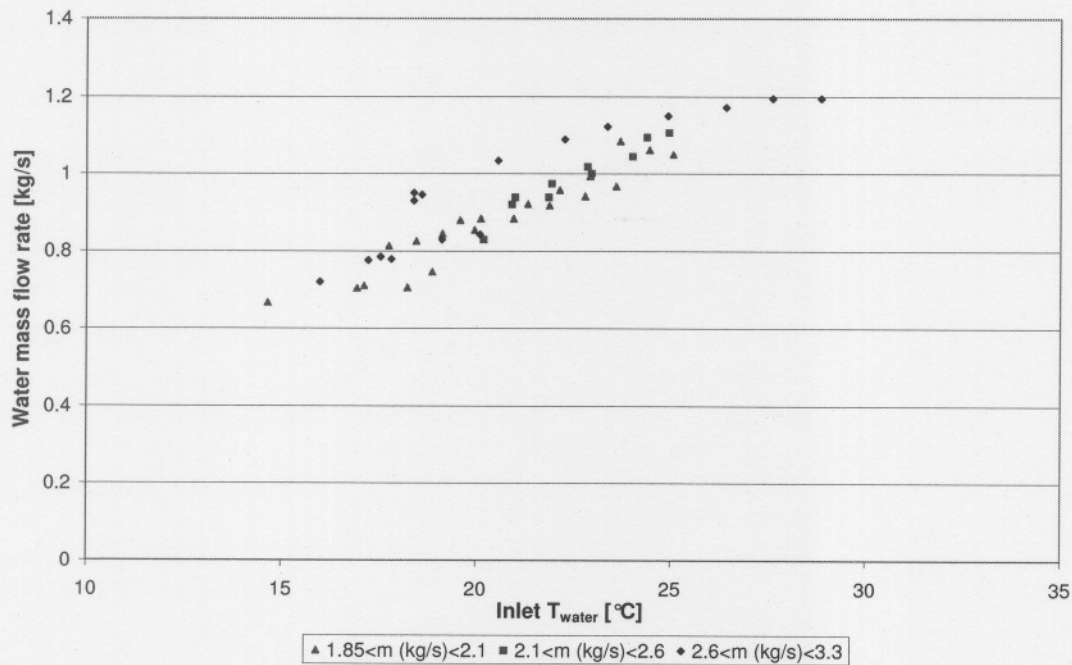


Figure 3.11: Change in water mass flow rate.

Figure 3.12 shows the cooling capacity and COP observed during the tests. The cooling capacity varied between 66.5kW and 82.8kW with an average cooling capacity for the test range of 74.0kW. It should be noted that the unit was designed to deliver about 80kW at wet bulb conditions approaching 32.5 °C. This was only achieved with higher volume flow rates though the unit. The reason for this is that during the laboratory tests it was difficult to obtain high inlet wet bulb temperatures at lower volume flow rates mainly due to the effectiveness of the water spray system explained in section 3.5.1. At higher air flow rates the water droplets was absorbed much easier by the air than at lower flow rates.

Another important aspect of any refrigeration installation is the COP. Looking back at the explanation of a heat pump in section 3.2 it was stated that a cooling COP of about 2 is usually obtained. Due to the high ambient wet-bulb conditions underground it is evident from Figure 3.12 that the COP_{cooling} of the ACU ranges between 3.5 and 4.8 based on the inlet wet bulb conditions. According to the

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manufacturer's specifications listed in Table 3-1 the compressor should have a COP of 5.03 when operated at $15^{\circ}\text{C}_{\text{evap}} / 50^{\circ}\text{C}_{\text{cond}}$ ARI conditions. A COP of about 4.8 is obtained during the tests which is close to the claimed COP. However from experience with compressors the manufacturer's COP is rarely obtained in practice.

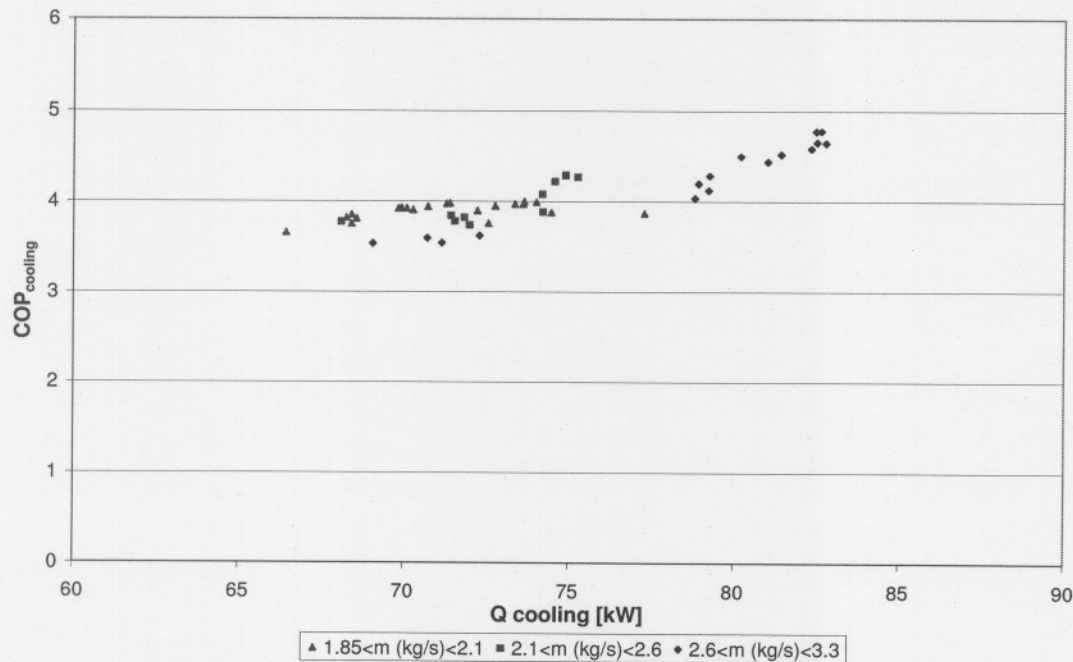


Figure 3.12: COP vs. cooling capacity.

3.6. Characterisation of the ACU

The performance of the ACU is mainly a function of three parameters, i.e. inlet wet bulb temperature, inlet water temperature, and the air volume flow rate. Based on the measured data the ACU was characterised to develop a model that can be implemented in the simulation of a mine water reticulation network. The detail of the reticulation network will be discussed in the next chapter.

Using Figure 3.10 a linear fit was done for each of the three air flow ranges. The air outlet wet bulb temperature of the ACU can be determined from:

$$\begin{aligned}
 T_{wb,out} [^{\circ}\text{C}] &= (1.345 \cdot T_{wb,in} - 18.02) \text{ for } 1.85 \leq \dot{m}_{air} \text{ (kg/s)} < 2.1 \\
 T_{wb,out} [^{\circ}\text{C}] &= (0.8065 T_{wb,in} - 2.877) \text{ for } 2.1 \leq \dot{m}_{air} \text{ (kg/s)} < 2.6 \\
 T_{wb,out} [^{\circ}\text{C}] &= (1.098 \cdot T_{wb,in} - 8.199) \text{ for } 2.6 \leq \dot{m}_{air} \text{ (kg/s)} \leq 3.3
 \end{aligned}
 \tag{3-3}$$

The average accuracy of the temperature correlation is 98.27% with a maximum error of 6.6%. Figure 3.13 shows the accuracy of the correlations. Accuracies was calculated from

$(T_{\text{experimental}} - T_{\text{predicted}}) / T_{\text{experimental}}$, using each data point in the series to calculate the average accuracy and maximum error.

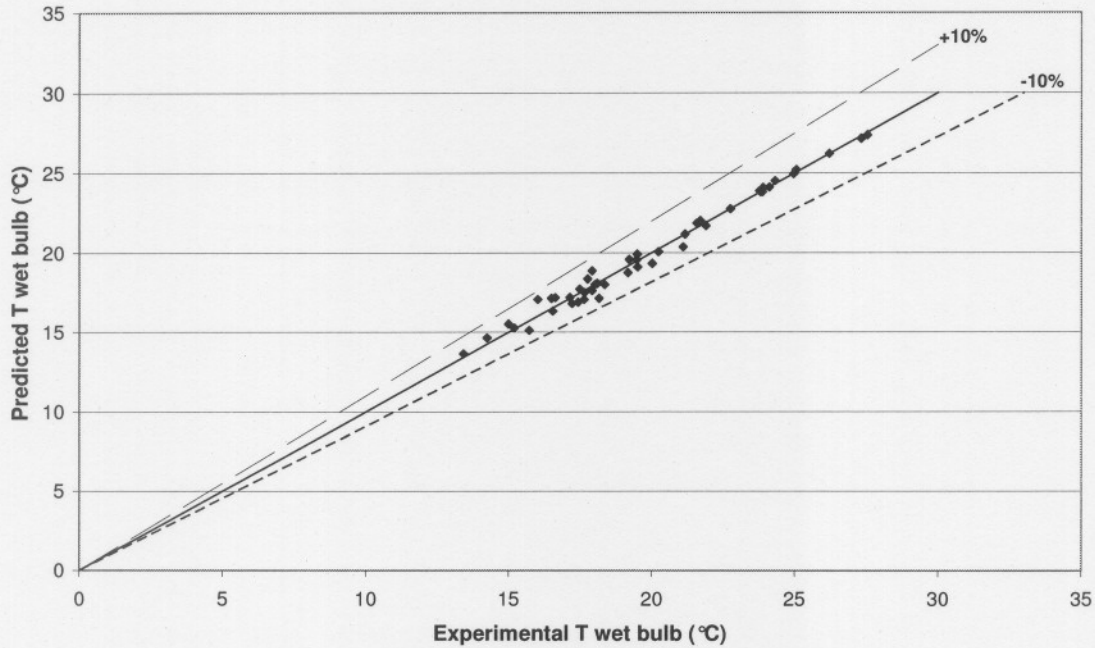


Figure 3.13: Predicted vs. experimental outlet wet bulb temperatures.

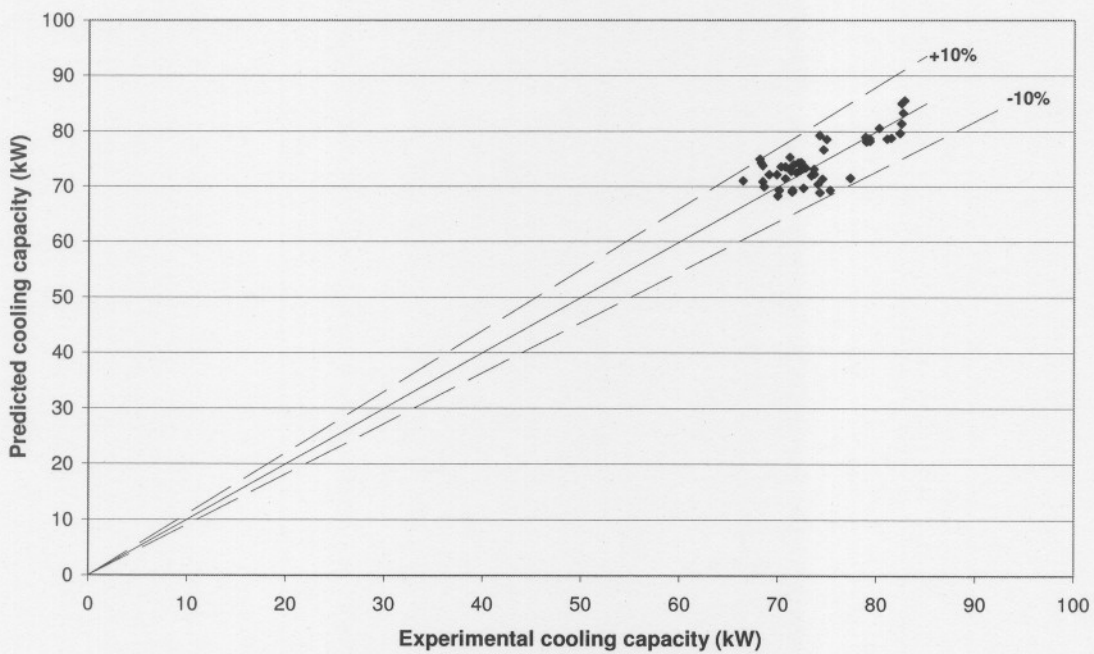


Figure 3.14: Predicted vs. experimental cooling capacity.

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Using the outlet wet bulb temperature correlations combined with the inlet wet bulb temperatures and air mass flow rates the cooling capacity of the ACU can be predicted from Equation 3.2. The average accuracy of the cooling capacity prediction is 96.53% with a maximum error of 10%.

Figure 3.14 shows the accuracy of the prediction. Using Figure 3.12 a linear fit was done to determine the $COP_{cooling}$ as a function of cooling capacity for each of the three air flow ranges. The $COP_{cooling}$ of the ACU can be determined from:

$$\begin{aligned} COP_{cooling} &= (0.0189 \cdot Q_{cooling} + 2.5456) \text{ for } 1.85 \leq \dot{m}_{air} \text{ (kg/s)} < 2.1 \\ COP_{cooling} &= (0.0793 \cdot Q_{cooling} - 1.8048) \text{ for } 2.1 \leq \dot{m}_{air} \text{ (kg/s)} < 2.6 \\ COP_{cooling} &= (0.0914 \cdot Q_{cooling} - 2.926) \text{ for } 2.6 \leq \dot{m}_{air} \text{ (kg/s)} \leq 3.3 \end{aligned} \quad (3-4)$$

The average accuracy of the COP correlation is 96.3% with a maximum error of 13.5%. Figure 3.15 shows the accuracy of the correlations.

Using the COP correlation along with the cooling capacity calculation the power input of the compressor can be determined. The average accuracy of the power input calculation is 97.52% with a maximum error of 8.15%. Figure 3.16 shows the accuracy of the prediction.



Figure 3.15: Predicted vs. experimental COP.

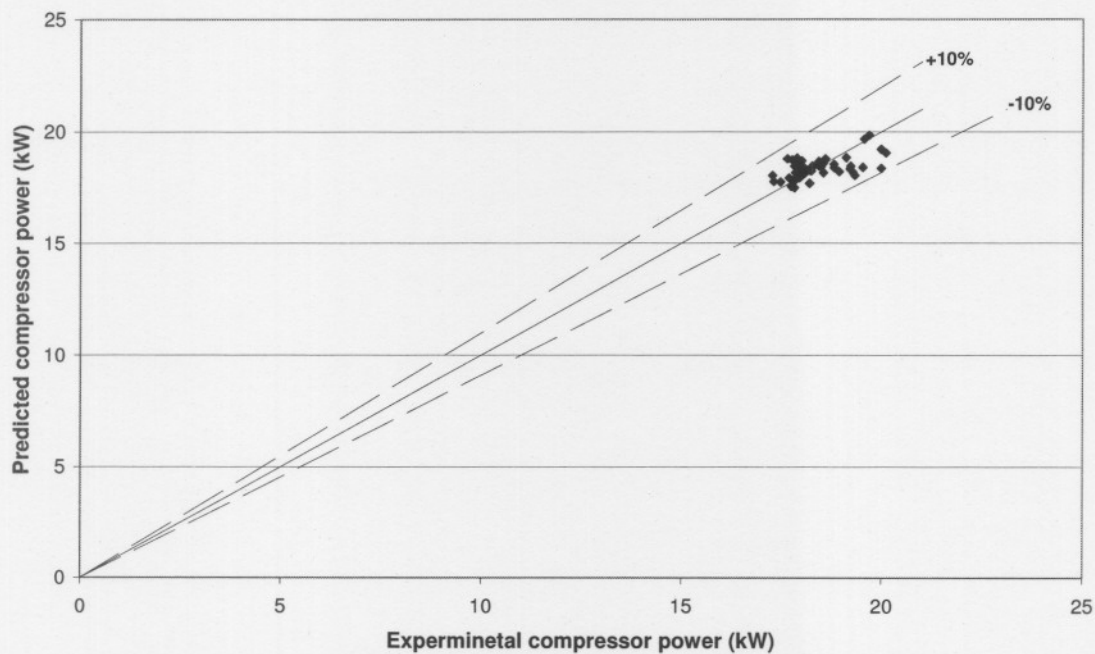


Figure 3.16: Predicted vs. experimental power input.

The heating capacity of the ACU is calculated from:

$$Q_{\text{heating}} = Q_{\text{cooling}} + 0.9 \cdot P_{\text{compressor}} \quad (3-5)$$

During the laboratory tests the water outlet temperature was controlled at an average value of 45°C with a standard deviation of 1.5°C. To simplify the model the decision was taken to keep the outlet temperature constant at 45°C. With the calculated heating capacity, the fixed outlet temperature and the known inlet temperature the water mass flow rate can be calculated using Equation 3.1. The average accuracy of the flow rate calculation is 95.2% with a maximum error of 18.4%. Figure 3.17 illustrates the accuracy of the prediction.

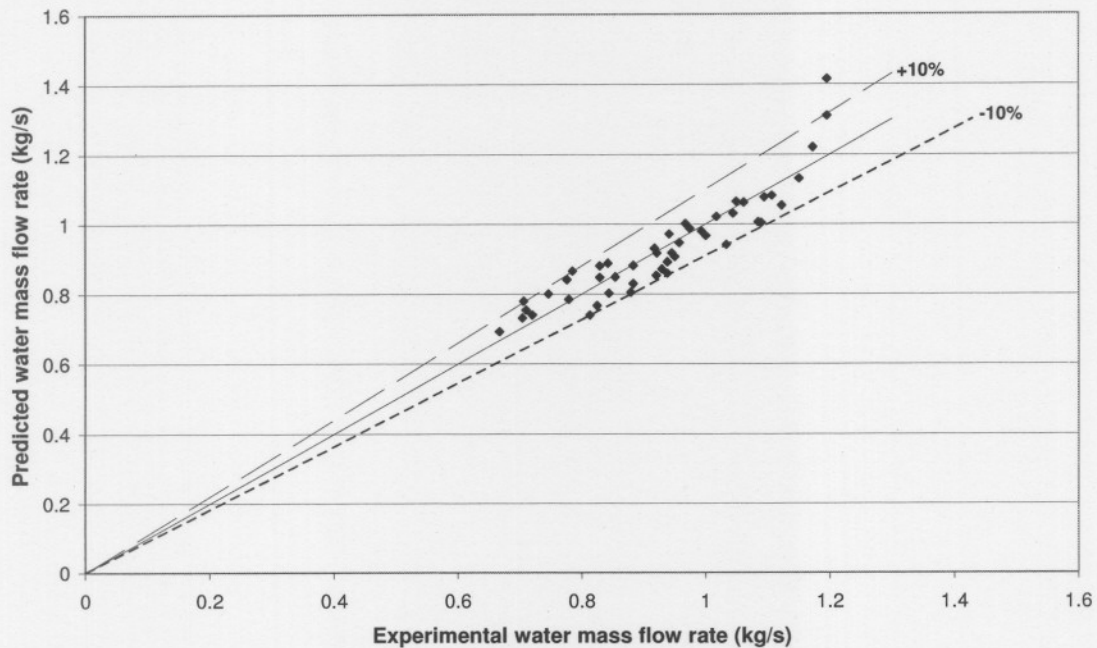


Figure 3.17: Predicted vs. experimental water flow rate.

3.7. Characterisation of the CWCs

For this study the chilled water cooling cars of two different manufacturers were characterised. The physical specifications of the CWCs used for the characterisation are listed in Table 3-4.

Table 3-4: CWC specifications.

Description	CWC #1	CWC #2
Nominal rating (kW)	300	300
Flow conditions		
Air flow rate (m ³ /s)	10.0	10.0
Water flow rate (l/s)	9.0	6.1
Dimensions		
Finned area height (mm)	762	1400
Finned area length (mm)	1750	1500
Rows deep	6	8
Fins/inch	7	7
Circuiting		
Tubes high	20	44
Circuits	10	44
Tubes/circuit	12	8

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Tube		
Inside diameter (mm)	16	10.8
Arrangement	Staggered	Staggered

The CWCs were simulated with HeatFlow 98 ver.10.32.2 for the range of conditions shown in Table 3-5. HeatFlow 98 is a simulation tool developed and used by an international coil manufacturer to design and simulate their range of commercial heat exchangers.

The CWCs are not equipped with water flow control valves and therefore has a constant water flow rate independent on the inlet water temperature.

Table 3-5: CWC simulation conditions.

Condition	Range
Water inlet temperature	$10^{\circ}\text{C} < T_{\text{water, in}} < 30^{\circ}\text{C}$
Air inlet temperature	$25^{\circ}\text{C} < T_{\text{wb, in}} < 33^{\circ}\text{C}$

Based on the simulation results for the test matrix the two CWCs were characterised to develop a model that can be implemented in the simulation of a mine water reticulation network. Using Engineering Equation Solver V7.697 (F-Chart Software, 2007) a data fit was done to obtain coefficients $a[0] \dots a[5]$ in the following equations. The cooling capacity of CWC #1 can be determined from:

$$Q_{\text{cooling}} [\text{kW}] = 190 \left(\begin{array}{l} 0.08662 + 0.04329 \cdot T_{\text{wb, in}} - 0.04904 \cdot T_{\text{water, in}} \\ + 0.001298 \cdot T_{\text{wb, in}}^2 - 0.0008423 \cdot T_{\text{wb, in}} \cdot T_{\text{water, in}} \\ - 0.0003503 \cdot T_{\text{water, in}}^2 \end{array} \right) \quad (3-6)$$

When fitted to the experimental data the average accuracy of the cooling capacity for CWC #1 is 99.73% with a maximum error of 10.9%. Figure 3.18 illustrates the accuracy of the correlation.

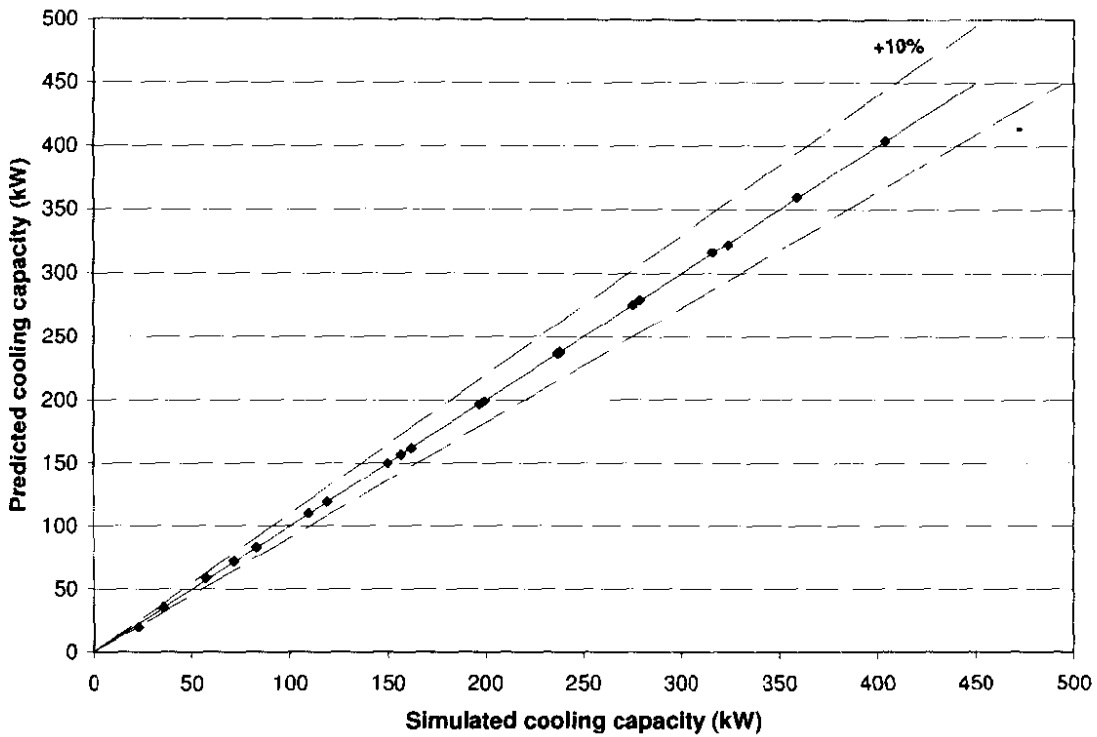


Figure 3.18: Predicted vs. simulated cooling capacity for CWC #1.

The cooling capacity of CWC #2 can be determined from:

$$Q_{cooling} [kW] = 200 \begin{pmatrix} -0.1398 + 0.06923 \cdot T_{wb,in} - 0.05846 \cdot T_{water,in} \\ +0.0007483 \cdot T_{wb,in}^2 - 0.0007488 \cdot T_{wb,in} \cdot T_{water,in} \\ -0.0001756 \cdot T_{water,in}^2 \end{pmatrix} \quad (3-7)$$

When fitted to the experimental data the average accuracy of the cooling capacity for CWC #2 is 99.65% with a maximum error of 12.3%. Figure 3.19 illustrates the accuracy of the correlation.

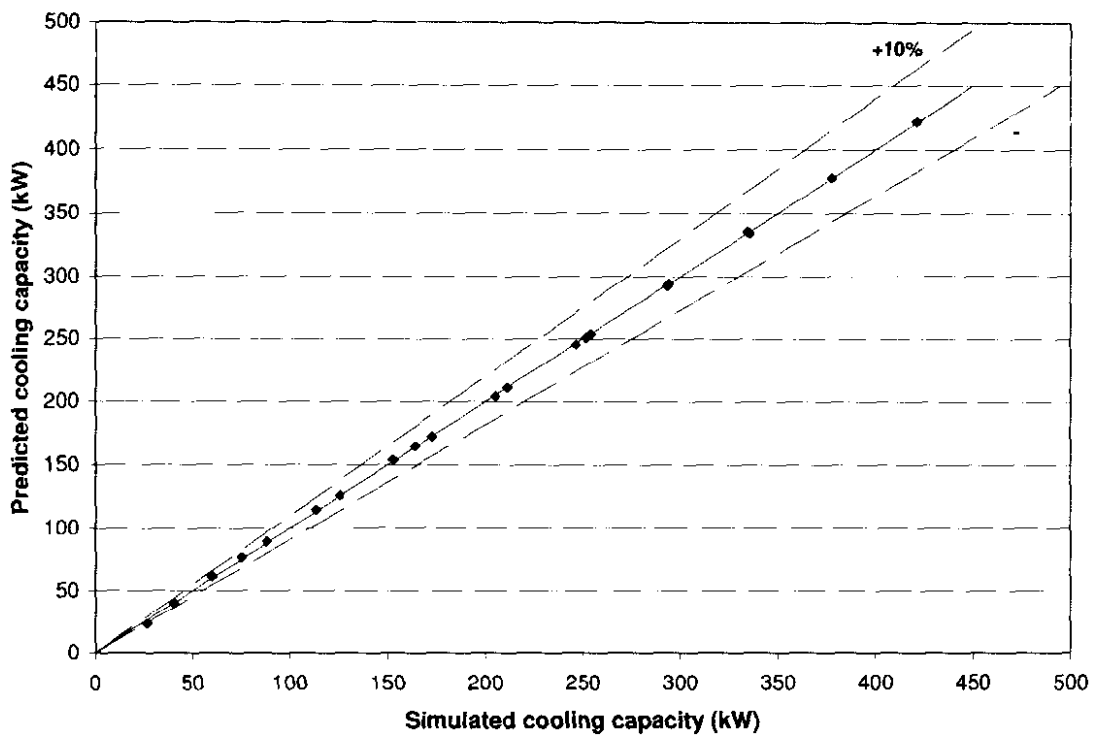


Figure 3.19: Predicted vs. simulated cooling capacity for CWC #2.

3.8. Summary

With the characterisation of the ACU as well as two different CWCs completed, the next step will be to incorporate these models in a water reticulation network simulation. In the next chapter the development of the mine system simulation will be discussed in more detail.

CHAPTER 4

MINE SYSTEM SIMULATION

CHAPTER 4

4. SYSTEM SIMULATION

To evaluate the energy efficiency potential of an ACU in a mining operation it is necessary to simulate the unit in varying underground conditions. This chapter aims to describe the generic mine system model developed for power consumption comparisons between the new ACU technology and existing CWC technology. It must be noted beforehand that the system model was not developed for mine ventilation, cooling and environmental control, like for example the software package VUMA (VUMA, 2006). It is in fact a simplified model to show the ACU potential at different depths in a mine, for both existing levels and future expansion.

The first part of this chapter will explain the workings of the generic system model listing all the assumptions and simplifications made. Thereafter the accuracy of the model will be verified using underground monitoring results from the prototype ACU. A case study will follow in Chapters 5 and 6 to evaluate the potential of the ACU technology if implemented for deep mine expansion.

4.1 Mine system model

As stated in Chapter 3 the performance of both an ACU and CWC is mainly a function of the inlet water temperature and the wet bulb temperature of the air on-coil. It is therefore essential to predict these conditions as close as possible to the actual underground conditions found in a mine section to accurately compare the ACU and CWC methodologies. The simplified underground network consists mainly of the following elements:

- Water pipes, both vertical and horizontal.
- Air elements around water pipes, both vertical and horizontal.
- ACU and CWC elements.
- Chilled water supply dams and hot water return dams.
- Pumps in return water line.
- Primary cooling of return water, either on surface or underground.

Figure 4.1 shows a typical layout of the water and air network found in a mine. For discussion purposes of the different components listed above the layout below is equipped with a surface chiller plant. The first part of this section will describe the simulation philosophy behind each of the different elements found in the system. This will be followed by an explanation of how the elements are implemented in the system simulation.

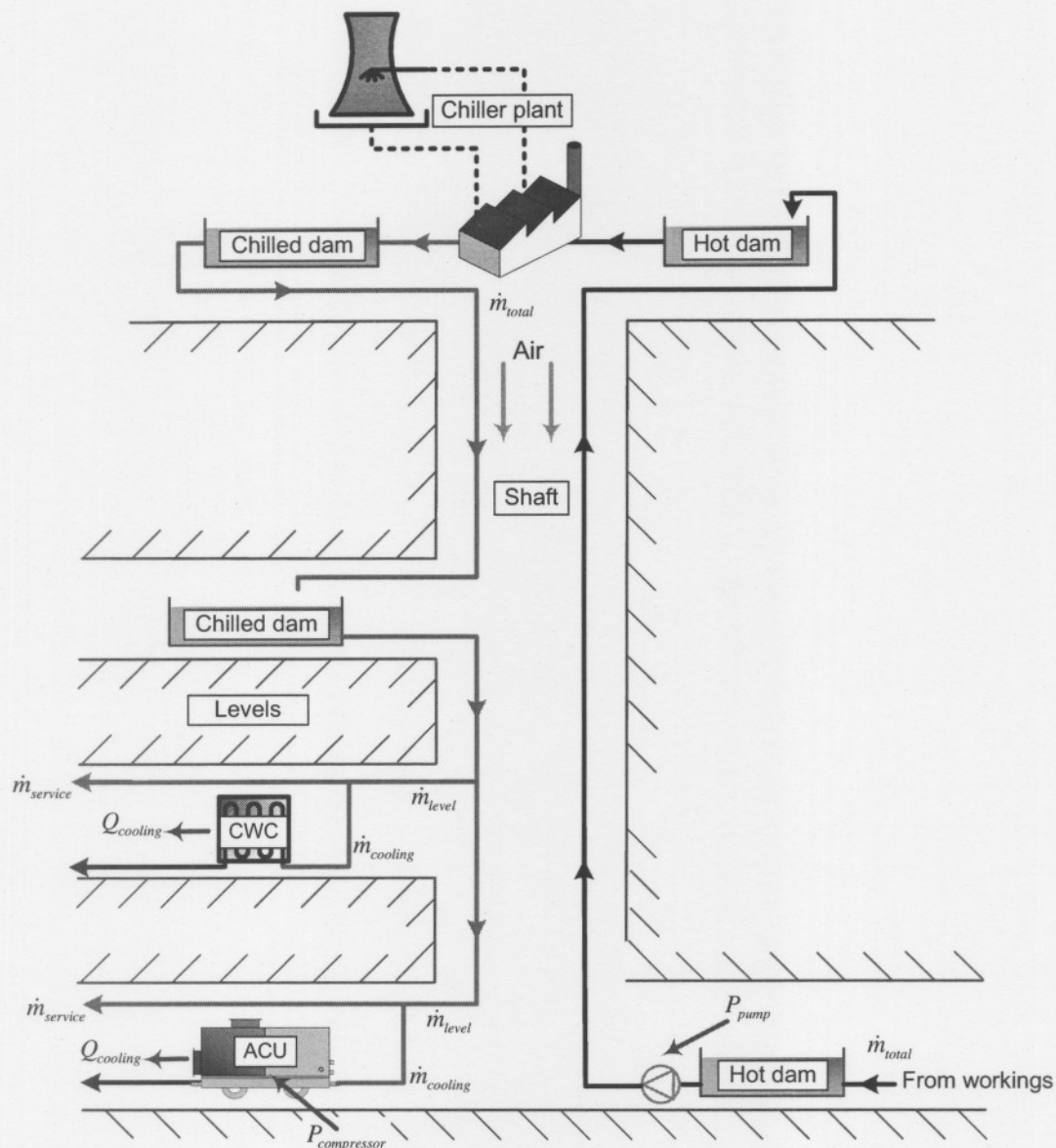


Figure 4.1: Mine system layout.

4.1.1 Pipe element

The water network consists of two types of pipe elements namely vertical and horizontal pipes. Vertical pipe elements are used to connect different levels down the shaft and horizontal pipes are used on each level. For vertical pipe elements the assumption was made that the effect of auto compression is the dominant factor. This resulted in only implementing the laws of conservation of energy and mass, and not that of momentum. It must be noted that if friction was included it would have affected the resultant pressure levels calculated, and therefore the fluid properties used in the energy equation. However, the change in fluid properties of water is negligible and therefore momentum was omitted from the simplified simulation. For horizontal pipe elements the overriding

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factor was taken as the heat transfer between the surrounding air and the water. This also resulted in applying only the laws of conservation of energy and mass.

4.1.1.1 Conservation of energy

For vertical pipes the heat transferred between the air flowing down the shaft and the water inside the pipes was also included, in addition to the auto compression effect. The conservation of energy equation used for both types of pipe elements is given by (Shames, 1992):

$$\dot{m}_{out} (h_{out} + g \cdot z_{out}) = \dot{m}_{in} (h_{in} + g \cdot z_{in}) + Q. \quad (4-1)$$

The heat transfer (Q) between the air and water is solved using the ϵ -NTU method (Incropera & DeWitt, 2002) based on the resistance circuit shown in Figure 4.2.

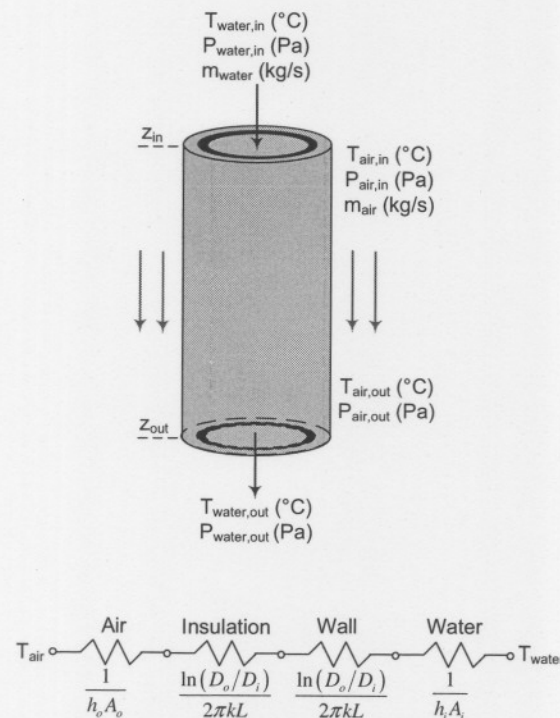


Figure 4.2: Pipe element.

The convection heat transfer coefficient of the air on the outside of the insulation was calculated using the equation proposed by Burrows (1989):

$$h_o = 0.2 \cdot \frac{k}{D_o} \cdot \text{Re}^{0.6}. \quad (4-2)$$

Equation 4.2 is generally used in the mining industry for air flow over a pipe.

The convection heat transfer coefficient of the water on the inside of the pipe is given by (Incropera & DeWitt, 2002):

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$$\text{For laminar flow, } h_i = 4.36 \cdot \frac{k}{D_i} \quad (4-3)$$

$$\text{For turbulent flow, } h_i = 0.023 \cdot \frac{k}{D_i} \cdot \text{Re}^{0.8} \cdot \text{Pr}^{0.4} \quad (4-4)$$

The flow inside the pipe system will generally be turbulent with laminar flow only occurring in remote locations where minimal mining and/or cooling takes place.

With friction neglected the pressure increase in a vertical pipe element is calculated from:

$$\Delta p = \rho \cdot g \cdot \Delta z \quad (4-5)$$

The gravitational acceleration (g) used in Equations 4.1 and 4.5 was taken as 9.79 m/s^2 for South Africa (Burrows, 1989).

For a horizontal pipe element on a level the same set of equations as listed above was used with $\Delta z = 0$.

For water flowing inside a pipe the temperature change over a section of pipe is calculated from:

$$\Delta h = c_p \cdot \Delta T \quad (4-6)$$

4.1.1.2 Conservation of mass

The conservation of mass applied to nodes where the shaft links to a level is given by:

$$(\rho \cdot V \cdot A)_{\text{shaft},i} = (\rho \cdot V \cdot A)_{\text{level},i} + (\rho \cdot V \cdot A)_{\text{shaft},i+1} \quad (4-7)$$

For pipes connecting on a level the conservation of mass is given by:

$$(\rho \cdot V \cdot A)_{\text{pipe},i} = (\rho \cdot V \cdot A)_{\text{pipe},i+1} \quad (4-8)$$

4.1.2 Air element

Since the main focus of the simulation is on the water network the following assumptions were made for the air side. Due to the complexity of air temperature changes down a mine it was decided to approach the air in the shaft and on a level in different ways.

For vertical air flowing down a shaft, measured data obtained for the specific mine under investigation is implemented. These measurements are usually taken at the stations on successive levels for volume flow rate, dry bulb and wet bulb temperatures. The main reason why measured data is used instead of attempting to simulate the changes in air temperature is due to the uncertainties involved. These uncertainties include among others:

- The shaft wall surface can be wet or dry depending on the flow of fissure water.

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- Heat flow from the surrounding rock and other mechanical equipment.
- Uncertainty about the existence and/or condition of lining material of the shaft wall.

For depths beyond the current mine configuration the assumption was made that the measured wet bulb and dry bulb temperatures can be extrapolated. This assumption was based on temperature measurements obtained from mining personnel at the case study mine shown in Figure 4.3. If however insufficient data is available the wet bulb temperatures will be extrapolated combined with a relative humidity of 90% to obtain the air properties for depths below 3500m. This assumption is based on the downcast shaft air measurements obtained. Figure 4.4 shows that with increasing depth the relative humidity approaches 90%. The pressure change for vertical air flow is also calculated with Equation 4.5.

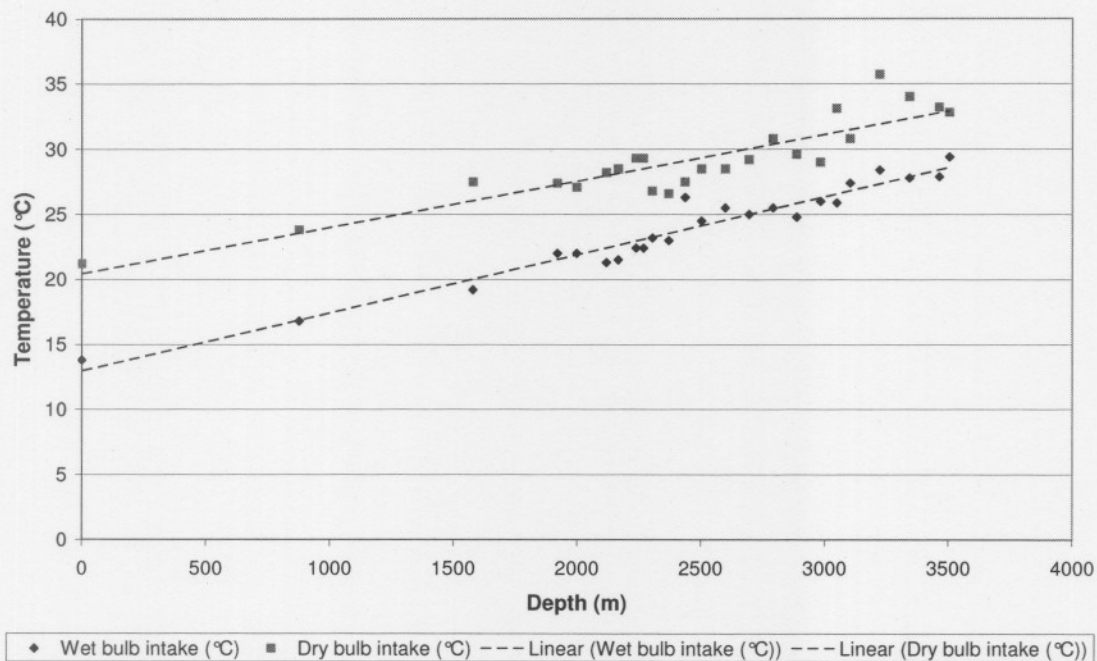


Figure 4.3: Change in temperatures with depth.

For horizontal air flow into the levels the measured conditions at the stations were used as reference point. For a section of tunnel only the effect of heat flow from the surrounding rock is taken into account. The heat flow from other sources like mechanical equipment and blasting was not taken into account. The heat transfer from rock into the ventilation air never reaches a steady state but after approximately the first six months of a new development the changes are relatively small and a quasi-steady state approach is valid. The heat flow from the rock is calculated using the technique described by Burrows (1989).

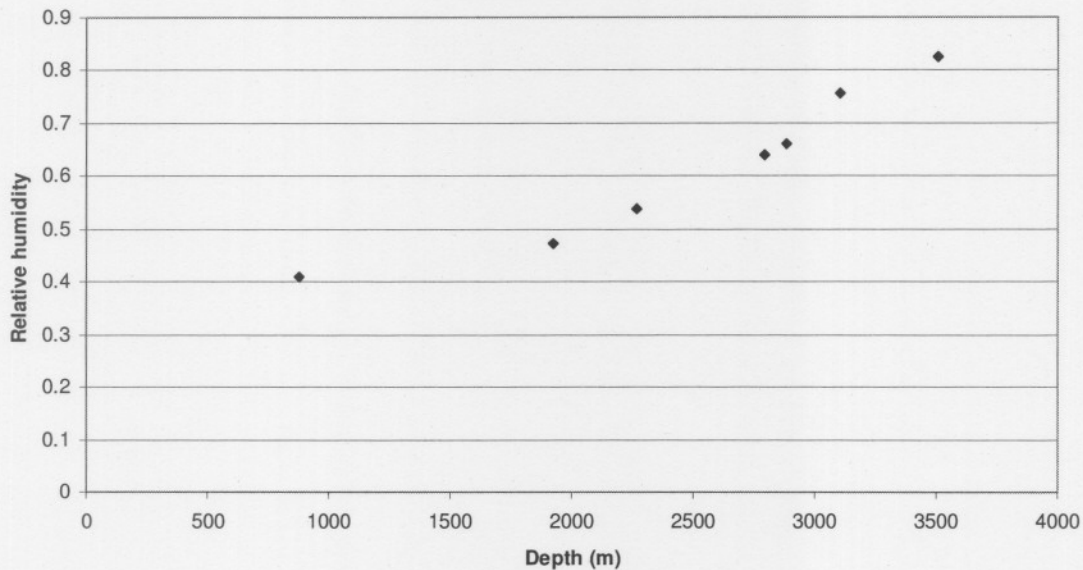


Figure 4.4: Change in relative humidity with depth.

The rate of heat flow from the rock into the tunnel per unit length is calculated from:

$$Q = 2\pi k_{rock} X (T_{VRT} - T_S). \quad (4-9)$$

T_{VRT} is the virgin rock temperature (VRT), T_S is the rock surface temperature and X is an age factor. The approximate VRT ($^{\circ}\text{C}$) for different regions in South Africa can be determined from (Patterson, 1992):

$$\begin{aligned} \text{VRT}_{\text{Witwatersrand}} &= 18 + 9 \cdot d \\ \text{VRT}_{\text{Carletonville}} &= 16 + 10.5 \cdot d \\ \text{VRT}_{\text{Klerksdorp}} &= 22 + 10.5 \cdot d \\ \text{VRT}_{\text{Free State}} &= 20 + 14.6 \cdot d \\ \text{VRT}_{\text{Bushveld}} &= 21.5 + 22 \cdot d \end{aligned} \quad (4-10)$$

where d is the vertical depth in km.

The rock surface temperature (T_S) in Equation 4.9 is generally taken as the dry bulb temperature of the air flowing over the surface. The age factor (X), taking into account the age and size of the tunnel and the properties of the rock, can be calculated from:

$$X = \frac{0.685}{F_o^{0.146}}. \quad (4-11)$$

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The Fourier number (F_o) is given by:

$$F_o = \theta \left(\frac{k}{\rho \cdot c_p} \right)_{rock} \left(\frac{D}{2} \right)^2 \quad (4-12)$$

where θ is the age in seconds and D the equivalent tunnel diameter. For the simulation the rock was taken as quartzite and working with typical property values the Fourier number reduces to:

$$F_o = 315 \left(\frac{\text{Age in years}}{D^2} \right). \quad (4-13)$$

The change in air temperature through a tunnel section is then calculated from Equation 4.1.

For horizontal air flow on a level the assumption was made that the humidity ratio w (kg_w/kg_a) stays constant for air flowing in supply tunnels. The outlet dry bulb and wet bulb temperatures of a section can easily be obtained as a function of enthalpy and humidity ratio. The humidity ratio assumption was based on temperature measurements obtained from mining personnel at the case study mine for a mining level approximately 3350m below surface. The calculated humidity ratios shown in Figure 4.5 have an average value of 0.0237 with a standard deviation of 0.002, thus confirming that the assumption made is accurate enough for the simplified approach.

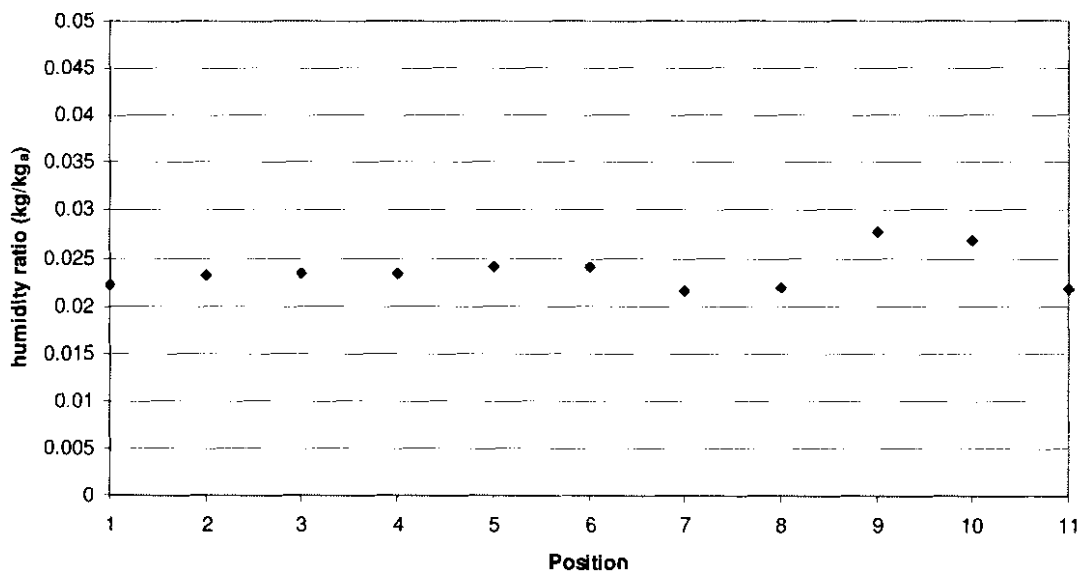


Figure 4.5: Calculated humidity ratios at different locations.

4.1.3 ACU element

The characterisation of the ACU, as described in Chapter 3, was based on laboratory testing of the unit at a barometric pressure of approximately 86.5kPa. Due to varying barometric conditions with

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depth the correlations derived cannot be implemented as is when predicting underground performance. The reason for this is that the correlations derived in Chapter 3 are based on a measured wet bulb temperature change over the evaporator at a fixed barometric pressure. For different barometric pressures the wet bulb temperature change will vary for the same amount of cooling.

To accommodate the correlations the following logic was implemented in the system model. For any underground level the inlet air wet bulb temperature and mass flow rate to the evaporator are used as input to the ACU model containing the correlations derived in Chapter 3. This model then calculates the cooling capacity of the unit as if the air inlet wet bulb temperature was at a barometric pressure of 86.5kPa. The calculated cooling capacity is then used with the correct underground barometric conditions to determine the real off-coil air conditions.

4.1.4 CWC element

The cooling capacity correlations derived in Chapter 3 for the two different CWCs were at a barometric pressure of 120kPa. The cooling capacity correlations are implemented as is in the system simulation.

4.1.5 Chilled and hot dams

As was shown in Figure 4.1 there is a chilled water dam in the supply line before the working levels and a hot dam at the bottom of the mine to collect warm waste water from operations. Both dams were simplified to only an energy balance of the in- and outflow of water.

$$\sum_{i=1}^n \dot{m}_{in,i} \cdot T_{in,i} = \sum_{j=1}^n \dot{m}_{out,j} \cdot T_{out,j} \quad (4-14)$$

4.1.6 Pump element

The water consumption of either the ACU or CWC has an effect on the pumping power required to return the waste water to the primary cooling installation. For the pumping the assumption was made that the potential energy is the dominating factor, thus neglecting the effects of friction and secondary losses in the system model. The pumping power required is calculated from:

$$P_{pump} = \frac{g \cdot \Delta z \cdot \dot{m}}{\eta} \quad (4-15)$$

The pump efficiency (η) was taken as 0.75 based on information obtained from the case study mine regarding the type of pumps currently in use.

4.1.7 Primary cooling element

The service water used for underground cooling is usually cooled by a primary cooling installation. The primary cooler cools the return water from underground operations back down to the required supply

temperature before the water is pumped into the mine again. The primary cooling is done by one of several methods of which water chilling plants and cooling towers are the most widely used.

4.1.7.1 Surface chiller plant

A large portion of South African mines are equipped with surface chiller plants as was explained in Chapter 2. These plants usually supply water at a temperature between 5°C and 10°C into the mine. The required cooling capacity is determined from the total water mass flow and temperature to the mine as well as the return water temperature from the mine:

$$Q_{cooling} = \dot{m}_{water, total} \cdot c_p \cdot (T_{water, return} - T_{water, supply}). \quad (4-16)$$

The required power input is calculated using the COP of either a typical surface chiller plant or that of the unit installed at the specific mine under investigation:

$$P_{compressor} = \frac{Q_{cooling}}{COP_{cooling}}. \quad (4-17)$$

On the condenser side the plant is usually equipped with a cooling tower to reject heat to the atmosphere. The cooling tower capacity is calculated from:

$$Q_{cooling\ tower} = Q_{cooling} + 0.9 \cdot P_{compressor}. \quad (4-18)$$

Another important factor to evaluate is the amount of water evaporated during the heat rejection process. This has an effect on the total running cost of the plant. The evaporated water is determined from:

$$Q_{cooling\ tower} = \dot{m}_{evaporated} \cdot h_{fg}. \quad (4-19)$$

4.1.7.2 Underground chiller plant

Underground water chiller plants are getting more attention these days due to mining operations expanding to greater depths. Many of these units consist of a water chiller plant with a condenser cooling tower for heat rejection into the return airway. This air is then directed to surface via the ventilation shaft using the main fans. As stated in Chapter 2 the biggest problem is heat rejection into already warm air resulting in a reduced COP compared to surface plants. The cooling capacity of an underground plant can be calculated using Equation 4.16.

The required power input is calculated using the COP of either a typical underground chiller plant or that of the unit installed at the specific mine under investigation. The power input is calculated from Equation 4.17. The cooling tower capacity and amount of water evaporated is calculated using Equations 4.18 and 4.19.

4.1.8 System simulation philosophy

The system simulation was implemented in the software Engineering Equation Solver (EES). The system can be divided into two main sub-networks namely the supply network and the return network. The philosophy behind each of the sub-networks will be discussed in more detail below.

4.1.8.1 Supply network

The supply network consists of all the elements connecting the primary cooling plant with an ACU or CWC. In short the supply side consists of vertical and horizontal pipes connected to form a distribution network down the shaft into different levels via a chilled water dam. The change in air temperatures, explained in Section 4.1.2, is also taken into account down the shaft and into a level up to the position where the cooling is required. The water and air supply conditions are then used as input conditions to the ACU or CWC model.

The following are used as inputs to the supply network.

- Physical mine information.
 - The physical location in South Africa from which the surface barometric pressure is determined.
 - Depth below surface of each mine level for both existing levels and future developments.
 - Shaft dimensions.
 - Dimensions of water pipes in the shaft and on levels.
 - Depths of chilled water dams and hot dams.
- Water-side information.
 - The supply temperature from the primary cooling plant into the mine.
 - The service water flow rate to each level. Most mines are currently equipped with monitoring systems measuring the water consumption on all levels.
- Air-side information.
 - The supply temperature of the air into the mine and at the entrances to each level, both dry bulb and wet bulb.
 - The air volume flow rate to each level.
- Cooling information.
 - Position and type of primary cooling plant.
 - Distance into each level where an ACU or CWC should be located.
 - The required amount of cooling for different positions per level.

The total water supply into the mine from the primary cooling plant is the sum of the service water to each level and the cooling water consumption of the ACUs or CWCs on each level.

$$\dot{m}_{water, total} = \sum \dot{m}_{level} \quad (4-20)$$

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where

$$\dot{m}_{level} = \sum \dot{m}_{service} + \sum \dot{m}_{cooling} \quad (4-21)$$

The reason for including the service water in the network simulation and not only the chilled water consumption of the coolers is to obtain realistic flow rates and temperature distributions through the mine for the actual pipe sizes used.

One of the inputs to the simulation is the amount of cooling required at certain distances on different levels. The required cooling is used to determine how many ACUs or CWCs need to be placed in an area based on the performance of the units at the local ambient conditions. This has an effect on the amount of cooling water supplied to each level and therefore the total supply to the mine.

The most important outputs from the supply network include:

- Water mass flow rate to each level for cooling.
- Water temperature distribution down the shaft and into each level.
- Heat losses/gains to or from the surrounding air for a section of pipe.
- Outlet water temperature of a CWC. For ACUs the outlet temperature is fixed at 45°C as explained in Chapter 3.
- Air off-coil conditions for either an ACU or CWC.
- Actual cooling capacity of the ACUs or CWCs on different levels.
- Power input of ACUs on each level.
- Number of ACUs or CWCs needed to satisfy the cooling load requirements at different positions.

4.1.8.2 Return network

The typical South African mine uses the main shaft for the supply and return water pipelines as shown in Figure 4.1. Therefore, all of the physical shaft inputs used for the supply side are applicable for the return network with only the dimensions of the return pipes specified separately.

In actual mining operations the water returning from different levels is collected in drains and then either pumped or gravity fed to hot water dams from where the water is pumped out of the mine or back to the primary cooling plant. To simplify this, the return network consists of only one hot dam on the lowest level of the mine with all the waste water accumulating there before being pumped back to the primary cooling plant. The pumping power is calculated using Equation 4.15.

Because the water return network is in the same shaft as the supply network the shaft air conditions are again applied to determine heat transfer between the down cast air and return water. Based on the return water temperature reaching the primary cooling plant and the desired supply temperature, the required cooling capacity can be determined. This is then used to determine the other parameters of primary cooling plant listed in Section 4.1.7.

The most important outputs from the return network include:

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- Pumping power required from hot dam to primary cooling installation.
- Heat losses/gains to or from the surrounding air for a section of pipe.
- Return water temperature reaching the primary cooling plant.
- Cooling requirements of the primary cooling plant.
- Power input of the primary cooling plant.
- Water evaporated in the condenser cooling tower of the primary plant.

4.1.8.3 *Economic evaluation*

To determine the viability of an ACU cooling configuration compared to a CWC cooling configuration it is necessary to evaluate the life cycle cost of the configurations. The following is a list of aspects needed for a proper evaluation:

- Capital cost of an ACU and CWC.
- Capital cost of a new primary cooling system (surface or underground).
- Power required for pumping water to the primary installation.
- Power consumption of an ACU.
- Power consumption of the primary cooling system.
- Cost per litre of water evaporated in cooling towers.
- Running cost per year per MW cooling.
- Total capital investment per MW cooling.

The majority of mines in South Africa are on an electricity tariff known as Megaflex (Eskom, 2006b).

The main characteristics of this tariff are:

- Seasonally and time differentiated active energy charges.
- Peak, standard and off-peak time periods.
- A network access charge during all time periods of R5.91/kVA excl. VAT.
- A network demand charge during peak and standard periods of R6.69/kVA excl. VAT.

The active energy charge used for calculations is given in Table 4-1.

Table 4-1: Active energy charge (excl. VAT).

High-demand season (June – August)	Time Period	Low-demand season (September – May)
52.22c/kWh	Peak	14.82c/kWh
13.81c/kWh	Standard	9.20c/kWh
7.51c/kWh	Off-peak	6.52c/kWh

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The time periods are defined as follows:

Table 4-2: Defined time periods.

	Weekdays	Saturday	Sunday
Peak	07:00 – 10:00 18:00 – 20:00	N/A	N/A
Standard	06:00 – 07:00 10:00 – 18:00 20:00 – 22:00	07:00 – 12:00 18:00 – 20:00	N/A
Off-peak	22:00 – 06:00	12:00 – 18:00 20:00 – 07:00	All hours

Other costs used for the economic evaluation are listed in Table 4-3.

Table 4-3: Costs used for economic evaluation.

Description	Value (ex VAT)
ACU capital cost per unit	R260,000-00
CWC#1 capital cost per unit	R200,000-00
CWC#2 capital cost per unit	R130,000-00
Surface water chiller plant per kW cooling	R3,390-00/kW
Surface cooling tower per kW cooling	R2,300-00/kW
Underground water chiller plant per kW cooling	R6,215-00/kW
Cost per kilolitre of water purchased	R5-00/kl
Interest rate	6.5%

To do a proper evaluation between ACUs and CWCs all comparisons are related back to a “per MW effective cooling” approach at a specific location underground.

Because the purchase of cooling equipment is usually seen as a long term investment there is usually a point where one option becomes more viable than another. For the purpose of this study the evaluation will be done for a period of one to nine years as the life expectancy of a CWC is usually six to eight years based on information from the case study mine. The evaluation is done in terms of the simple present value (*SPV*) calculated as follows (Burrows, 1989):

$$SPV = (\text{Initial capital investment}) + (\text{Yearly running cost}) \left(\frac{1 - \frac{1}{(1+i)^n}}{i} \right) \quad (4-22)$$

where n is the time in years and i is the compound interest rate as a decimal. The initial capital investment differs for different cooling configurations selected. The different configurations for possible deep mining will be discussed and evaluated in a case study in Chapter 5. The running cost consists

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of the power consumption of the primary cooling plant, the pumping power, the power consumption of the ACUs (if applicable), and the cost per kilolitre of water evaporated in cooling towers.

4.2 Verification of the generic model

After development and manufacturing the prototype ACU was in operation since November 2003. For the first eighteen months the unit was only operational for short periods in time due to additional laboratory testing and refinement of the design. Since July 2005 the unit has been fully operational in the case study mine. The unit is located on a level approximately 3225m below surface and about 3250m into the level. The ACU is used in a development area and was at one stage about 40m from the blasting face.

The shaft downcast air conditions to each mining level, shown in Figure 4.3, were obtained for a time period when the ACU was also operational. A trend line was plotted for the dry bulb temperature data with depth and the same was done for the wet bulb temperature data. The dry bulb temperature at any depth in the shaft can be obtained from:

$$T_{db} [^{\circ}C] = 0.0036 \cdot Depth + 20.418. \quad (4-23)$$

The wet bulb temperature at any shaft depth can be calculated from:

$$T_{wb} [^{\circ}C] = 0.0045 \cdot Depth + 12.962. \quad (4-24)$$

The surface water inlet temperature into the shaft was taken as 11 °C based on data for the same period as the down cast measurements. Implementing the model explained in Section 4.1 the water supplied into the ACU is predicted as 21.4 °C. The actual measurements from data loggers in the ACU showed an average inlet water temperature of 22.1 °C with a standard deviation of 1.95 °C. The actual temperature is a function of the surface inlet water temperature that varies typically between 10 °C and 12 °C. Compared to the measured average inlet water temperature the model predicts the temperature with an accuracy of 96.8%.

On the airside Equations 4.23 and 4.24 are used in the model to determine the inlet conditions to the level housing the ACU. From the shaft to where the ACU is positioned the model incorporates heat transfer between the surrounding wall and the air as explained in Section 4.1. The model predicts the inlet to the ACU as 29.8 °C_{wb}/41.8 °C_{db} compared to measured conditions of 31.0 °C_{wb}/41.0 °C_{db}. The model under-predicts the wet bulb temperature by 3.9% and over-predicts the dry bulb temperature by 2%. A possible explanation for this is that the model makes use of Equations 4.23 and 4.24 which are based on measurements taken by ventilation personnel typically equipped with a Sling Psychrometer. For these psychrometers a measuring error of up to 0.5 °C is possible which has a significant effect on the model prediction.

Keeping in mind that the main purpose of the generic system model is to compare the ACU technology with conventional CWC technologies, the accuracies listed above is deemed acceptable. If a detailed ventilation analysis is required a software package like VUMA should be considered.

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4.3 Summary

This chapter discussed the methodology behind the mine system simulation developed to evaluate the energy efficiency and DSM potential of the ACU. In the next chapter a case study mine will be used as basis to show the potential of the ACU technology when compared to existing CWC technologies for potential deep mine expansion.

CHAPTER 5

CASE STUDY - TECHNICAL ANALYSIS

CHAPTER 5

5. CASE STUDY - TECHNICAL ANALYSIS

With deepening of South Africa's mines being the current hot topic of discussion it is necessary to evaluate the contributions the ACU technology may make to this process. To demonstrate the potential a case study mine was selected and simulated for depths extending beyond current mining operations. For these depths a few potential cooling configurations involving ACUs and CWCs will be compared for amongst others the total power requirement of each option. In the next chapter this will be extended to include an economic analysis of these configurations. This will ultimately result in an assessment of the cost effectiveness as well as the energy efficiency and DSM potential of the ACU technology.

The first part of this chapter will explain the different deep mine cooling configurations selected for the evaluation. This will be followed by a detail technical comparison of the results for the different configurations.

5.1 Mine cooling configurations

5.1.1 Existing configuration

Before the different deep mine configurations are evaluated it is necessary to take a look at the existing mine water configuration that will be taken as the baseline from where the proposed configurations can be evaluated. A simplified layout of the chilled water reticulation network is shown in Figure 5.1. The chilled water network consists of a surface water chiller plant cooling the return water from the mine and supplying it back down again via a surface chilled water dam. The chilled water flows down the mine via interim dams and is finally supplied to the workings on the different levels. The hot waste water from the workings is gathered in a so-called hot dam at the bottom of the mine from where it is pumped back to surface via interim hot dams to be cooled again by the surface refrigeration installation.

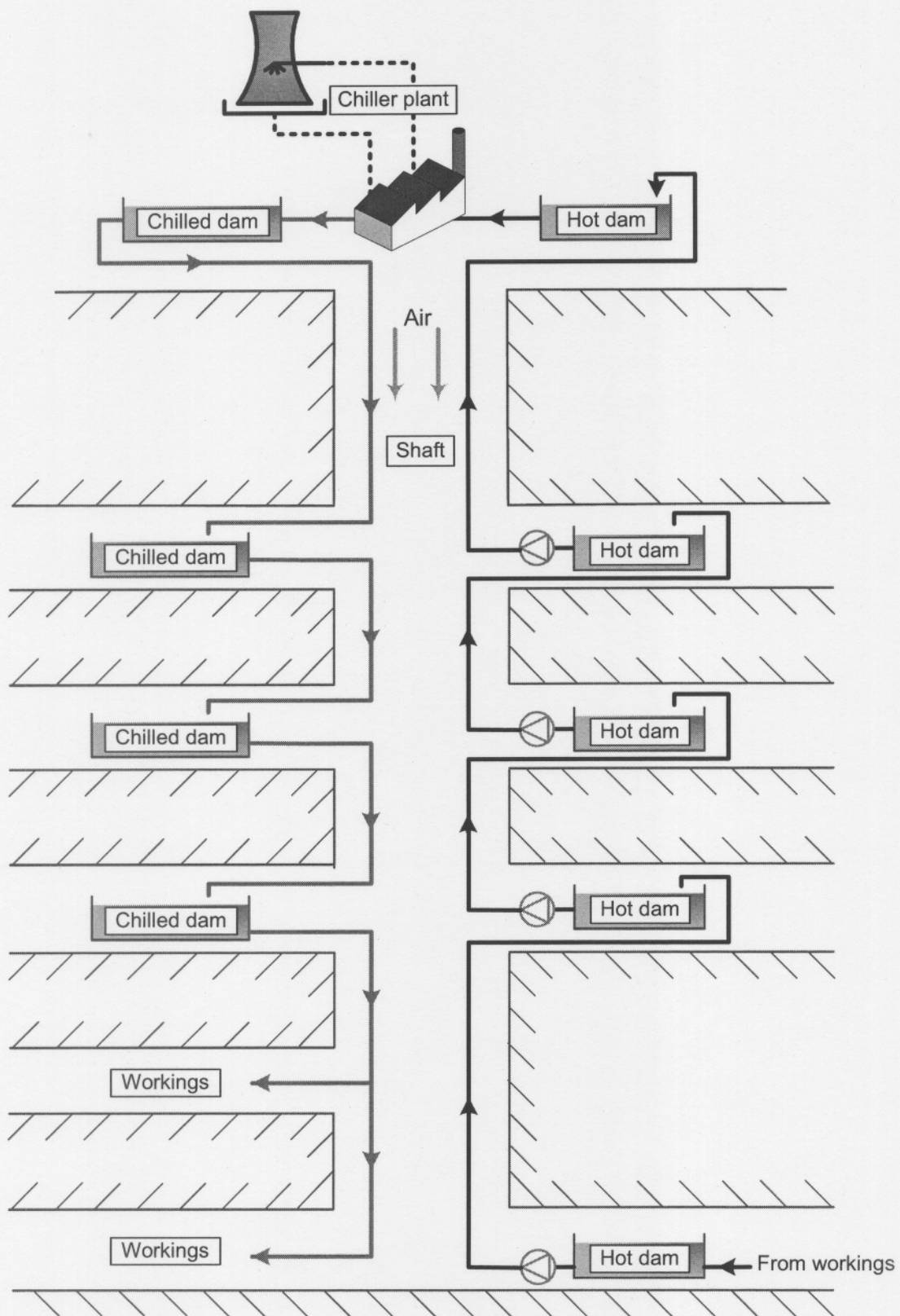


Figure 5.1: Existing mine cooling configuration.

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5.1.2 Deep mine cooling configurations

In order to weigh up the ACU technology with the conventional CWC technology the following main configurations will be compared for deep mining, using the existing configuration of Figure 5.1 as baseline from where the development is done:

- Surface water chiller plant with either ACUs or CWCs.
- Underground water chiller plant with CWCs.
- Underground hot dam with either ACUs or CWCs.

The first method of supplying chilled water to the new levels entails the installation of an additional surface chiller plant or expanding the existing plant. This chilled water can be supplied to either ACUs or CWCs as shown in Figure 5.2 using the existing reticulation network expanded to the additional new levels. For effective comparison this scenario will be simulated for two different chilled water supply temperatures namely 5°C and 10°C. The reason for this is that the design specifications obtained for the case study mine chiller plant states a delivery temperature of 5°C. However in practice measurements show that the plant often operates at temperatures in the region of 10°C. This increase in delivery temperature has an effect on the water mass flow rate through the network affecting the pumping requirements.

The next configuration to be evaluated consists of an underground water chiller plant supplying chilled water to CWCs on the deeper levels as illustrated in Figure 5.3. These underground plants are usually equipped with condenser cooling towers rejecting heat into the RAW. This air is then directed to surface using the main fans in the ventilation shaft. The reason why an underground chiller plant combined with ACUs are not considered is because the ACU is in fact a self-contained underground chiller which is an alternative to a fixed underground facility. As with the surface chiller plant configuration of Figure 5.2 evaluations will be done for water supply temperatures of 5°C and 10°C.

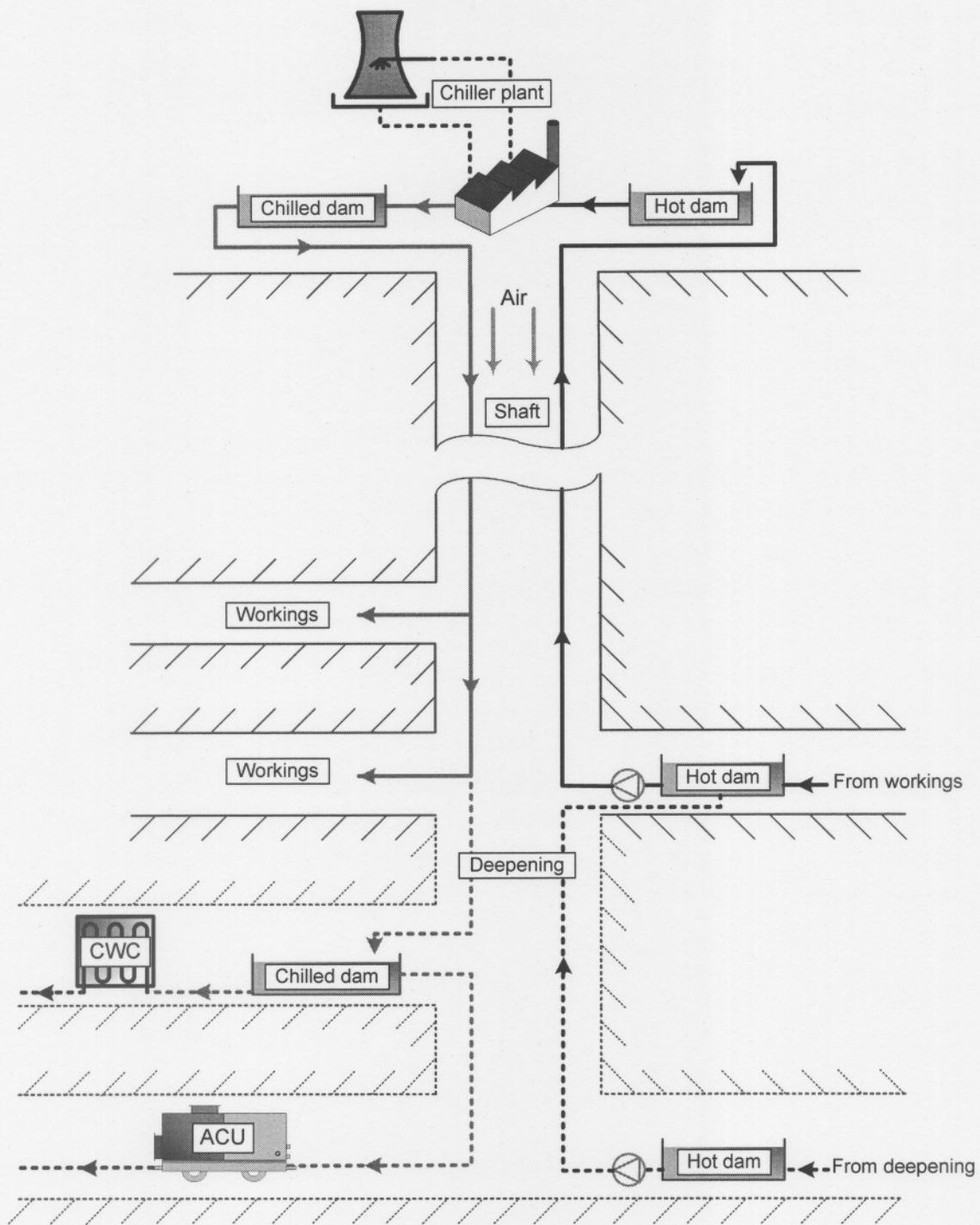


Figure 5.2: Surface chiller plant with ACUs or CWCs.

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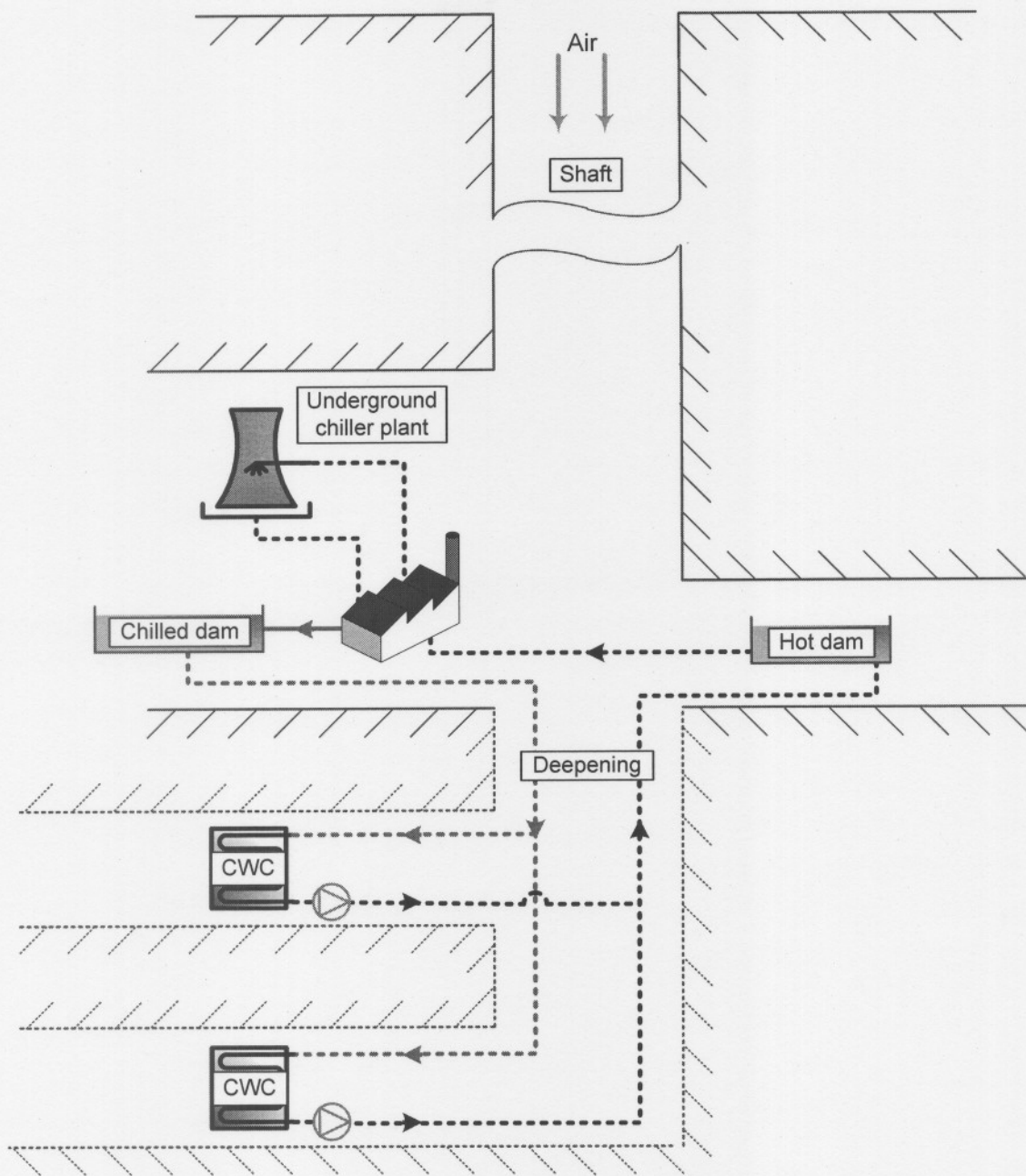


Figure 5.3: Underground chiller plant with CWCs.

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The final main configuration to be evaluated is the use of an underground hot dam to supply water to ACUs or CWCs on deeper levels as shown in Figure 5.4. The philosophy behind the underground dam configuration is to cycle water from the dam through the ACUs or CWCs and back via a collector hot dam. The hot water dam used as water source accumulates the waste service water from the existing mining operation before being pumped out of the mine to be chilled again. The hot dam used is located on the last level of current mining operations.

Later on in this chapter it will become evident that this option is only viable combined with ACUs. The reason why CWCs were included is to show the effect that the high water temperatures have on the performance of these units and the resultant effect on the reticulation network.

Due to the outlet water temperature of the ACUs being 45°C this will imply that over time the hot dam will start to increase in temperature. In practice this increase should be minimal due to the large volume of these underground dams compared to the flow rates through the ACUs. However, an increase in dam temperature will result in the water, when pumped to surface, arriving warmer than normal resulting in a larger cooling capacity required to cool the water again. This effect will be taken into account during the simulations.

A modification of the final main configuration is shown in Figure 5.5. To cater for the increase in water temperature that results in a need to increase the surface chiller plant capacity, the effect of using a natural draught pre-cooling tower on surface will be investigated. The purpose of the cooling tower is to extract the additional energy added by the ACU network before the water flows to the existing water chiller plant. Since a cooling tower is considerably less expensive than a surface chiller plant, this modification should have a positive effect on the capital cost of the installation.

In the next section these different scenarios will be analysed and compared to determine the effect they have on the mining operation.

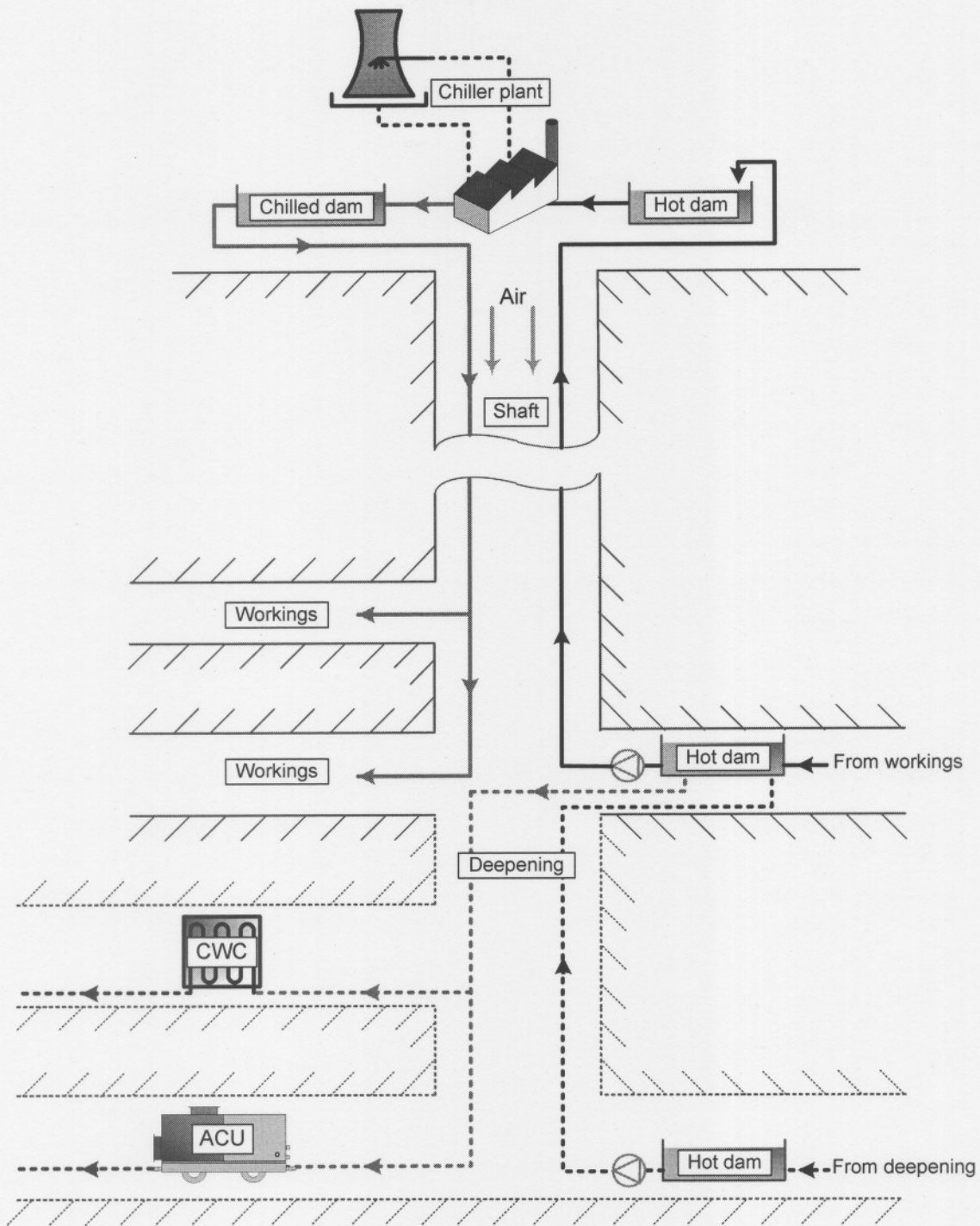


Figure 5.4: Underground hot dam with ACUs or CWCs.

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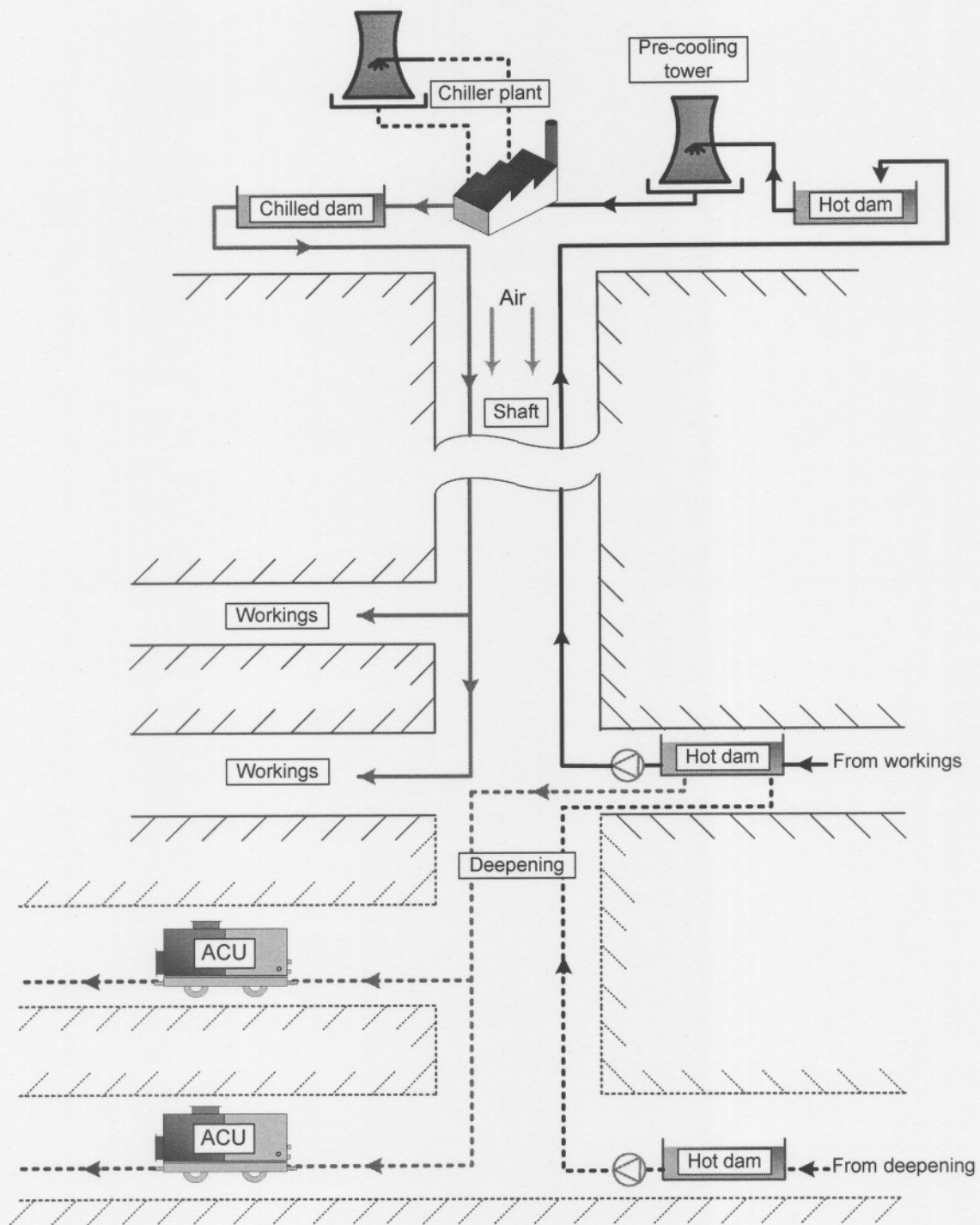


Figure 5.5: Additional pre-cooling tower.

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5.2 Configuration analysis

To conduct a proper evaluation of the deep mining configurations listed in Section 5.1 a comparison will be done for depths from 4000m to 5000m at 500m increments. On each level a comparison will be done for 1MW of effective cooling spread over a distance of 500m.

5.2.1 Existing mine cooling configuration

Figure 5.6 shows the calculated heat loads of the existing mine configuration for a supply temperature of 5°C . The supply side containing a number of chilled water dams, as shown in Figure 5.1, was simplified by neglecting the interim water dams and working with only the final dam before the water goes to the workings. On the return side the network was also simplified by only keeping the hot water dam on the lowest level and theoretically pumping water straight from there to surface.

Based on measured data obtained from the case study mine the total chilled water flow rate to the different levels below the final chilled water dam was on average 433 kg/s for the period during which the measurements were made. It should be noted that there are currently limited operations on the levels above the final chilled dam with mine data logs showing zero flow to these levels. The flow rate of 433 kg/s was kept fixed for the existing mine network and used throughout in the deep mine simulations to follow.

An assumption that was made is that the water returning from the workings will end up in the hot water dam on the lowest level to be pumped out of the mine from there. The temperature of the waste water arriving at the dam was taken as 28°C . Due to limited information on the water temperature in the hot dam this assumption was based on observations gained from a commercial pump scheduling project at another mine with a similar setup.

As discussed in Chapter 4, on the supply side the effects of auto compression and heat transfer between the water and the surrounding air are taken into account. The heat gains shown in Figure 5.6 represent the heat transfer component. On the return side the effects of auto compression, heat transfer and pump work required to return the water to surface were taken into account.

The mine load indicated in Figure 5.6 between the chilled dam and the hot dam is the summation of the heat loads per level. The heat load on each level is a function of the supply temperature and flow rate to the level with a fixed return temperature of 28°C .

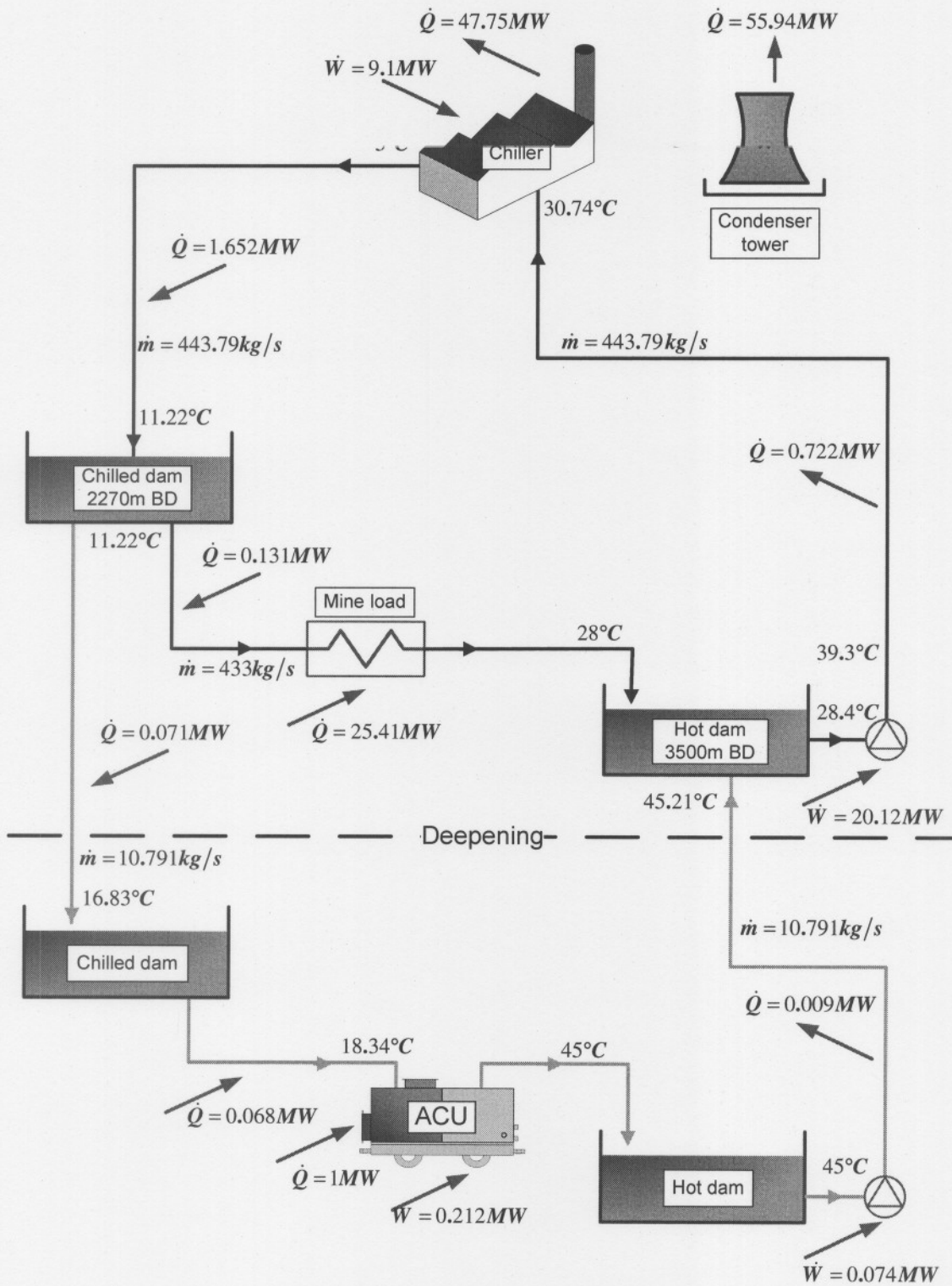


Figure 5.7: Surface chiller plant with ACUs.

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What is evident from Figure 5.7 is that for every 1MW of air cooling done by the ACUs the surface plant needs to do about 1.9MW of additional cooling to keep the supply temperature at 5°C. This additional heat load consists mainly of the ACU condenser heat dissipated into the water and the additional pumping power returning the water to surface. Working with a COP of 5.25 as obtained from the design specifications for the surface chiller plant, the additional compressor input power is calculated as 0.367MW. This results in a total additional input power of 1.143MW per 1MW of cooling. The total input power consists of the chiller plant compressors, pumping power and ACU compressors.

Figure 5.8 illustrates the surface chiller configuration equipped with type #1 CWCs for a depth of 4000m. For each 1MW of air cooling done by the CWCs the surface plant needs to do about 3MW of additional cooling to keep the supply temperature at 5°C. This is mainly due to the 1MW transferred by the CWC into the water and the additional pumping power returning the water to surface. The additional surface chiller plant compressor input power required is calculated as 0.59MW, resulting in an added total input power of 2.34MW per 1MW of cooling. For the CWC configuration the total input power consists of the chiller plant compressors and pumping power.

When Figure 5.8 is compared to Figure 5.7 it is evident that the ACU configuration circulates approximately one third of the water used by the CWCs resulting in less pumping power required. Even with the power input from the ACU compressors added, the total additional power consumed by the ACU configuration is close to 50% of the CWC configuration.

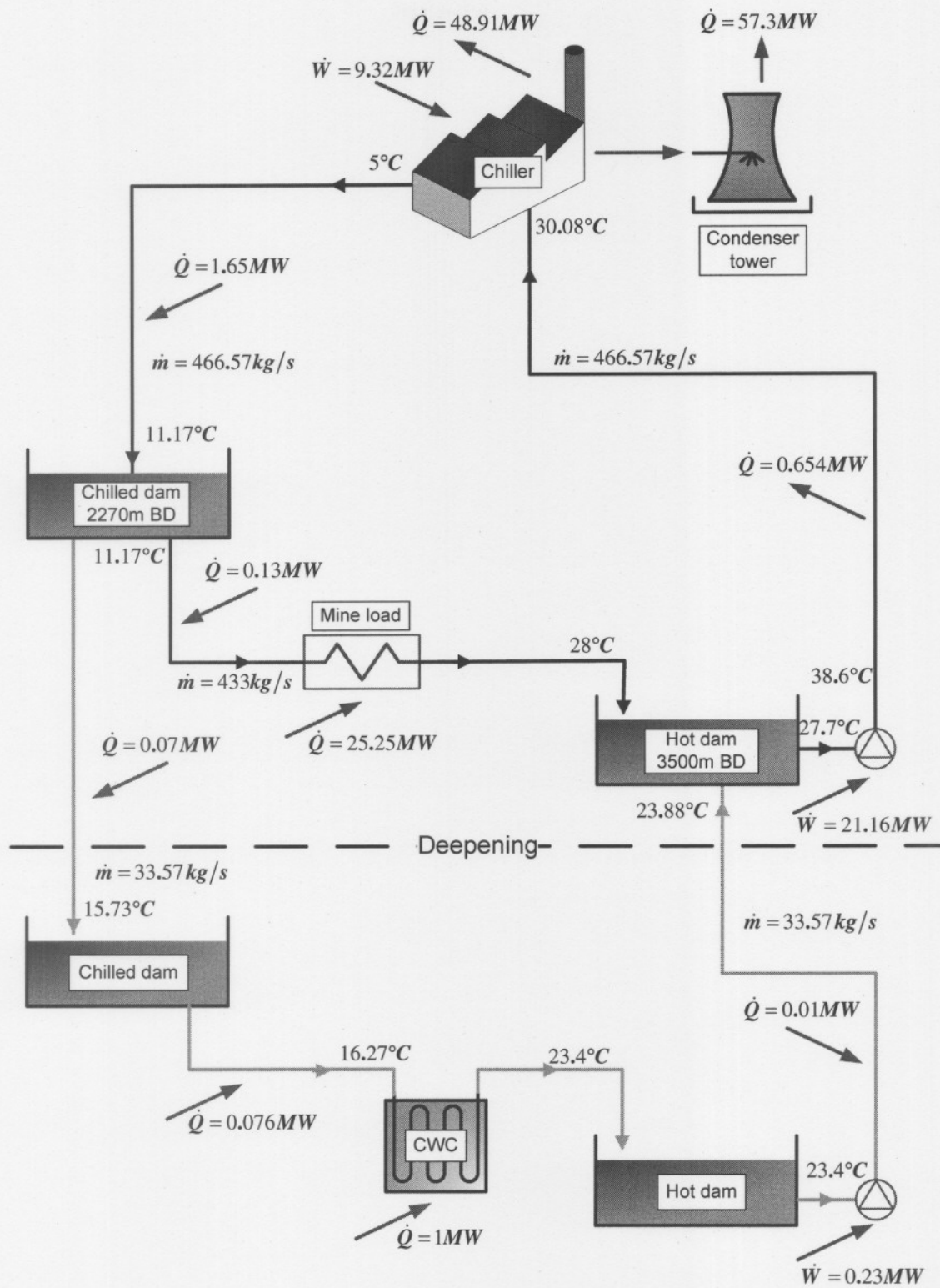


Figure 5.8: Surface chiller plant with CWCs.

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Figure 5.7 and Figure 5.8 showed illustrations of the results for two of the surface chiller plant configurations. Table 5-1 to Table 5-3 lists the results per depth for the different configurations consisting of the ACU, CWC#1 and CWC#2 options. Comparisons are listed for both 5°C and 10°C surface chiller supply temperatures. The results shown are only the incremental effects i.e. excluding the existing mine network.

Table 5-1: Comparison per 1MW cooling 4000m below depth for a surface chiller configuration.

Description	ACU (5°C)	ACU (10°C)	CWC #1 (5°C)	CWC #1 (10°C)	CWC #2 (5°C)	CWC #2 (10°C)
Pumping power (MW)	0.563	0.656	1.753	2.401	1.178	1.610
Chiller power (MW)	0.367	0.377	0.589	0.703	0.466	0.540
ACU power (MW)	0.212	0.212	-	-	-	-
Total input power (MW)	1.143	1.245	2.341	3.104	1.643	2.151
Water mass flow rate (kg/s)	10.79	12.56	33.57	45.99	22.55	30.85
Surface cooling (MW)	1.929	1.979	3.090	2.837	2.445	2.837
Cooling tower water (l/s/MW)	0.941	0.966	1.508	1.800	1.193	1.384
No of cooling units required	12	12	4	6	4	6

Table 5-2: Comparison per 1MW cooling 4500m below depth for a surface chiller configuration.

Description	ACU (5°C)	ACU (10°C)	CWC #1 (5°C)	CWC #1 (10°C)	CWC #2 (5°C)	CWC #2 (10°C)
Pumping power (MW)	0.678	0.793	1.815	2.419	1.242	1.654
Chiller power (MW)	0.395	0.408	0.603	0.707	0.482	0.551
ACU power (MW)	0.208	0.208	-	-	-	-
Total input power (MW)	1.128	1.409	2.418	3.126	1.724	2.205
Water mass flow rate (kg/s)	11.55	13.50	30.90	41.19	21.149	28.15
Surface cooling (MW)	2.074	2.141	3.168	3.711	2.531	2.892
Cooling tower water (l/s/MW)	1.012	1.044	1.546	1.811	1.235	1.411
No of cooling units required	12	12	4	5	4	5

Table 5-3: Comparison per 1MW cooling 5000m below depth for a surface chiller configuration.

Description	ACU (5°C)	ACU (10°C)	CWC #1 (5°C)	CWC #1 (10°C)	CWC #2 (5°C)	CWC #2 (10°C)
Pumping power (MW)	0.810	0.953	1.871	2.436	1.307	1.700
Chiller power (MW)	0.427	0.444	0.618	0.713	0.500	0.563
ACU power (MW)	0.208	0.208	-	-	-	-
Total input power (MW)	1.446	1.606	2.490	3.149	1.807	2.263
Water mass flow rate (kg/s)	12.41	14.60	28.67	37.33	20.03	26.04
Surface cooling (MW)	2.244	2.333	3.247	3.741	2.622	2.956
Cooling tower water (l/s/MW)	1.095	1.138	1.585	1.825	1.280	1.442
No of cooling units required	12	12	4	5	4	5

From the tables and Figure 5.9 it is clear that ACUs combined with a surface chiller plant results in a reduction in total input power per MW of cooling compared to the other options. For a depth of 4000m combined with a 5°C supply temperature there is a reduction of 51% in the additional power requirement if ACUs are used instead of CWCs#1. A reduction of 30% is realised compared to a CWC#2 configuration. The reason for the lower reduction is due to a CWC#2 only consuming 6.1l/s

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compared to 9.0l/s for a CWC#1. For a 10°C supply temperature there is a reduction of 60% in the power requirement if ACUs are used instead of CWCs #1 and a reduction of 42% compared to a CWC#2 configuration.

The reason behind the steeper increase in total power input with depth of the ACU configurations is due to an increase in water flow as a result of an increase in inlet water temperatures with depth. On the other hand the CWCs have a constant flow rate and become more effective as the ambient conditions increase with depth resulting in a lower flow rate per effective cooling.

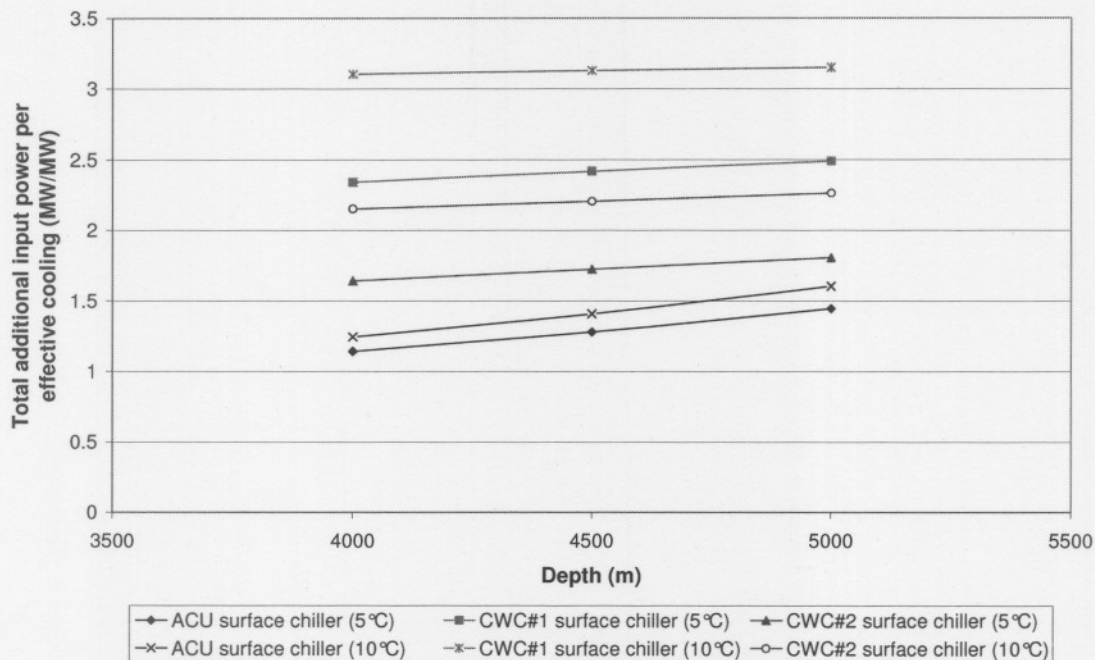


Figure 5.9: Additional input power with depth.

A by-product of the reduced surface chiller cooling capacity for ACUs compared to CWCs is a decrease in water evaporated in the condenser cooling tower of the refrigeration installation. This has an effect on the potable water the mine needs to purchase monthly from the water supplier.

Another aspect shown in Table 5-1 to Table 5-3 is the number of cooling units required to do the 1MW of effective cooling as this has an effect on the capital investment the mine will have to make. At a depth of 4000m 12 x ACUs are required compared to 4 x CWC #1s and 4 x CWC #2s. If the 300kW nominal capacity of a CWC is taken as benchmark it looks like 1.2MW worth of cooling cars are installed. However, due to the temperature conditions these units only deliver approximately 255kW.

Looking back at the figures and tables listed above it shows that the ACU has got energy efficiency potential if combined with a surface chiller. This will be analysed in more detail in Chapter 6.

5.2.3 Underground chiller configuration

As shown in Figure 5.3 and explained in Section 5.1 the underground chiller configuration will only be evaluated in combination with CWCs. Figure 5.10 shows that for every 1MW of cooling done by the CWC#1s the underground chiller needs to extract 1.27MW of energy from the water for a supply temperature of 5°C. Working with a COP of 2.57 for the underground chiller plant the compressor input power is calculated as 0.5MW, resulting in a total input power of 0.65MW per 1MW of cooling. This total power requirement is lower than for the surface chiller configurations mainly due to a smaller pumping power requirement based on the shorter pumping distances. The CWCs are also more effective due to the chilled water source being so close to them compared to surface configurations. Table 5-4 to Table 5-6 lists the results per depth for the different underground chiller configurations for both a 5°C and 10°C supply temperature.

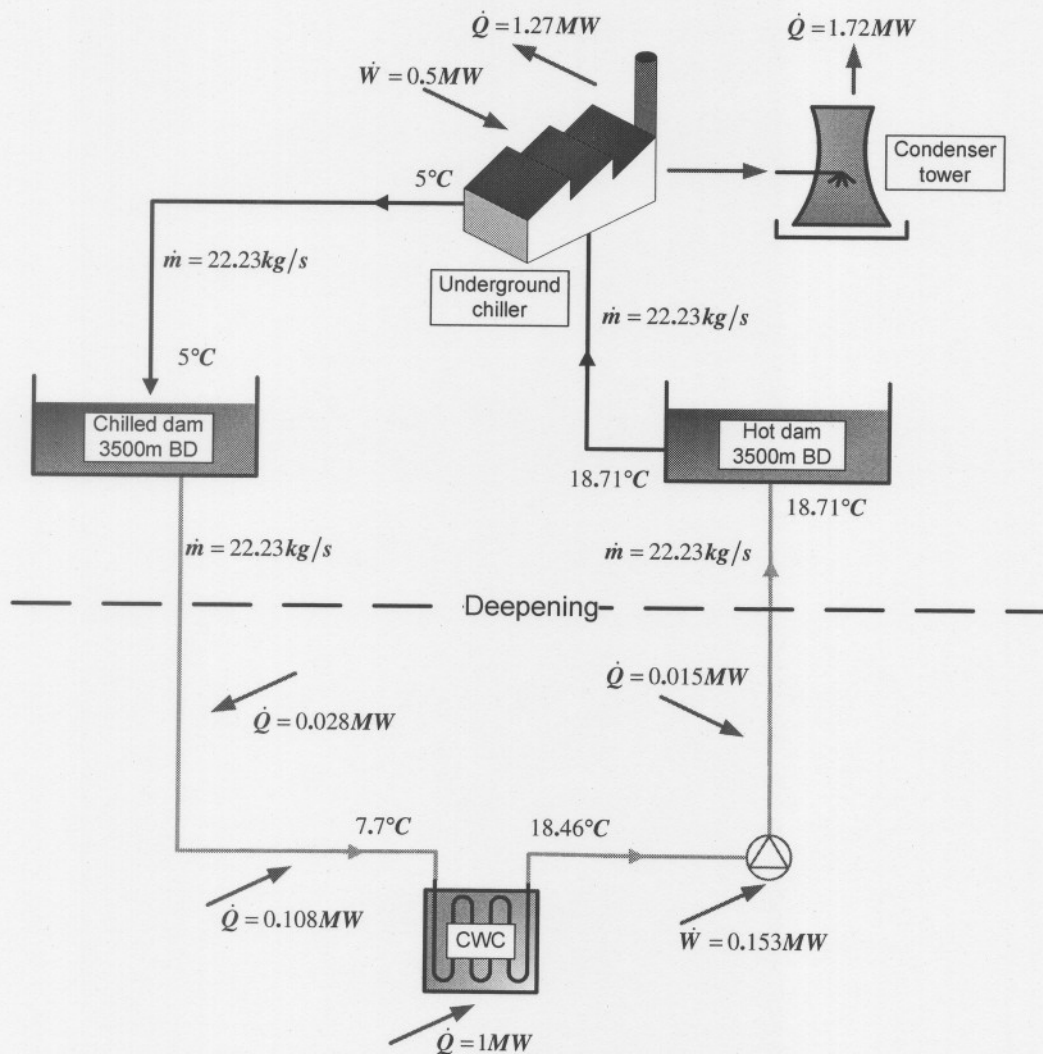


Figure 5.10: Underground chiller with CWCs.

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Table 5-4: Comparison per 1MW cooling 4000m below depth for an underground chiller.

Description	CWC #1 (5°C)	CWC #1 (10°C)	CWC #2 (5°C)	CWC #2 (10°C)
Pumping power (MW)	0.153	0.186	0.101	0.124
Chiller power (MW)	0.496	0.502	0.477	0.480
Total power input (MW)	0.648	0.688	0.577	0.604
Water mass flow rate (kg/s)	22.23	27.05	14.69	17.99
Underground chiller cooling (MW)	1.274	1.290	1.225	1.234
Cooling tower water (l/MW)	0.717	0.607	0.689	0.561
No of cooling units required	3	4	3	4

Table 5-5: Comparison per 1MW cooling 4500m below depth for an underground chiller.

Description	CWC #1 (5°C)	CWC #1 (10°C)	CWC #2 (5°C)	CWC #2 (10°C)
Pumping power (MW)	0.280	0.336	0.188	0.227
Chiller power (MW)	0.554	0.567	0.521	0.527
Total power input (MW)	0.833	0.903	0.710	0.754
Water mass flow rate (kg/s)	20.89	25.07	14.06	16.91
Underground chiller cooling (MW)	1.423	1.457	1.340	1.355
Cooling tower water (l/MW)	0.698	0.733	0.629	0.650
No of cooling units required	3	3	3	3

Table 5-6: Comparison per 1MW cooling 5000m below depth for an underground chiller.

Description	CWC #1 (5°C)	CWC #1 (10°C)	CWC #2 (5°C)	CWC #2 (10°C)
Pumping power (MW)	0.393	0.467	0.270	0.321
Chiller power (MW)	0.609	0.628	0.566	0.576
Total power input (MW)	1.002	1.095	0.836	0.897
Water mass flow rate (kg/s)	19.75	23.45	13.55	16.12
Underground chiller cooling (MW)	1.564	1.613	1.455	1.480
Cooling tower water (l/MW)	0.799	0.847	0.708	0.737
No of cooling units required	3	3	3	3

Figure 5.11 shows the total additional input power requirement of the underground chiller configurations compared to the results of the surface configurations in Section 5.2.2. For a depth of 4000m combined with a 5°C supply temperature there are reductions of 39% and 46% respectively if the underground configurations are used instead of ACUs combined with a surface chiller. Reductions of 45% and 52% are realised for 10°C supply temperatures.

What is evident from Figure 5.11 is that the input power for the underground configurations increases dramatically with depth. This is due to the increase in distance that the CWCs are from the underground chiller resulting in an increased pumping power and an increased load on the chiller installation due to increased return water temperatures.

From a power consumption perspective the underground chiller configuration looks like the most viable option for deep mining compared to surface water supplies. In Chapter 6 the underground configurations will be evaluated from an investment perspective to see if this is still the best solution.

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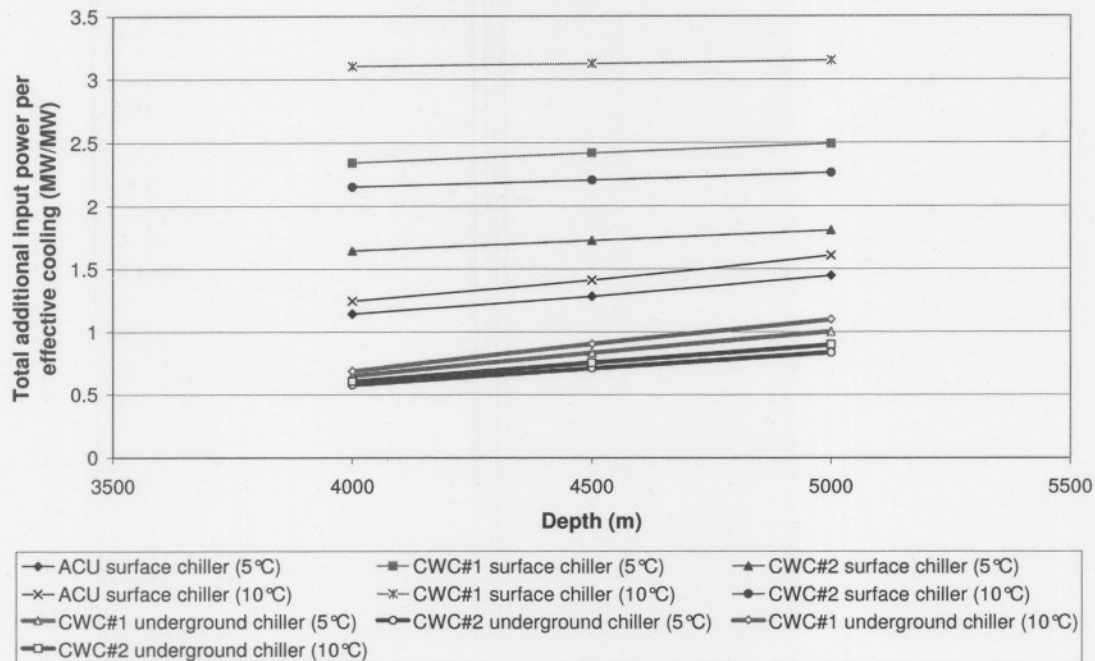


Figure 5.11: Additional input power with depth.

5.2.4 Underground dam configuration

The final option to be evaluated is the underground dam configuration shown in Figure 5.4. Figure 5.12 illustrates the process for an ACU configuration and Figure 5.13 for a CWC configuration at a depth of 4000m. It is important to note that only the effect of the additional energy on the existing mine network is taken into account as the water would have been pumped to the surface chiller for cooling regardless of the additional deepening network. However, due to the additional heat load the water arrives at a slightly higher temperature at the surface chiller thus resulting in an increased load on the plant. The only pumping power taken into account is the required pumping between the lower levels where the ACUs/CWCs are located and the level of the hot dam.

When comparing Figure 5.12 with Figure 5.13 it is clear that using CWCs to do the cooling results in an increased underground pumping requirement due to the amount of water circulated to and from the hot dam. Due to a supply temperature of 28°C from the hot dam the CWCs has a cooling capacity of only about 40kW leading to the installation of 24 units to do the cooling. In Chapter 6 it will be shown that this option is not economically viable.

As explained above, the increase in temperature of the water pumped from the mine has an effect on the surface refrigeration installation. A possible solution for this problem is the installation of a pre-cooling tower to handle the additional load as explained in Section 5.1.2 and Figure 5.5. Figure 5.14 illustrates the effect of the pre-cooling tower responsible for the additional 1.3MW of cooling required.

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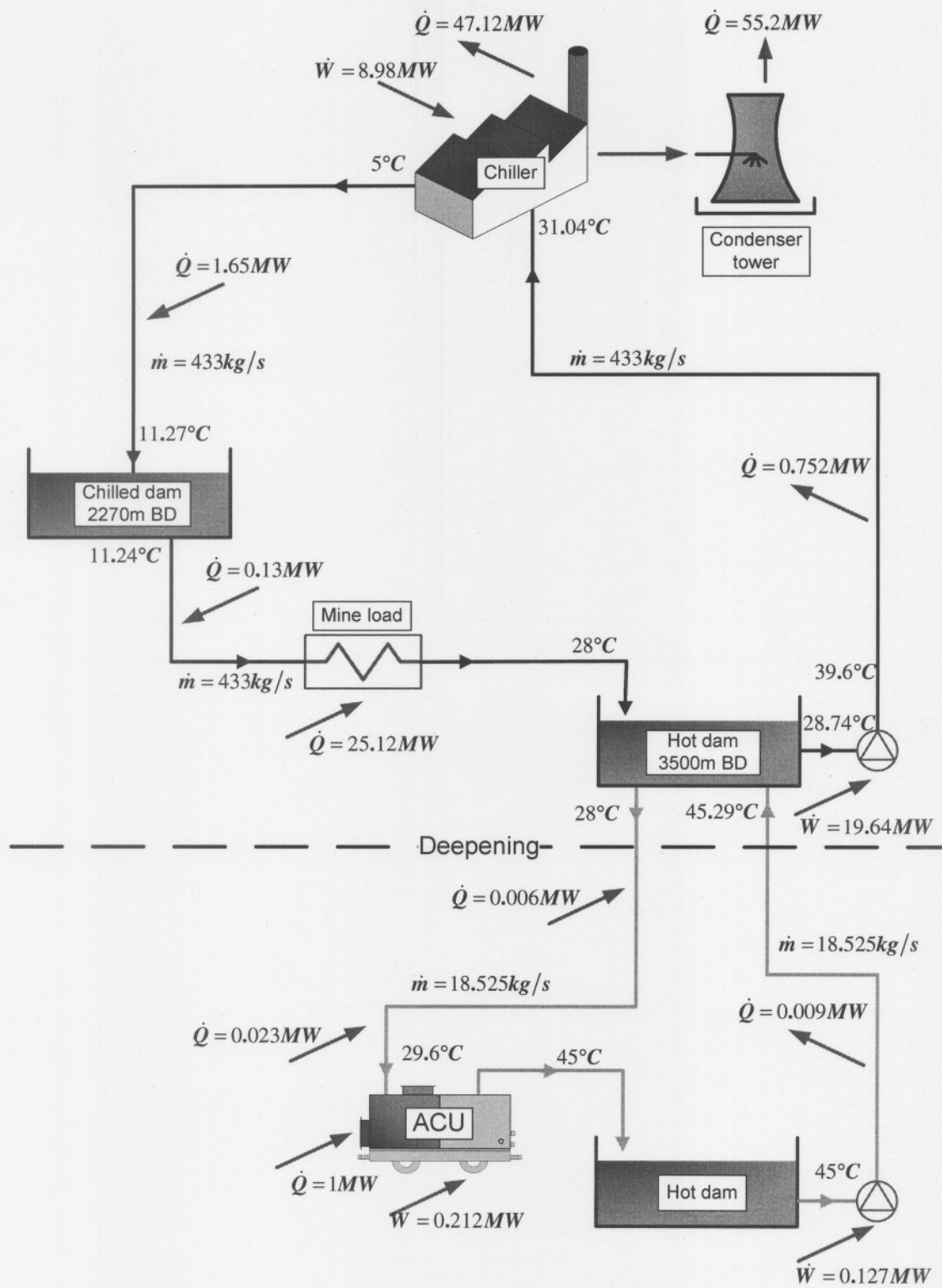


Figure 5.12: Underground dam with ACUs.

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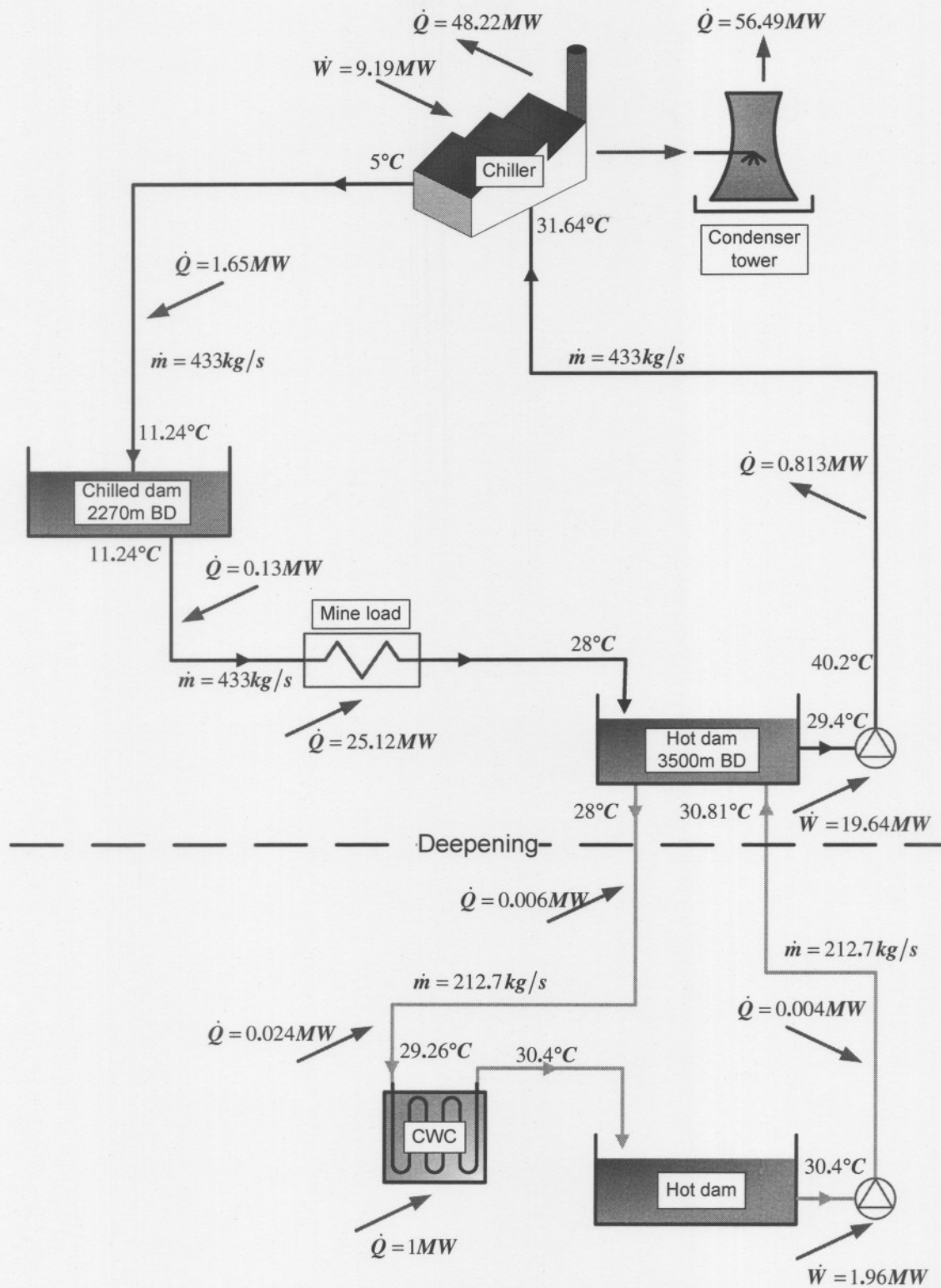


Figure 5.13: Underground dam with CWCs.

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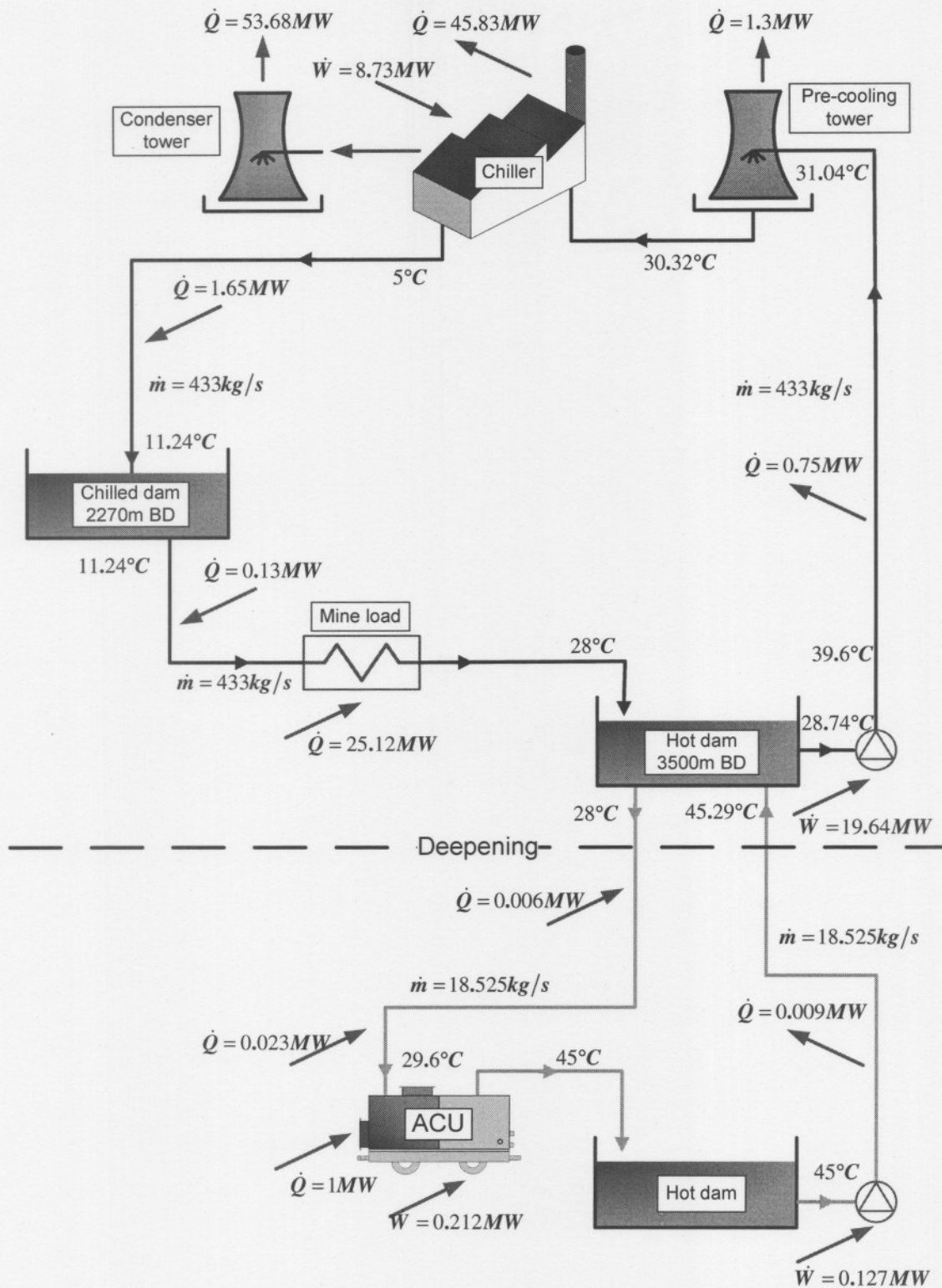


Figure 5.14: Underground dam with an additional pre-cooling tower.

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Table 5-7 to Table 5-9 compares the three possible underground dam configurations indicating the components that contribute to the total power consumption of each option.

Table 5-7: Comparison per 1MW cooling 4000m below depth for an underground dam configuration.

Description	ACU	CWC #1	ACU (Pre-cooling tower)
Pumping power (MW)	0.127	1.460	0.127
Chiller power (MW)	0.248	0.457	-
ACU power (MW)	0.212	-	0.212
Total power input (MW)	0.587	1.917	0.339
Water mass flow rate (kg/s)	18.53	24.0	18.53
Surface cooling (MW)	1.302	2.399	1.297
Cooling tower water (l/MW)	0.635	1.171	0.544
No of cooling units required	12	24	12

Table 5-8: Comparison per 1MW cooling 4500m below depth for an underground dam configuration.

Description	ACU	CWC #1	ACU (Pre-cooling tower)
Pumping power (MW)	0.271	1.876	0.271
Chiller power (MW)	0.275	0.535	-
ACU power (MW)	0.208	-	0.208
Total power input (MW)	0.755	2.410	0.479
Water mass flow rate (kg/s)	20.23	140.05	20.23
Surface cooling (MW)	1.445	2.806	1.44
Cooling tower water (l/MW)	0.705	1.370	0.604
No of cooling units required	12	16	12

Table 5-9: Comparison per 1MW cooling 5000m below depth for an underground dam configuration.

Description	ACU	CWC #1	ACU (Pre-cooling tower)
Pumping power (MW)	0.445	2.059	0.445
Chiller power (MW)	0.309	0.571	-
ACU power (MW)	0.208	-	0.208
Total power input (MW)	0.963	2.630	0.653
Water mass flow rate (kg/s)	22.34	103.39	22.34
Surface cooling (MW)	1.624	2.997	1.651
Cooling tower water (l/MW)	0.793	1.463	0.679
No of cooling units required	12	12	12

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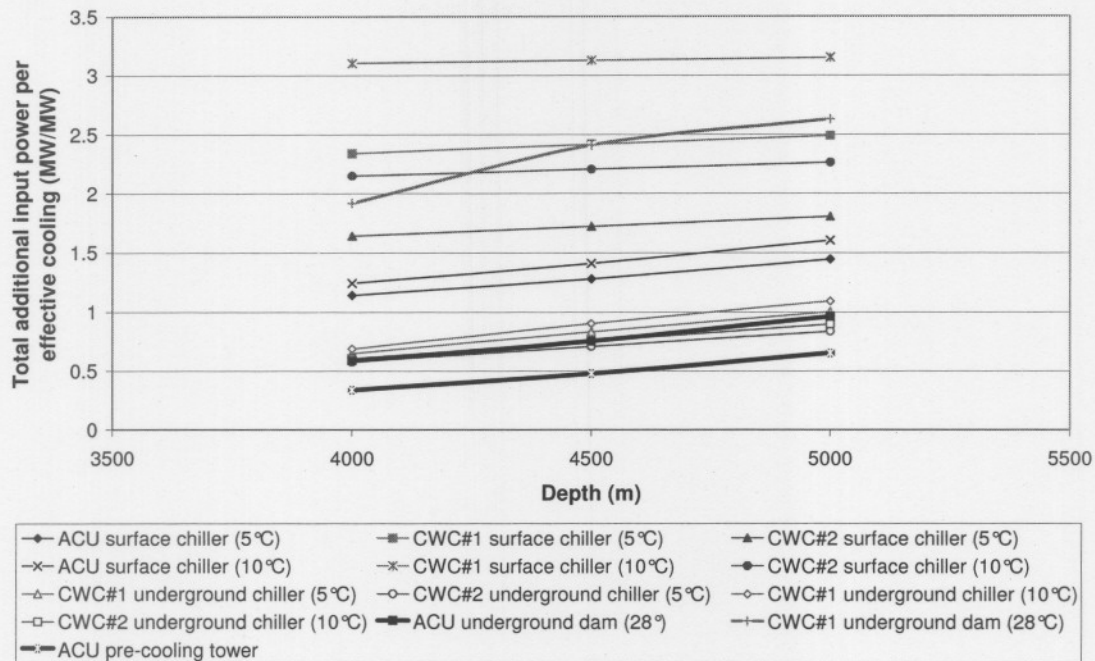


Figure 5.15: Power requirements with increase in depth.

Figure 5.15 shows how the total power requirements of these options compare to the other configurations previously discussed. From the graph it is clear that the input power for the CWC combination results in this option requiring as much power input as surface CWC combinations. The input power of the standard underground dam ACU configuration is similar to that of the CWCs combined with an underground chiller showing that this combination have potential for deep mine applications. From all combinations evaluated the total power input for the pre-cooling tower configuration is the lowest mainly due to less water circulated between the dam and the ACUs and the effect of the cooling tower coping with the additional load. The increase in power input with depth of the pre-cooling tower configuration is due to an increase in pumping power as a result of an increased water temperature with depth.

5.3 Summary

Looking at the power input of the different combinations the options showing the most potential for deep mining include the underground chiller with CWCs and the underground dam with ACUs, with and without the additional cooling tower. These results are in line with the statement made by Sheer *et al.* (2001) based on an economic study they conducted. Their finding was that the most economic form of cooling system for great depths is a hybrid system consisting of a combination of underground and surface refrigeration machines. They felt that the maximum refrigeration capacity installed underground would be limited by the combined heat rejection capacities of the air and water streams returning to surface, additional refrigeration would generally need to be installed on surface. Sheer *et*

a). felt that ice systems would be the best option for the surface plant even considering the cost of such an installation and the complexity of the installation itself.

With this in mind the ACU technology is an alternative that can work in combination with standard surface equipment which is less expensive than ice systems. The installation of this alternative is also less complicated than for an ice system.

It should however be noted that the ACU technology was never developed to replace underground refrigeration installations completely but supply the mining industry with an alternative to use in development areas. Once operations are expanding on the new levels a primary underground refrigeration installation should be considered to cater for the increasing cooling demand. These underground refrigeration installations however take time to be installed. The ACU technology can be implemented much faster resulting in development starting earlier than usual. Once the underground refrigeration installation is up and running the ACUs can still be used in development ends to do localised cooling.

In Chapter 6 the different configurations discussed in this chapter will be evaluated from an economic perspective to determine which of them will be most viable for deep mine applications.

CHAPTER 6

CASE STUDY - ECONOMIC ANALYSIS

CHAPTER 6

6. CASE STUDY - ECONOMIC ANALYSIS

In the previous chapter the technical potential of the different configurations were shown by comparing the total power requirements. In this chapter an economic analysis will be done to quantify the cost effectiveness of the different options. The economic analysis will include the running cost, capital investment and life cycle cost of the configurations. Based on the analysis an assessment of the energy efficiency and DSM potential of the most viable options for deep mine development will also be done. All cost calculations are based on the parameters and prices listed in Section 4.1.7.3.

6.1 Economic analysis

6.1.1 Surface chiller configuration

Figure 6.1 compares the running cost per year per depth of the different options. The running cost was calculated using the total additional power input to the system of each option as well as the purchase cost of the water evaporated in the surface condenser tower. For a depth of 4000m combined with a 5°C supply temperature there is a reduction of 50% in the yearly running cost if ACUs are used instead of CWC#1s. A reduction of 29% is realised compared to a CWC#2 configuration. The lower reduction compared to CWC#1s is due to CWC#2s only consuming 6.1l/s per unit compared to 9.0l/s for CWC#1s. For a 10°C supply temperature there is a reduction of 59% in the yearly running cost if ACUs are used instead of CWC#1s and a reduction of 41% compared to a CWC#2 configuration.

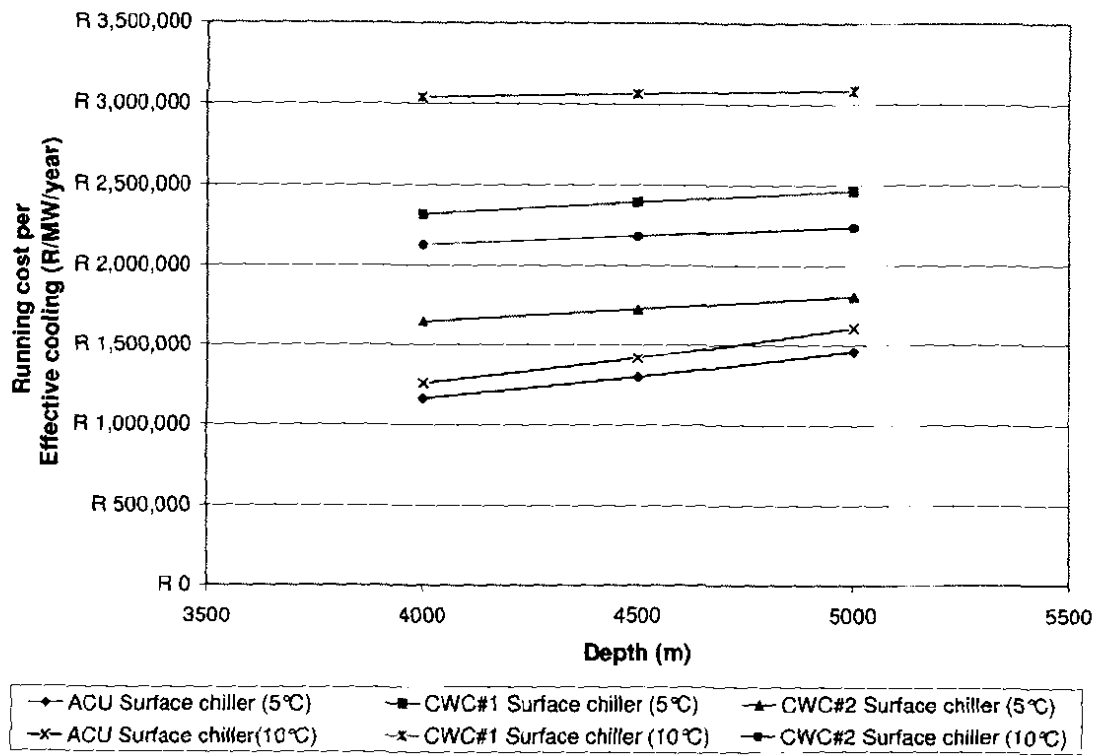


Figure 6.1: Running cost with increase in depth.

This shows that if a mine considers a surface cooling configuration the ACU system will result in a reduced yearly expense, saving the company money in the long run. This cannot be used as the only parameter to select a configuration. The next aspect to be evaluated is the capital investment to acquire the equipment for the different options. Figure 6.2 shows the capital cost per MW effective underground cooling for the different options. The capital cost consists of the purchase cost of a new surface chiller installation to accommodate the additional load due to deepening, and the cost of the number of ACUs or CWCs required for the cooling. The capital cost of the equipment used for this calculation was listed in Chapter 4, Table 4.3.

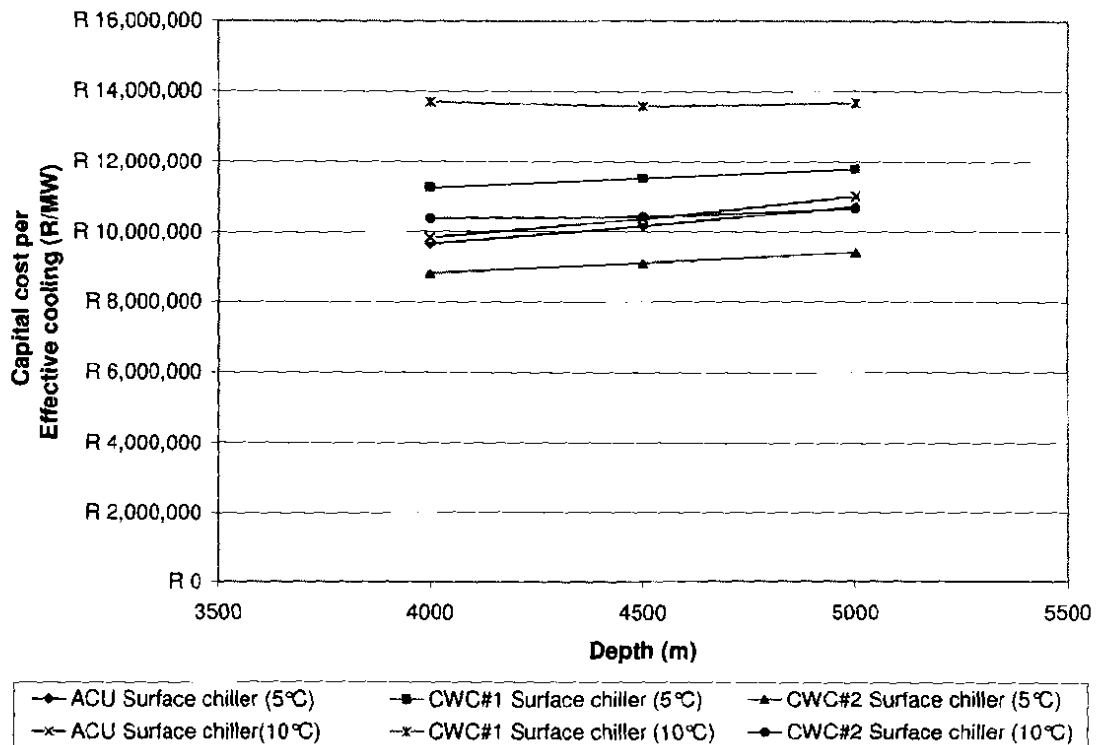


Figure 6.2: Total capital cost with increase in depth.

Looking at Figure 6.2 it is evident that the CWC#2 combined with a 5°C supply is the least expensive option followed by the ACU 5°C supply temperature configuration. Even though the ACU capital cost per nominal kW is almost 4 times higher than the cost of a CWC#1, and 6.5 times higher than a CWC#2, the load on the surface refrigeration installation is reduced resulting in the smaller requirement of a surface plant. This brings the total capital cost of the ACU configurations back to within a competitive range.

The increase in capital cost of the ACU configurations with depth is due to an increase in surface cooling capacity required due to an increase in water flow rate with an increase in water temperature.

From an investment viewpoint it is necessary to evaluate the present value of the different configurations based on the life expectancy of the equipment installed underground. As shown in Equation 4.22 the present value is a function of the initial capital investment and the running cost per year including compound interest. Figure 6.3 to Figure 6.5 shows the present value of the configurations for a life expectancy of between one and nine years. Based on information from personnel at the case study mine the life expectancy of a CWC is between six and eight years due to the harsh underground environment. With the ACU now only in operation for about three years it is difficult to project the life expectancy of the unit but based on the components used the life expectancy should be at least the same as for the CWCs and even slightly better.

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Comparing the present value of the different configurations for a life expectancy between six and eight years it is evident that the ACU configuration is the most economical for all depths evaluated. It is apparent from the graphs that only the CWC#2 combination for a supply temperature of 5°C is competitive with the ACU combinations. The only problem with this CWC projection is that it's based on a design supply temperature of 5°C. However as was mentioned in Chapter 5 the plant often operates at temperatures in the region of 10°C resulting in a less competitive scenario.

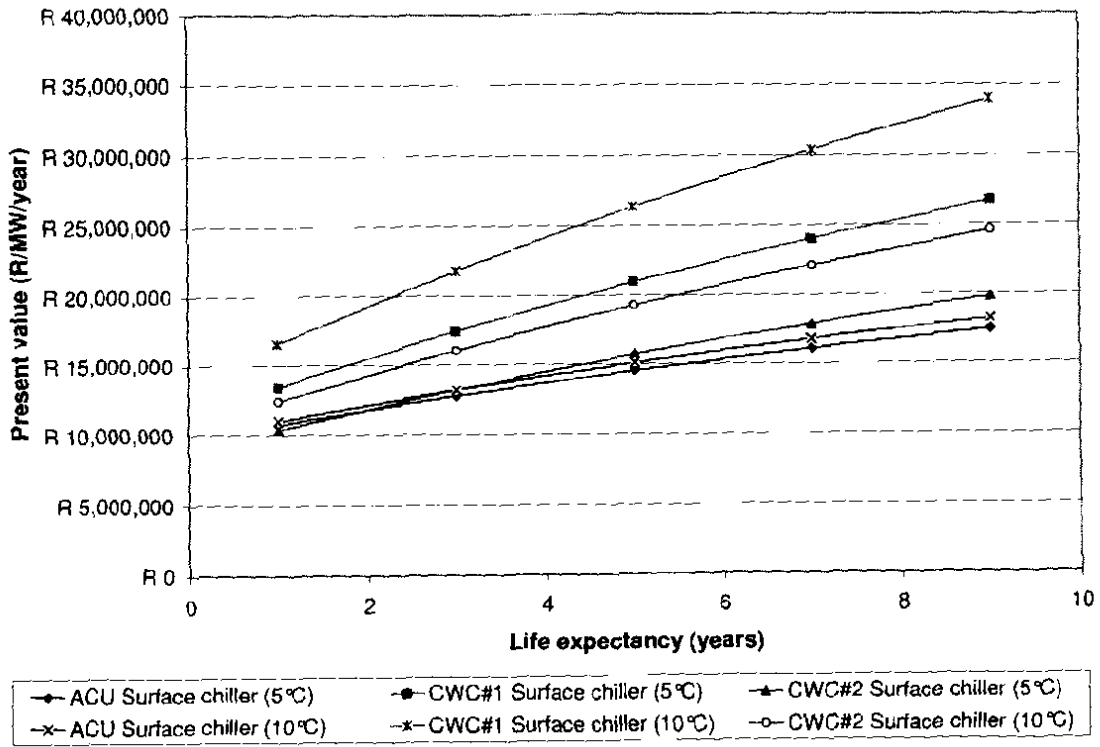


Figure 6.3: Present value for 4000m below depth.

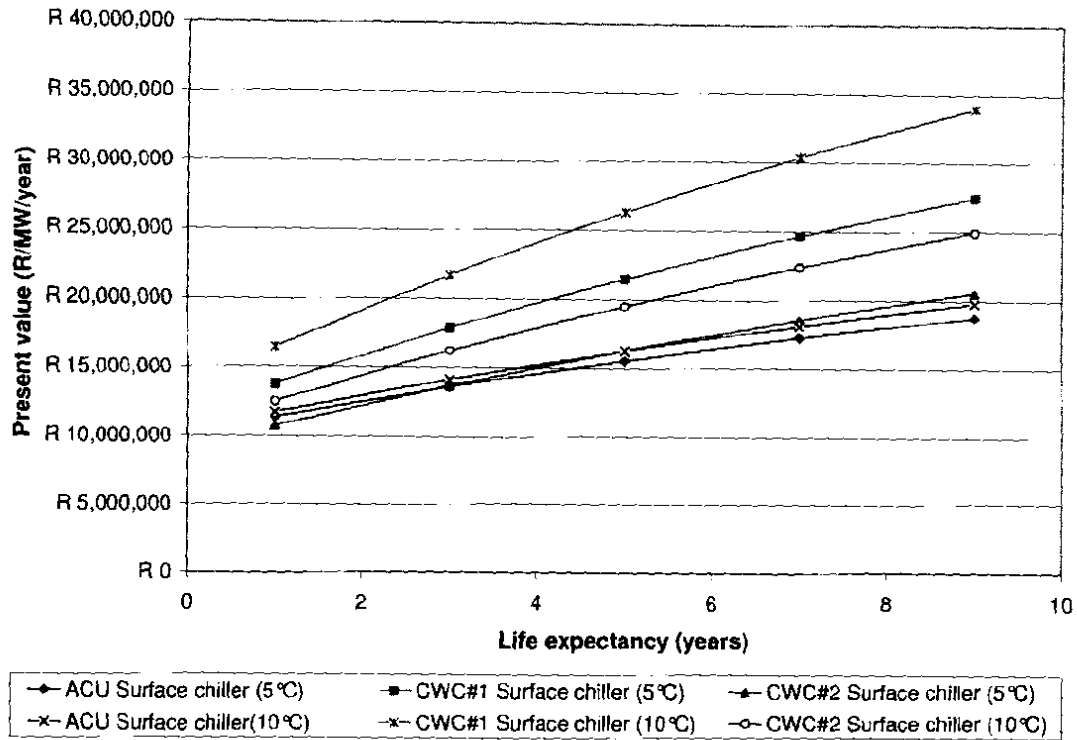


Figure 6.4: Present value for 4500m below depth.

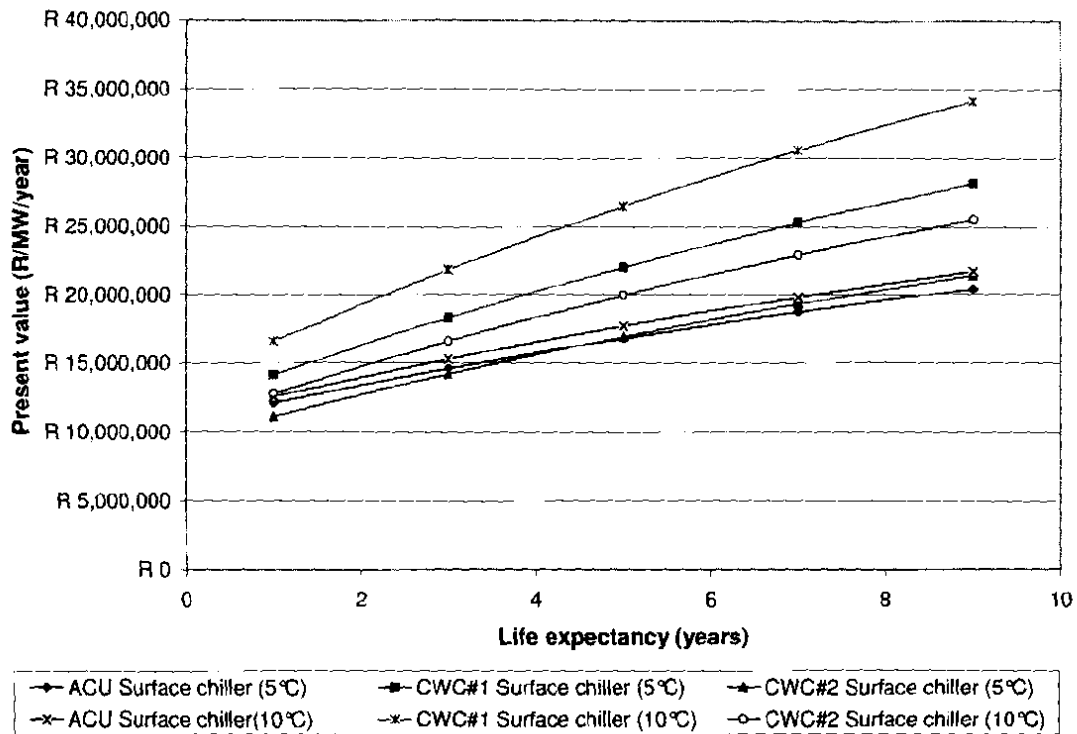


Figure 6.5: Present value for 5000m below depth.

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6.1.2 Underground chiller configuration

Figure 6.6 shows the running cost per year of the underground chiller configurations compared to the results of the surface configurations. For a depth of 4000m combined with a 5°C supply temperature there are reductions of 41% and 47% respectively if the underground configurations are used instead of ACUs combined with a surface chiller. Reductions of 44% and 50% are realised for 10°C supply temperatures. Due to the additional load on the underground chiller with depth and the extra pumping required the savings reduces to about 30% and 41% at a depth of 5000m.

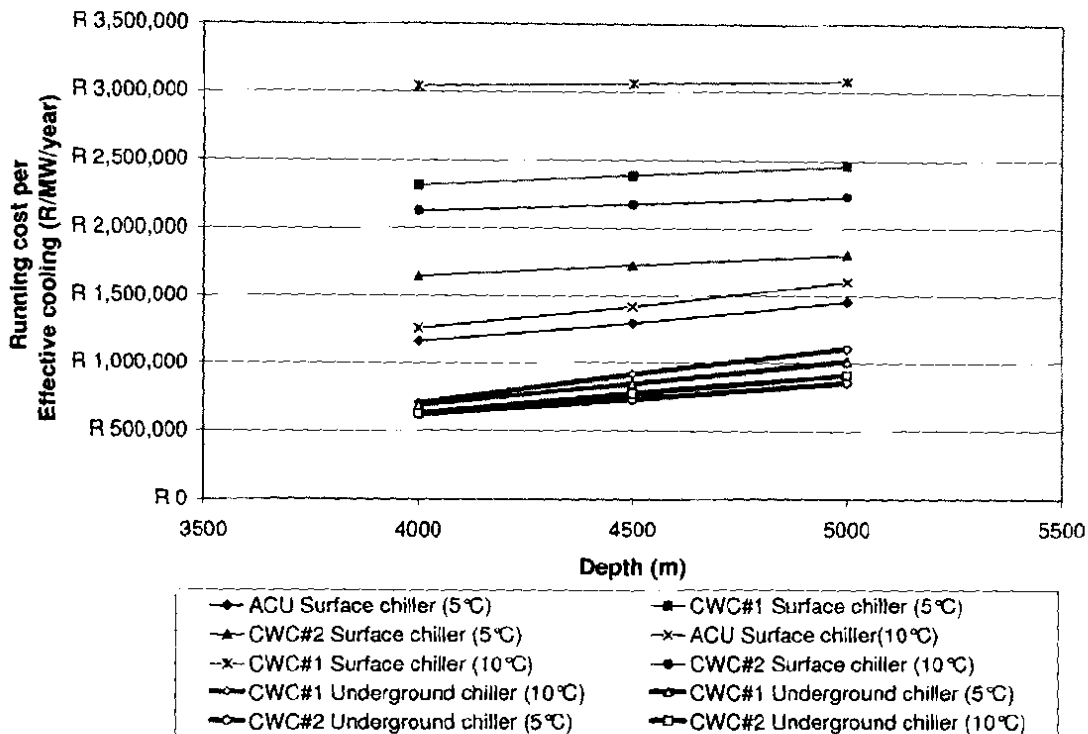


Figure 6.6: Running cost with an increase in depth.

As was shown in Table 4.3 the additional capital cost of an underground chiller installation is approximately R2,800-00/kW more than a surface chiller installation. However, because the cooling is done much closer to where the CWCs are located the losses in the system is minimal resulting in the CWCs performing at their nominal rating or above. This results in fewer CWCs necessary to do the required cooling per level. Figure 6.7 shows the capital cost compared to the previous options.

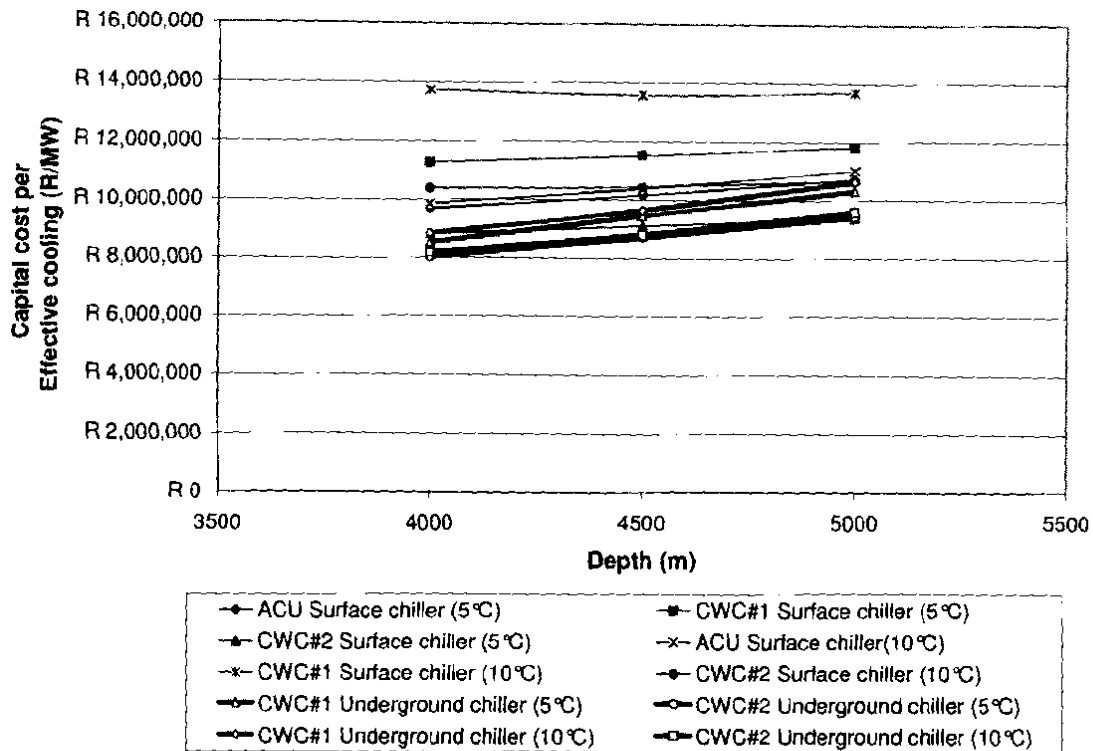


Figure 6.7: Total capital cost with an increase in depth.

If evaluating the present value of the underground chiller configuration it is evident from Figure 6.8 to Figure 6.10 that this option is economically more viable than the options previously discussed. Even when the more expensive CWC#1 units are used this option is still economically viable. The only factor that is not reflected in this comparison is the time consuming process of installing an underground chiller compared to a surface installation. This will have an effect on when development work can start on any new deep level. It is furthermore important to note from the graphs that with increase in depth the calculated present value increases. This is mainly due to the cooling being done further away from the cooling source resulting in increased water temperatures arriving at the CWCs resulting in the coils being less effective leading to the installation of additional units to cope with the heat load.

What can be concluded at this stage is that the conclusion drawn in Chapter 5 regarding the underground chiller configuration being ideal for deep mine applications was correct based on the economic evaluation thereof.

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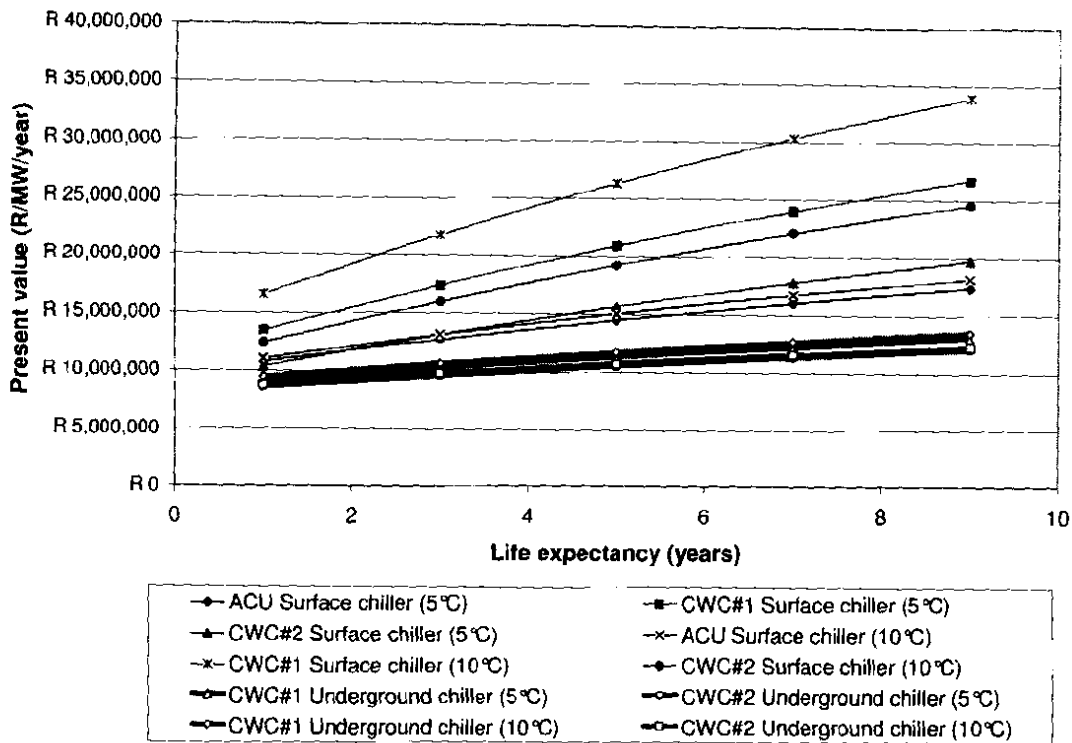


Figure 6.8: Present value for 4000m below depth.

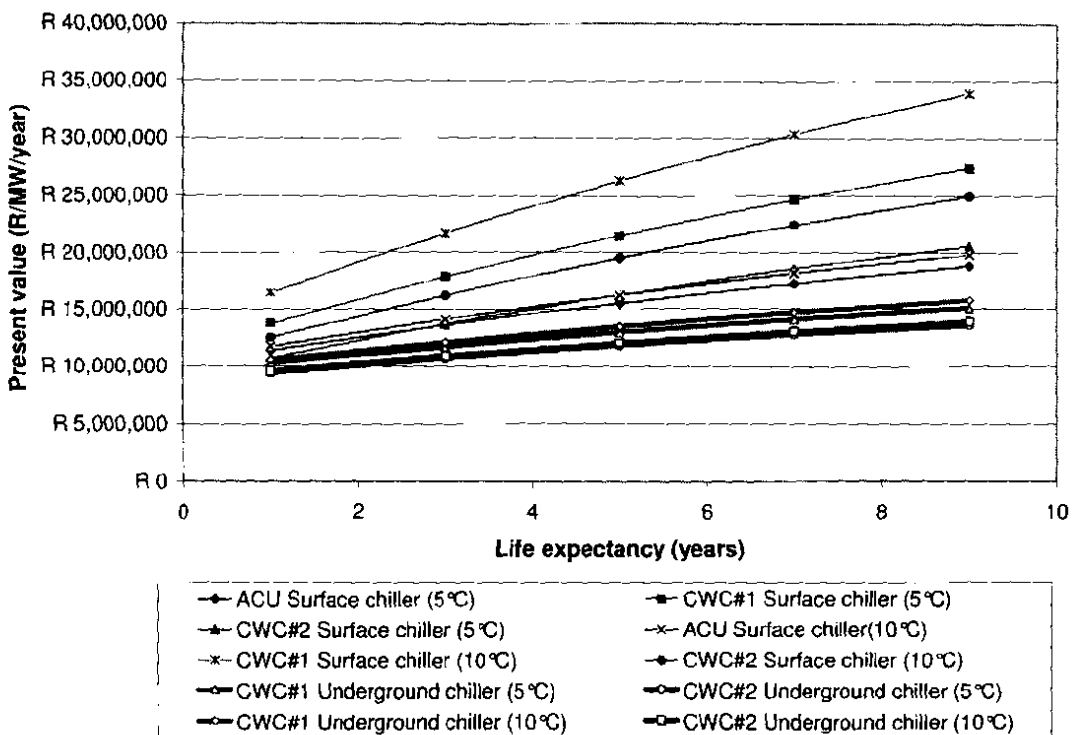


Figure 6.9: Present value for 4500m below depth.

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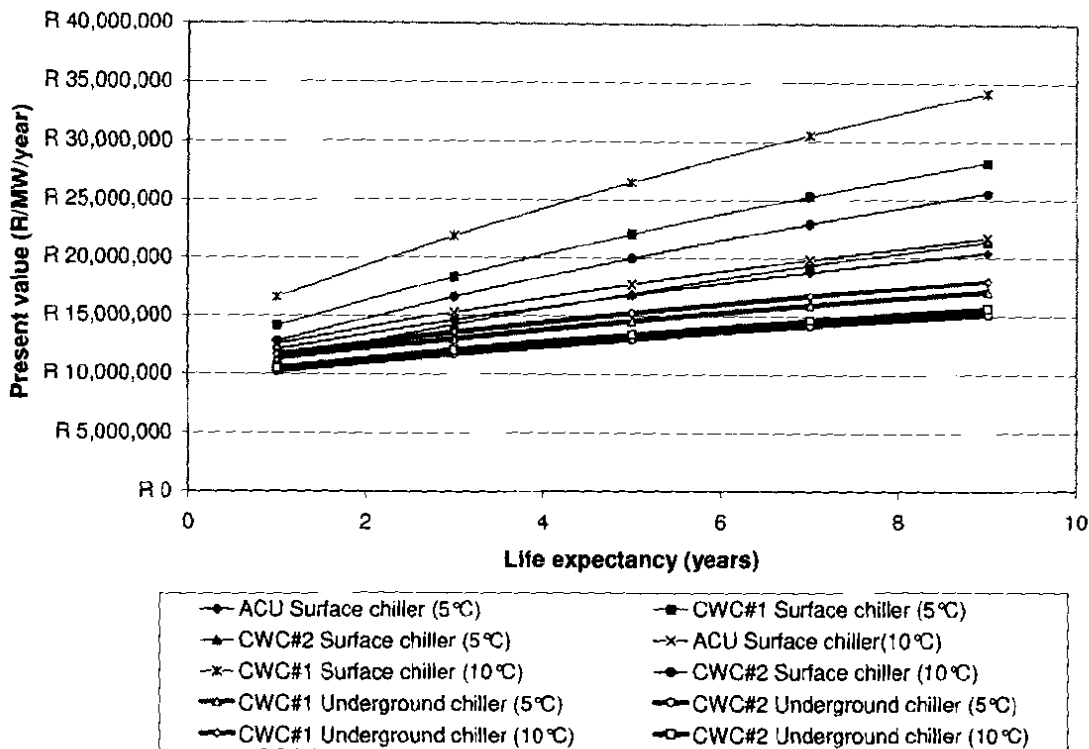


Figure 6.10: Present value for 5000m below depth.

6.1.3 Underground dam configuration

The final option to be evaluated is the underground dam configuration as explained in Chapter 5.1.2. Figure 6.11 shows a comparison of the running cost per year of the underground dam configurations compared to all other configurations evaluated. As can be seen the running cost for a CWC configuration increases dramatically with depth. This is mainly due to an increase in pumping power to return the large flow rates back to the surface refrigeration installation. The installed refrigeration capacity also increases to keep the supply water at the desired temperature. Again this reiterates the finding in Chapter 5 that this configuration should not be considered for deep mine expansion at all.

The running cost of the ACU configuration without the additional pre-cooling tower is virtually the same as for the underground chiller configurations, with the ACU pre-cooling tower having the lowest running cost. This is mainly due to the underground chiller plant having a lower COP than a surface chiller plant as well as less water circulated between the dam and the ACUs resulting in less pumping required. The ACU pre-cooling tower has a running cost reduction of 38% at 4000m and 20% at 5000m compared to a CWC#2 with a 5°C underground chiller supply temperature. The increase in running cost with depth of the ACU configurations is due to an increase in pumping power as a result of increased water temperatures with depth.

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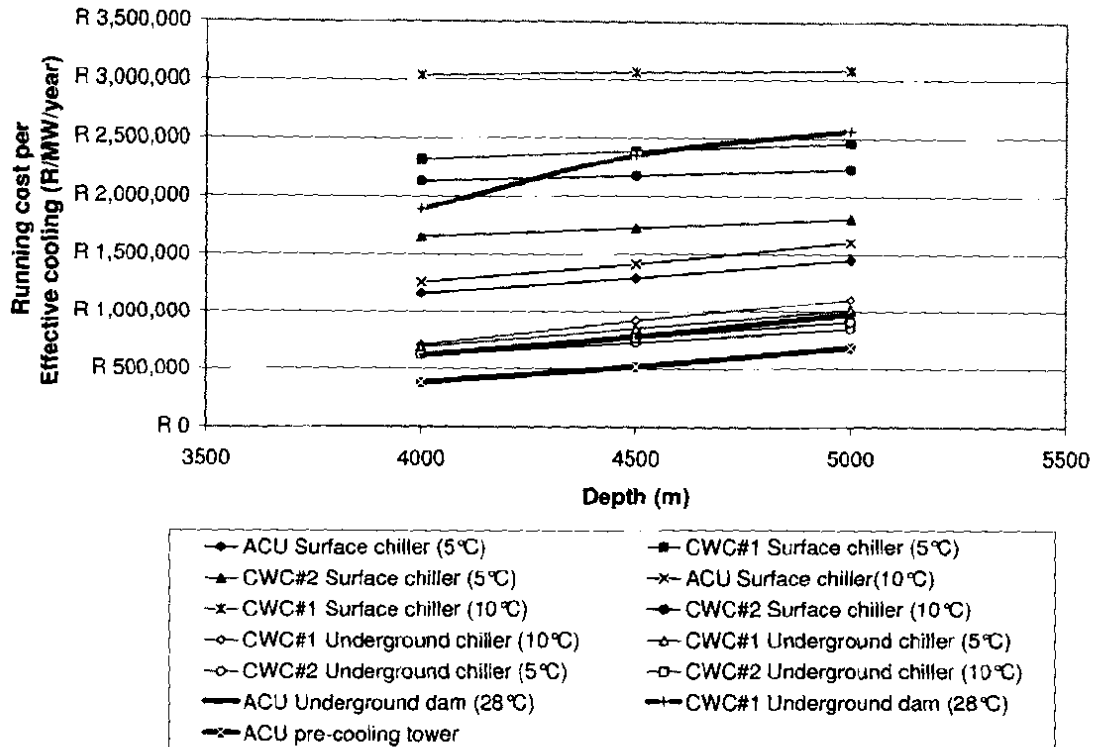


Figure 6.11: Running cost with increase in depth.

Figure 6.12 shows a capital cost comparison for all configurations. The capital cost of the pre-cooling tower configuration is on average 26% lower than for the CWC#2 with a 5°C underground chiller. This is mainly due to the cost of adding a cooling tower being about R3900/kW less expensive than an underground chiller plant as listed in Chapter 4, Table 4.3.

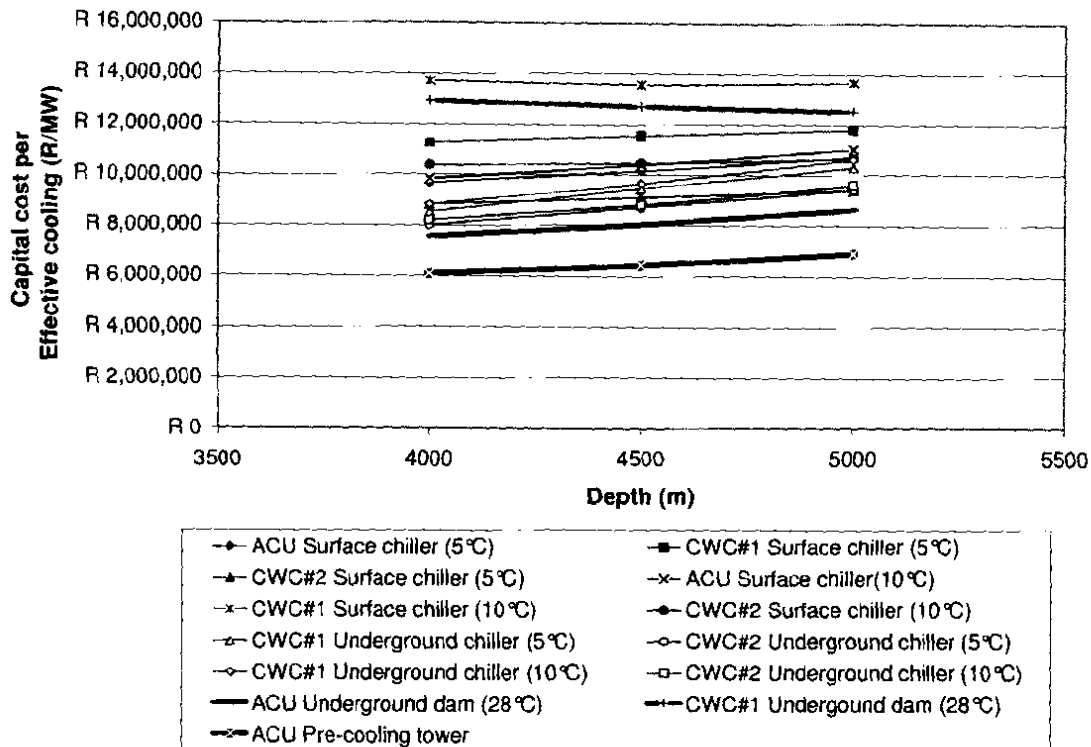


Figure 6.12: Total capital cost with an increase in depth.

Comparing the present value of the underground dam configurations in Figure 6.13 to Figure 6.15 two observations that can be made. Firstly, the ACU pre-cooling tower concept is economically the most viable option at any of the depths evaluated. Considering the practical application of this configuration, it should be easier to implement than any of the other options, especially an underground chiller plant.

Secondly, using ACUs with an expanded surface refrigeration installation can also be considered due to its present value being slightly better than the underground chiller configurations but again with the advantage of being easier to implement.

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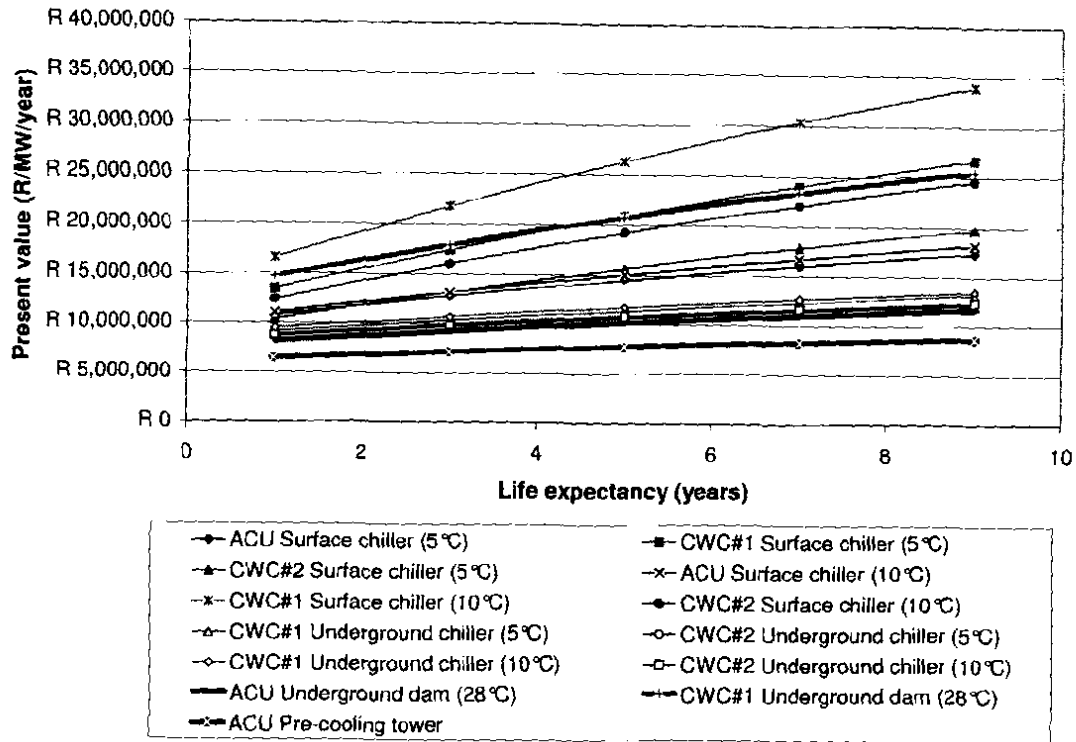


Figure 6.13: Present value for 4000m below depth.

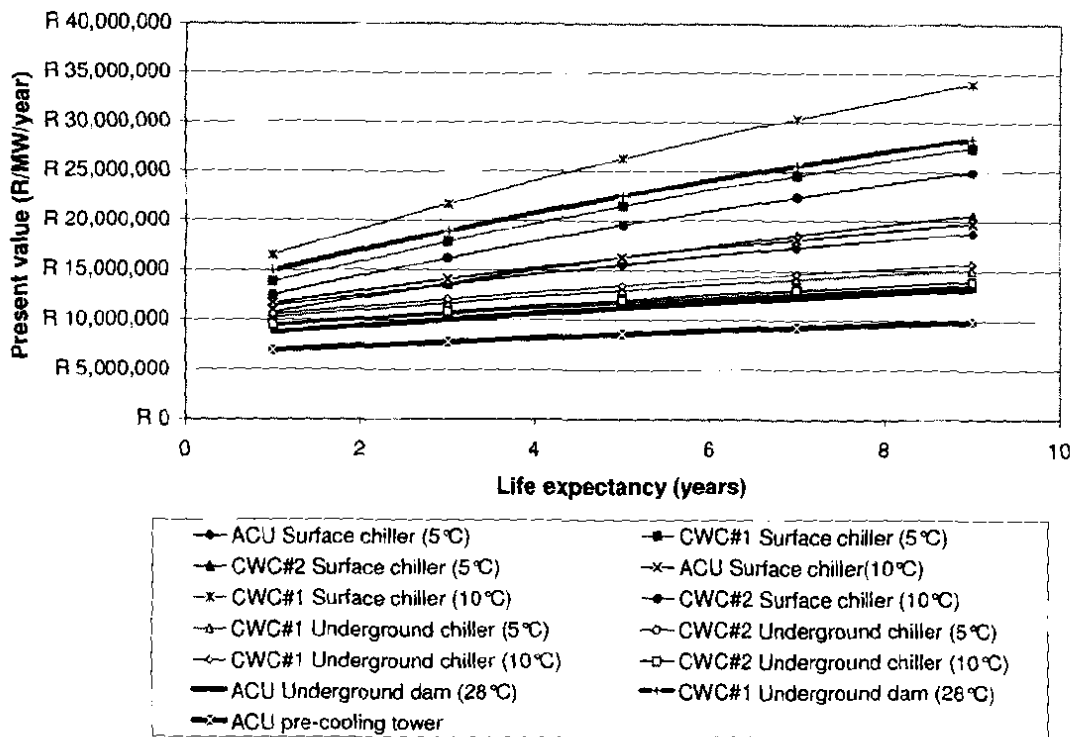


Figure 6.14: Present value for 4500m below depth.

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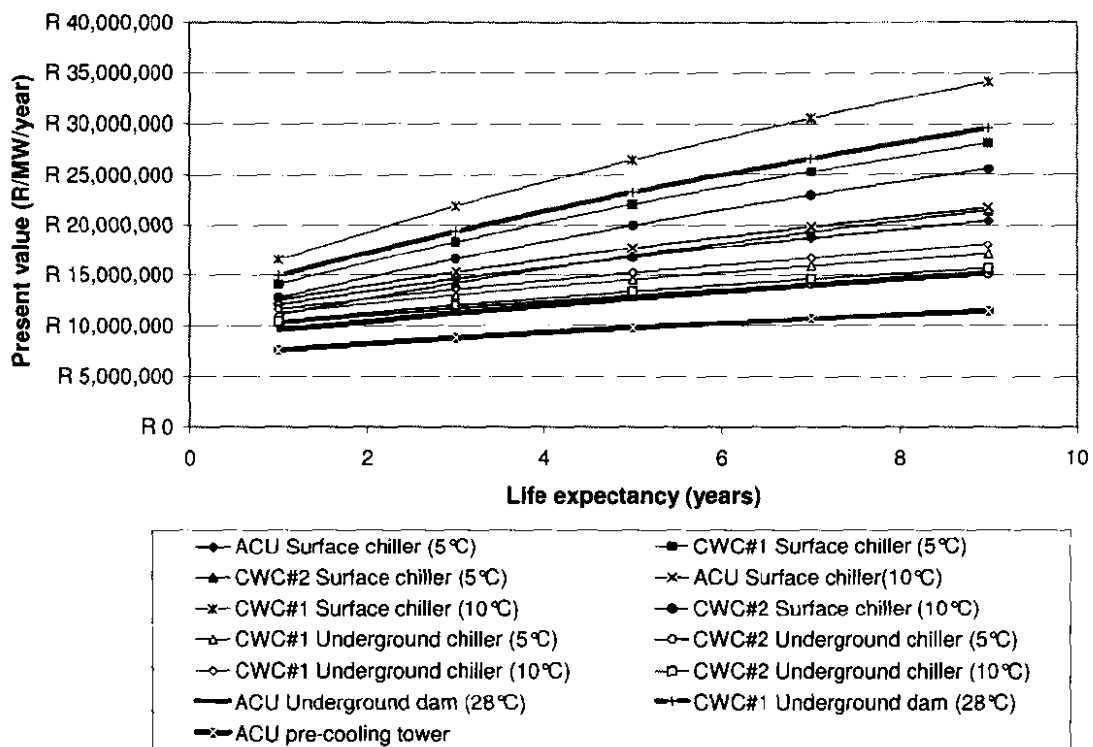


Figure 6.15: Present value for 5000m below depth.

As stated in Chapter 5, even with the ACU configurations being more economical there will be a need to install an underground refrigeration installation once the deep mine levels are fully operational to supply chilled service water for different mining operations. The ACU technology however gives an economically viable option while the underground development is still in progress. It has the added advantage of being modular so that it can be moved along as development progresses and stay part of the air cooling network.

6.2 Energy efficiency and DSM potential

To evaluate the energy efficiency and DSM potential of the different configurations it is necessary to choose one of the configurations as a baseline from where savings can be calculated. What became evident from the previous section is that only two main configurations should be considered for deep mine expansion, namely the underground chiller configuration and the underground dam configuration. This selection is indicated in Figure 6.16 by the highlighted area.

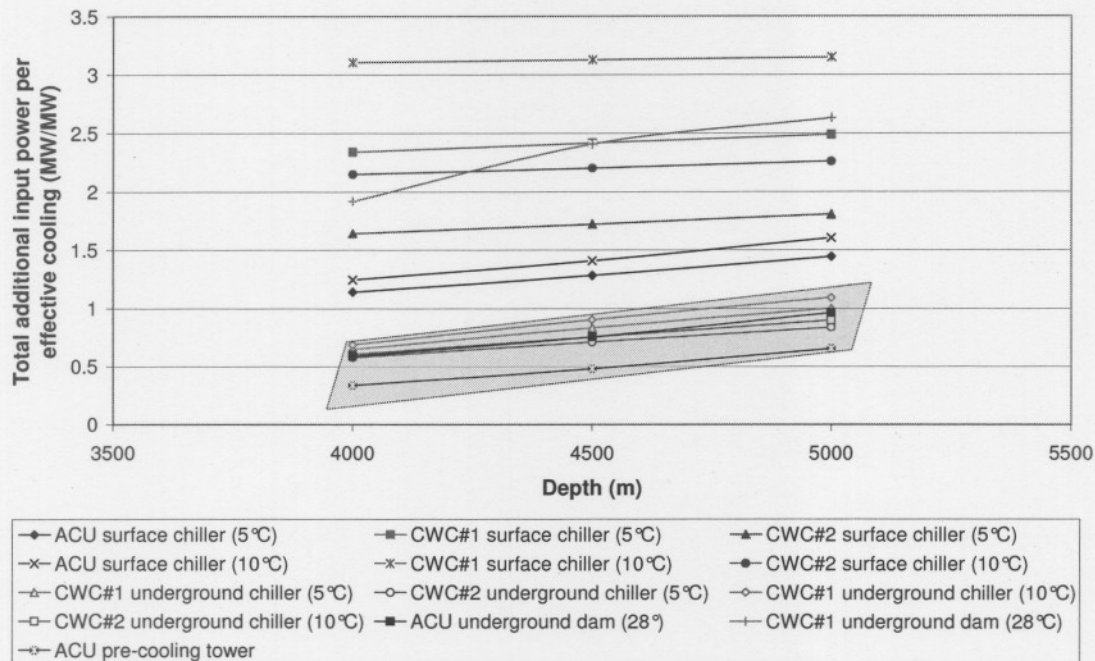


Figure 6.16: Configurations with energy efficiency and DSM potential.

Looking at this selection, the baseline will be assigned to the configuration with the highest additional input power, thus having no savings potential. In this case being the CWC#1 combined with a 10°C underground chiller configuration.

With this configuration as baseline the load reductions of the other configurations in the selection can be determined. Figure 6.17 shows the reductions in input power per MW cooling of the different configurations.

As stated in Chapter 2 Eskom initiated two incentives to encourage consumers to modify their patterns of electricity use as well as their total electricity consumption. Eskom is willing to invest funds to help companies reduce their peak demand and kWh consumption. When a mining operation expands there will be an increase in electricity consumption compared to current operations. This is due to additional mining infrastructure and cooling requirements. However, by installing energy efficient equipment reductions in electricity usage will be obtained compared to when less efficient technologies are installed.

If a company is willing to install energy efficient equipment, Eskom will fund R3000/kW reduction resulting from the installation of the new equipment. This is to help the client with the capital investment required to purchase the energy efficient equipment.

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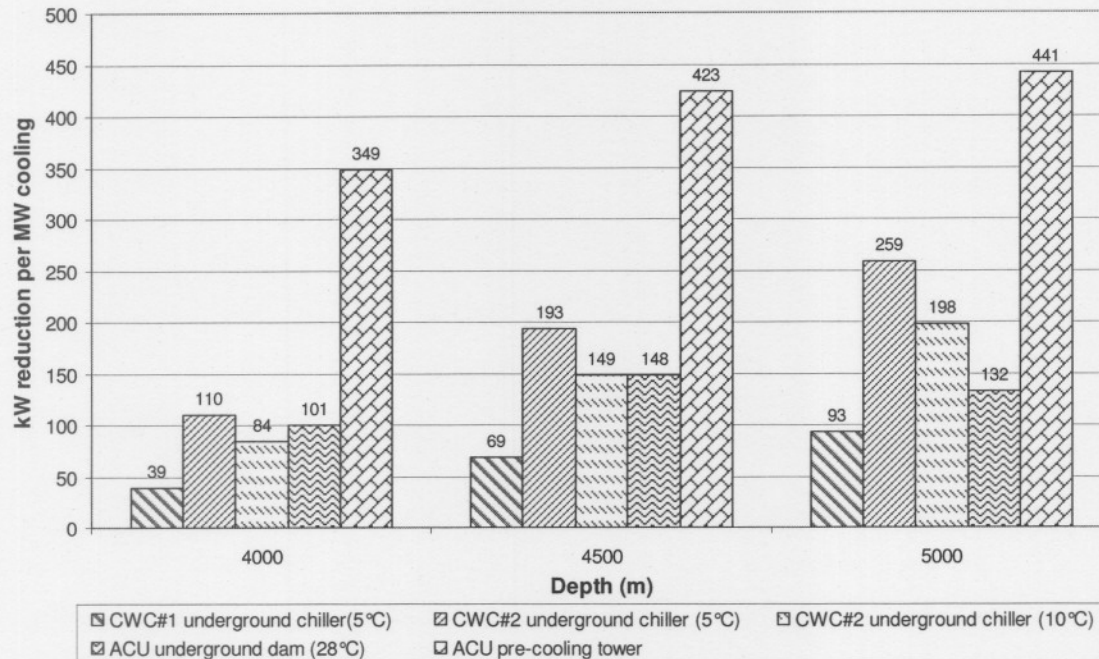


Figure 6.17: kW reduction compared to CWC#1 baseline.

The second incentive is for a peak demand reduction between 18:00 and 20:00. For this reduction Eskom is willing to pay R2.7-million/MW. The DSM potential for deep mine expansion lies in a reduction of added peak demand by the installation of energy efficient cooling equipment.

Because you may not apply for both types of funding for the same project, even if it has energy efficiency and DSM savings, the potential of both will be discussed separately.

Due to the constant underground heat load, independent of seasonal effects, the cooling system of a mine needs to be operational for 24 hours per day, 365 days per year. Therefore, the kW reduction generated per hour by any of the configurations will be the savings used to calculate the Eskom funding contribution as this saving should be constant for 365 days per year.

If the route of energy efficiency is taken, Eskom's funding contribution should be as indicated in Figure 6.18. The only significant contribution is for the ACU pre-cooling tower configuration with just over R1-million per MW of installed cooling capacity for a depth of 4000m increasing to a contribution of just over R1.3-million/MW for a depth of 5000m.

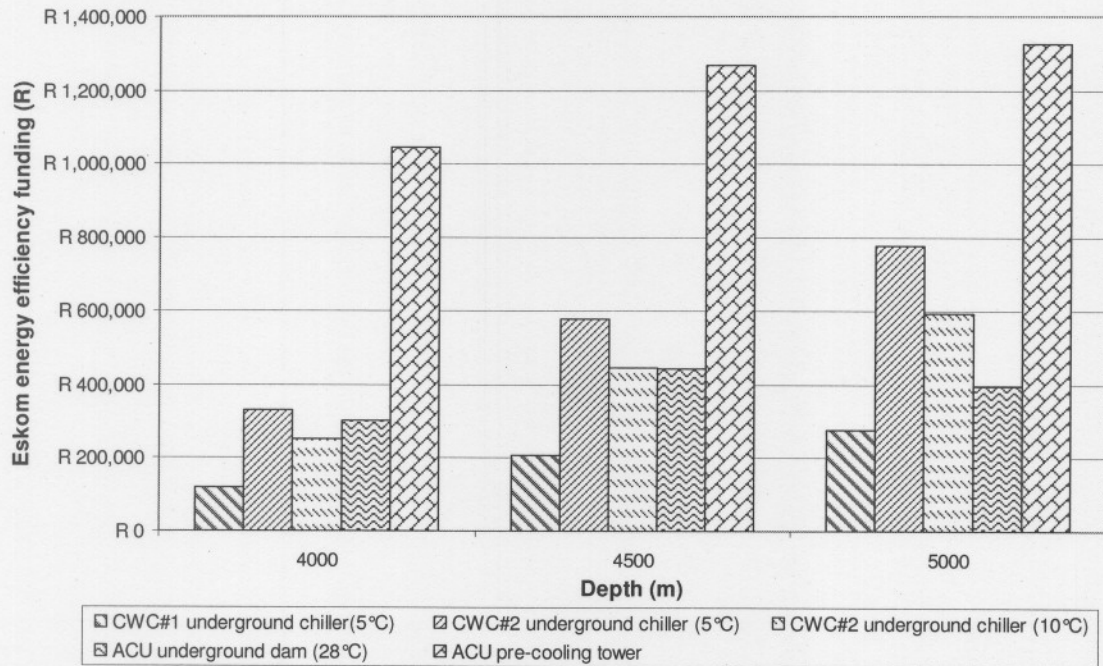


Figure 6.18: Eskom energy efficiency funding per MW of installed cooling capacity.

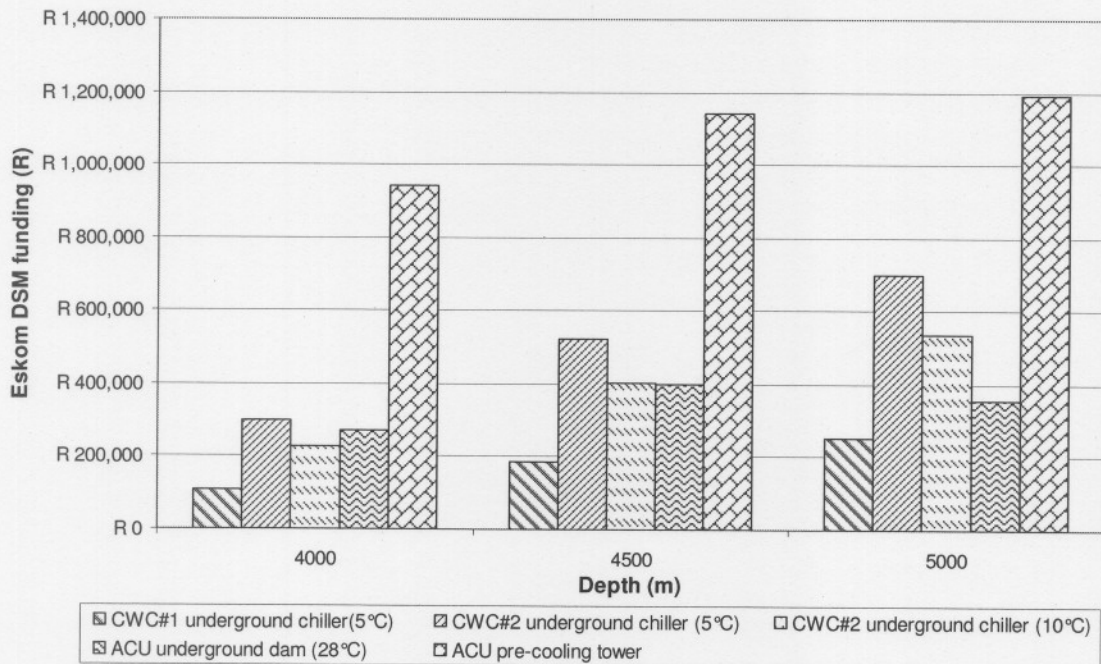


Figure 6.19: Eskom DSM funding per MW of installed cooling capacity.

If the route of DSM is taken Eskom's contribution should be as indicated in Figure 6.19. Again, the only significant contribution is for the ACU pre-cooling tower configuration with just under R1-million/MW for a depth of 4000m increasing to a contribution of just under R1.2-million/MW for a depth

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of 5000m. This shows that the biggest advantage for the mine will be to apply for energy efficiency funding. Eskom on the other hand will benefit from both the peak reduction during 18:00 and 20:00 and the kWh reduction associated with the ACU pre-cooling tower configuration if compared to the baseline configuration.

This energy efficiency contribution by Eskom results in a reduced capital investment required by the mine itself. The new capital investments are shown in Figure 6.20 compared to before Eskom funding. For the ACU pre-cooling tower configuration the capital investment is reduced by about 19%. This ultimately has an effect on the life cycle cost of the configurations as can be seen from Figure 6.21.

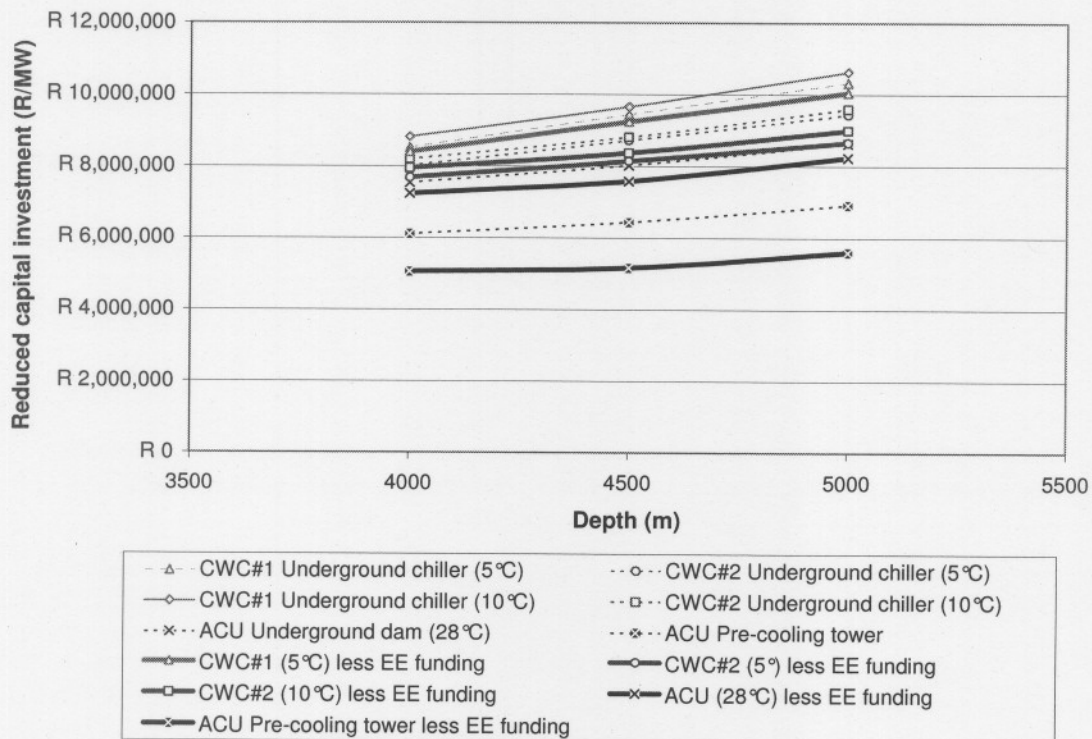


Figure 6.20: Reduction in capital investment due to EE funding.

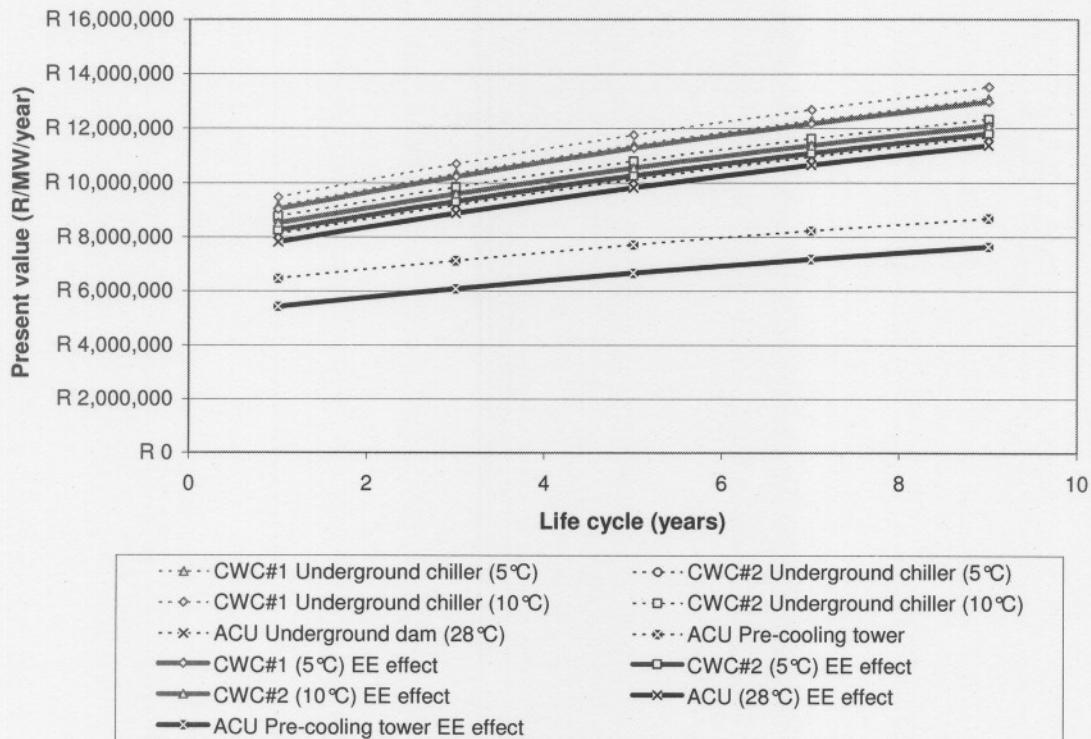


Figure 6.21: Reduction in present value due to EE funding.

6.3 Summary

Based on the results given in this chapter the following conclusions can be made:

From a mining perspective there is an economically viable alternative to a conventional underground chiller configuration for the cooling of development areas on new deep mining levels.

The ACU technology has energy efficiency potential compared to other deep mine configurations. The only drawback is that the funding from Eskom is less than 20% of the total investment cost to install the new technology. This is mainly due to the R/kW tariff used as incentive. However, the present value of the life cycle costs for the ACU with pre-cooling tower option is 34% less than the nearest of the underground chiller with CWCs options.

The ACU technology also holds DSM potential due to the reduction in input power compared to the conventional CWC technologies. This is mainly due to the constant operation of a mine cooling system to cope with the underground heat load. As stated before, Eskom benefit from both peak demand reduction and kWh reduction when the ACU technology is installed with only having to fund one of the processes.

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CHAPTER 7

CLOSURE

CHAPTER 7

7. CLOSURE

7.1 Conclusions

This study set out to develop a prototype ACU to supply the mining industry with a practical alternative to conventional cooling methods. Not only should this unit be able to establish a workable environment, but also be cost and energy efficient.

One of the main advantages of the nominal 80kW ACU that was developed is its very high positional efficiency due to its mobility and direct contact refrigerant-to-air evaporator. The test results showed that the unit produces a wet bulb temperature drop of between $5^{\circ}C_{wb}$ and $10^{\circ}C_{wb}$, depending on the volume flow rate. This is achieved using only a fraction of the water usually consumed by a CWC with the measured water flow rate less than $1.2l/s$ for the test range. A further distinct advantage of the new unit is that it can operate efficiently with high inlet water temperatures providing a COP of around 4.5 at an inlet water temperature of $30^{\circ}C$. This provides a degree of flexibility not obtained before since it can utilise normal service water as heat sink as opposed to chilled water at $5-10^{\circ}C$ that is required by conventional CWC's. Based on the experimental results it can be concluded that the unit has sufficient merit to establish a workable environment in remote underground locations.

To determine the energy efficiency potential of the ACU a mine system simulation model was developed specifically for this task. A potential shortcoming of the model is that it was developed with data from only one mining operation and will have to be verified against other mining scenarios. It is however important to note that the model was never developed to be a detail mine network analysis tool but only to compare the ACU technology with existing cooling configurations.

Using the system simulation a number of different cooling configurations were compared to determine the techno-economic potential. What is evident from the investigation results is that all of the ACU configurations provide viable alternatives for the mining industry when compared to the conventional CWC configurations at current mining depths. This is due to the resultant reduction in the total input power required for the system. However, for deep mine development only two main configurations showed true potential. The first of these is CWCs combined with an underground chiller of which the CWC#2 with a $5^{\circ}C$ chiller temperature showed the largest power input reduction. The second main configuration is the use of ACUs connected to a hot underground dam. The ACU pre-cooling tower configuration showed an average decrease in total input power of 32% compared to the best underground chiller combination. This configuration provides an alternative to conventional cooling methods to reduce the running cost per MW effective cooling.

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An economic evaluation of the configurations identified for deep mine development showed that the ACU pre-cooling tower option should provide the lowest life cycle cost by far. At a depth of 4000m the savings in life cycle cost for a six year life of plant is estimated at 28% compared to the best underground chiller option.

Based on the energy efficiency analysis only the ACU pre-cooling tower configuration showed sufficient potential to justify funding from Eskom. The funding however contributes less than 20% of the total capital investment that should be made. The ACU technology also showed DSM potential but due to the limited Eskom incentive in this regard an energy efficiency approach provides greater merit for the mining industry. Eskom on the other hand will benefit from both a peak demand reduction and a kWh reduction if the ACU technology is implemented.

To conclude, the ACU cooling technology described in this thesis offers the mining industry with a practical, cost effective and energy efficient alternative for existing mining operations as well as future deep mine undertakings.

7.2 Recommendations for further work

Based on the results of this study the following was identified as areas that need further attention.

- From a design perspective the concept of using a different refrigerant like R407C should be investigated to extend the capacity of the current unit to over 100kW nominal without changing it physically.
- To increase the ruggedness of the ACU during transport the use of screw type compressor technology should be investigated. Screw compressors are more expensive but in the long run may be more viable from a maintenance perspective.
- During the underground testing phase of this study it became evident that the mining industry would like to increase the air volume flow rate through the unit even if it results in a smaller temperature change. The effect of using a higher volume fan on the evaporator should be investigated.
- From a simulation perspective the mine system model developed should be tested against information from other mining operations to validate the accuracy and wider applicability. Other cooling configurations found in industry should also be evaluated to determine the impact of the ACU technology for possible deep mine expansion.
- Finally but most important the ACU technology should be applied on a wide scale in a mining operation to truly evaluate the impact of the technology for both the mining industry and Eskom.

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APPENDIX A

AIR COOLING UNIT DESIGN

A. AIR COOLING UNIT DESIGN

The design of the ACU will be divided into two main headings, namely a mechanical design and a refrigeration design. The design described here is based on the specifications listed in Chapter 3.3.

A.1. Mechanical design

To conform to the specification of being mobile the physical layout of the ACU was limited by the dimensions of the cage. The prototype ACU was developed based on cage dimensions of:

- Length - 3150mm
- Width - 1650mm
- Height - 1800mm

For the ACU to fit easily inside the cage and for the refrigeration components to fit inside the ACU the maximum dimensions were limited to:

- Length - 2580mm
- Width - 1530mm
- Height - 1650mm

Another critical parameter is the rail gauge of the mine specific transportation system. For the case study mine the rail gauge is 760mm.

For mobility the ACU was equipped with a rolling stock fixed to the cabinet housing the refrigeration components. This makes transportation of the unit by mining personnel more convenient.

For the physical layout of the ACU it was decided to divide it down the length into two compartments as shown in Figure A.1. The one compartment contains all the refrigeration components excluding the evaporator, and in the second compartment only the evaporator coil is installed. The reason behind the separate compartments is twofold. Firstly due to the dust content underground it is advisable to keep the compressors and electrical components in a semi dustproof section away from constant air movement. This will extend the life expectancy of the components. Secondly from a maintenance perspective cleaning the evaporator coil becomes relatively simple. Water can be sprayed from the top over the coil to remove any dust build-up without being concerned that the compressors or electrical components will be damaged. The only two connections between the compartments are the refrigerant supply line from the expansion valve to the evaporator and the suction line returning from the evaporator to the compressor. Where these two lines run through the separation plate it can be sealed with relative ease.

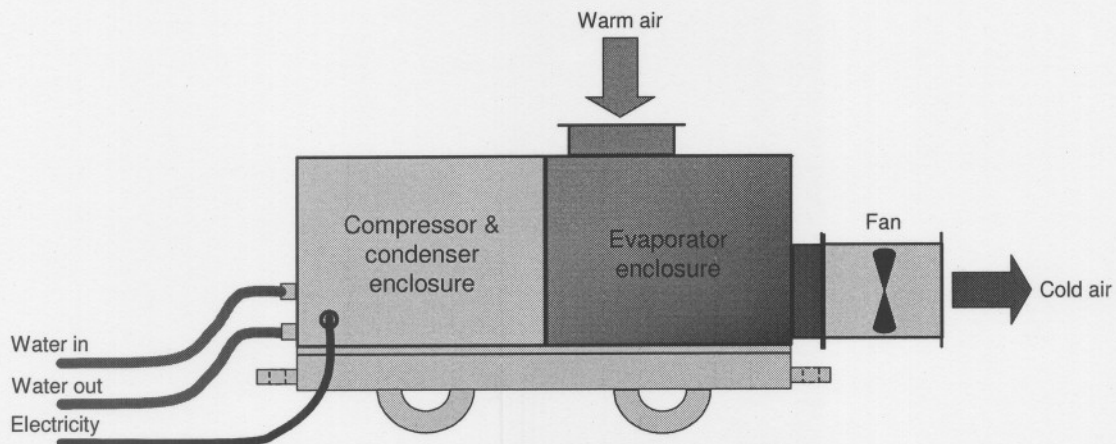


Figure A.1: Basic concept of the ACU.

The evaporator compartment is equipped with air inlet and outlet flanged connections, with the inlet at the top as shown in Figure A.1. The reason for this is that the unit can operate in two possible configurations. The first configuration consists of the ACU being connected to the fresh air ventilation ducting installed in the haulage. The fresh air is then cooled via the evaporator and blown directly into the area being cooled. This cooled air can also be ducted back into the fresh air supply and sent deeper into the mine.

The second configuration consists of the ACU being placed in a high temperature area requiring cooling. Warm surrounding air is circulated from the surroundings through the opening, over the evaporator and then blown directly via a fan back into the environment. The main advantage of this configuration is its high positional efficiency.

Figure A.2 shows the internal split layout of the prototype ACU, with the refrigeration components to be discussed in Section A.2.

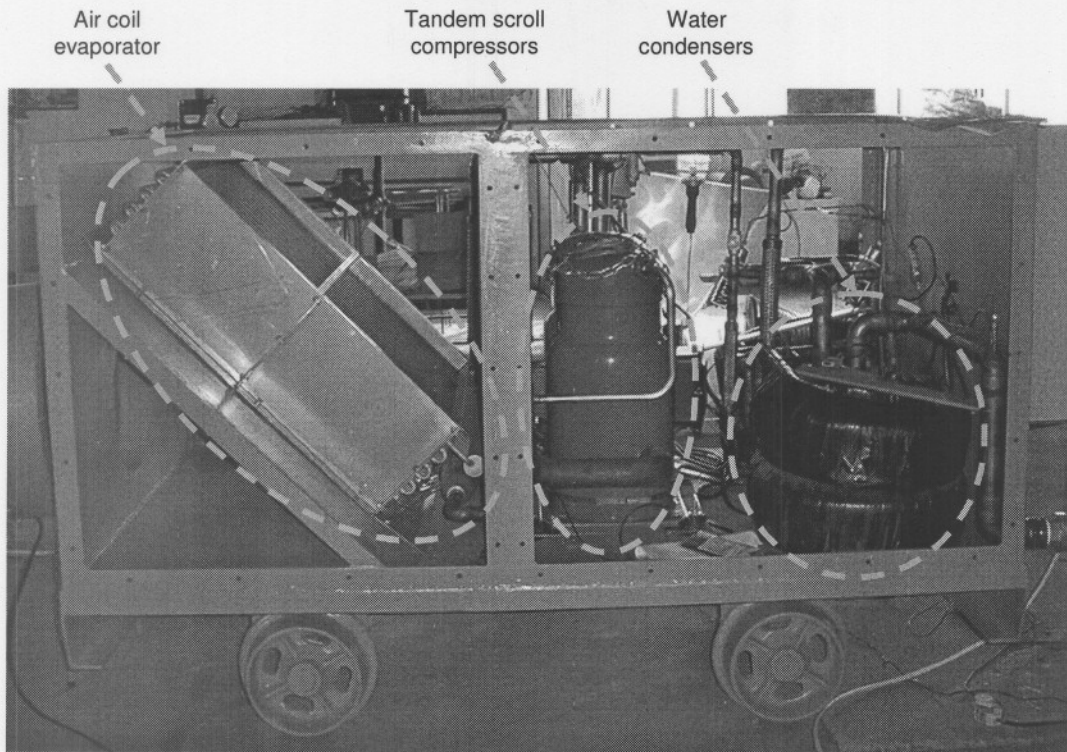


Figure A.2: Internal layout of the prototype ACU.

As stated earlier, the ACU has a fixed rolling stock for easy transport along the mine rail system as is evident from Figure A.2 and Figure A.3. The chassis is furthermore equipped with a buffer on both sides for connection to an underground train or other rolling stock being transported. In Figure A.3 ring like connections are visible on both sides of the buffer. This is used for rigging or slinging the ACU from the rail to its desired location.

For the first prototype the electrical panel was located on the outside of the unit as can be seen in Figure A.3. The main reason for this was that the monitoring equipment installed in this panel needed to be accessed easily for the retrieval of data. However, as will be discussed later in section A.3 the position of the electrical panel led to a few unforeseen problems. The water inlet and outlet are also located on the same side as the compartment housing the water condensers.

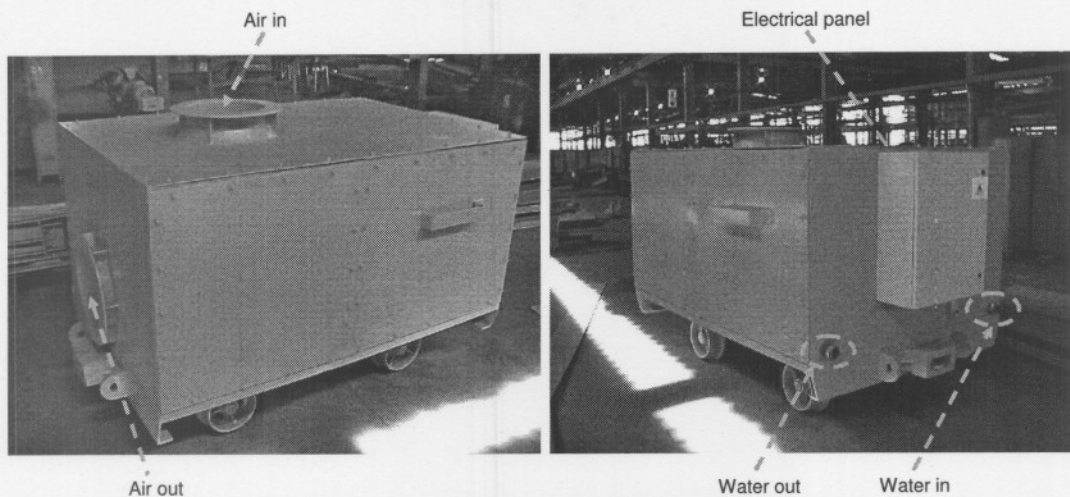


Figure A.3: External view of the prototype ACU.

A.2. Refrigeration design

In this section only the design and selection of the main components of the refrigeration cycle, shown in Figure 3.3, will be discussed. It must be noted that to comply with mine regulations regarding the use of environmentally friendly refrigerants the design was done for working fluid R134a.

A.2.1. Compressor

In order to make the ACU rugged but at the same time maximise the efficiency, it was decided to implement scroll compressor technology instead of conventional reciprocating compressor technology. During the design phase the largest commercially available R134a scroll compressor from Performer¹, the SZ185, had a cooling capacity of only 40.97kW at 15°C_{evap} / 50°C_{cond} ARI conditions. Due to the low capacity of a single compressor it was decided to opt for a tandem compressor configuration as shown in Figure A.4. The Performer SZ370 specifications at 15°C_{evap} / 50°C_{cond} ARI conditions are listed in Table A-1.

During the explanation of a heat pump cycle in Chapter 3.2 it was stated that the COP_{cooling} is usually approximately 2. As is evident from Table A-1 the COP_{cooling} of the scroll compressor is 5.03. The main two reasons for this are firstly that a scroll compressor has a higher efficiency than a conventional piston compressor normally used in heat pump applications. Secondly, the operating conditions of 15°C_{evap} / 50°C_{cond} contribute to the higher COP. In the commercial sector the evaporation temperature is usually between 5°C and 12°C depending on the ambient conditions resulting in a lower COP. Due to the constant warm underground temperatures the COP of the scroll compressor should be constantly high.

¹ <http://www.danfoss.com/products/categories>

Table A-1: Specifications of a Performer SZ370 tandem scroll compressor.

Description	Value
Cooling capacity	84.55kW
Power consumption	16.82kW
COP _{cooling}	5.03

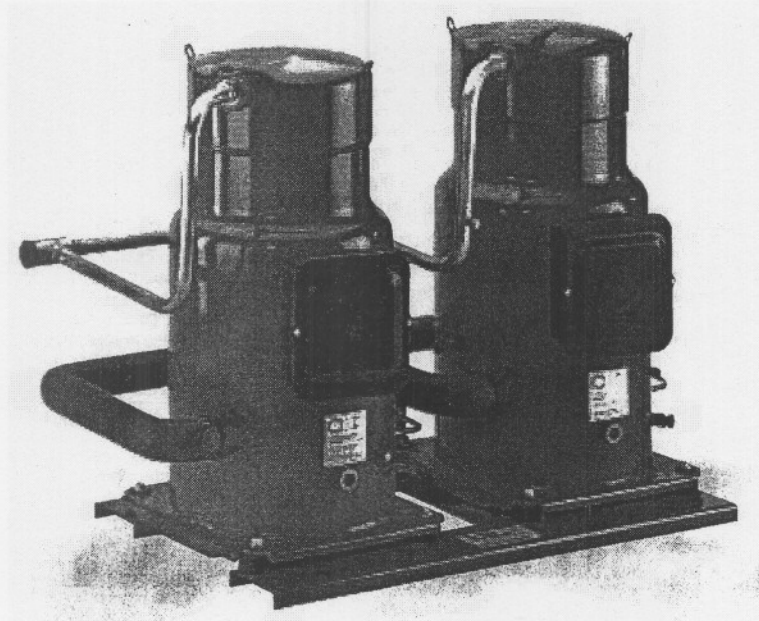


Figure A.4: Tandem configuration.

A.2.2. Evaporator

Direct contact air-to-refrigerant finned heat exchanger technology was selected to cool the air. This was done instead of producing chilled water which can then be circulated through a CWC to cool the air in the conventional manner. The direct contact method increases the positional efficiency of the ACU and reduce any additional losses involved with cooling water and then piping it to a CWC which itself has a low efficiency under certain conditions.

The evaporator design was done for a $15^{\circ}\text{C}_{\text{evap}}$ operating point. This decision was based on the fact that nearly constant high environmental temperature conditions exists underground which will result in the unit operating at an evaporation temperature close to 15°C . The evaporator sizing was done with the software package Recoil Coil Selector Version 1.0.5.0 (Recoil, 2006). The software has the capabilities to simulate evaporator coils with a wide selection of refrigerants. The final sizing of the evaporator was also simulated and confirmed by the manufacturer of the evaporator.

In commercial applications the fin spacing on an evaporator coil is typically 10 to 12 fins per inch, however due to the potential dust content in the underground air, specifically after blasting, the coil

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design was done with only 6 fins per inch to reduce the possibility of the coil blocking on the airside. Due to the width limitation on the ACU, as listed in section A.1, the coil dimensions had to be such that the evaporator could be installed from the top into the ACU. For the coil simulation the parameters listed in Table A-2 were used as specifications.

Table A-2: Coil simulation specifications.

Description	Value
Air side	
Barometer / Altitude	101kPa
Inlet dry bulb temperature	34.0°C
Inlet wet bulb temperature	32.5°C
Coil face velocity	> 2.5m/s
Fin spacing	6 fins/inch
Refrigerant side	
Compressor cooling capacity	84.5kW
Refrigerant	R134a
Evaporation temperature	15.0°C
Physical	
Dimension limitation	Width < 1100mm

After going through a few design iterations the evaporator coil listed in Table A-3 was selected.

Table A-3: Evaporator coil specifications.

Description	Value
Dimensions	
Finned area height	889mm
Finned area length	900mm
Depth	220mm
Rows deep	8
Fins	6 fins/inch
Circuiting	
Tubes high	28
Tube count	224
Circuits	28
Tubes/circuit	8
Tube	
Inside diameter	12.5mm
Arrangement	Staggered
Material	

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Fin	Aluminium
Tube	Copper
Casing	Stainless Steel

Due to the possibility of corrosive gasses in the underground air it was proposed by the manufacturers to apply an epoxy coating over the aluminium fins. Based on the above specifications the performance of the evaporator coil is listed in Table A-4.

Looking back at Figure A.2 it should be noted that the evaporator is installed at an angle of 45° with the distributor and header at the bottom. There are two reasons behind this configuration. Firstly the angle was mainly to optimise the space available for the coil and the path of the air through the coil. Secondly the distributor and header at the bottom are due to the fin direction with regards to the air flow. With this configuration the fins are horizontal with the effect that if dust starts to block the coil it will only happen between two fins instead of always blocking from the bottom if the fins were vertical.

Table A-4: Evaporator coil performance.

Description	Value
Total duty	84.69kW
Sensible duty	21.74kW
Air volume flow rate	2.53m ³ /s
Air face velocity	3.16m/s
Inlet dry bulb temperature	34.0°C
Inlet wet bulb temperature	32.5°C
Outlet dry bulb temperature	26.8°C
Outlet wet bulb temperature	26.7°C
Air pressure drop	147Pa
Refrigerant pressure drop	4.63kPa

A.2.3. Condenser

Based on the compressor cooling capacity of 84.5kW and a power input to the compressor of 16.8kW the condenser had to be sized to transfer at least 100kW from the refrigerant to the water. From experience gained through the design of heat pumps for the commercial sector it was decided to use a fluted tube-in-tube condenser. Such a condenser has a fluted inner tube with a smooth outer tube. The refrigerant flows in the twisted annulus though small channels formed due to the shape of the fluted tube, and the water flows in the inner tube, in a counter flow configuration. Figure A.5 illustrates the geometry of a typical fluted tube heat exchanger.

As stated by Rousseau *et al.* (2003) the refrigerant-to-water heat exchanger for condensing the refrigerant forms an important component of any heat pump or refrigeration cycle. One reason for this is that the overall effectiveness of the cycle is influenced by the effectiveness of heat exchange in the

condenser. It is desirable to keep the temperature difference, ΔT , between the refrigerant and the water in the condenser as small as possible. This leads to a lowering in the ΔT over which heat must be transferred and thus to an increase in the COP.

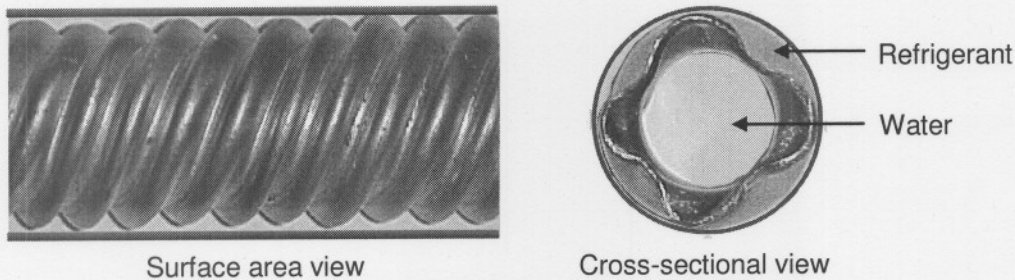


Figure A.5: Fluted tube geometry.

Fluted tube condensers are able to produce high heat transfer coefficients on both sides of the transfer surface by enhancing the flow conditions for both sides. The water heat transfer coefficient is improved by microcirculation of the water without a significant increase in the pressure drop. Heat transfer in the annulus is improved by mainly two phenomena. In the first instance the condensate is drawn towards the corners of the channels by surface tension, clearing the remaining surface in contact with the hot gas. Secondly, the liquid phase experiences some micro-circulation, particularly towards the outlet side, leading to the replacement of cold liquid next to the surface with hotter liquid. If designed correctly this leads to only a relatively small increase in the pressure drop.

In the condenser a counter-flow arrangement is used to obtain the maximum heat transfer where it is possible to achieve outlet water temperatures above the condensing temperature and approaching the superheat temperature of the refrigerant gas.

As stated in section A.2.1 the compressor selection was done at a condensation temperature of 50°C. The reason for this is that the water can be heated to a higher outlet temperature resulting in less water circulated through the ACU. The water outlet temperature is controlled by a water regulating valve reacting to the pressure on the refrigerant side. The water supply is furthermore not equipped with a water pump as the water pressure of the mine supply is high enough to circulate the water through the condenser.

To obtain the required 101.3kW it was necessary to install four Turbotec BTSSC-84 fluted tube condensers in parallel, each with a capacity of about 28kW.

A.2.4. Fan

The ACU was designed to operate with standard mine ventilation fans. For the testing of the prototype the unit was equipped with a 4.0kW fan of which the curve is shown in Figure A.6.



Figure A.7: Final external view of the prototype ACU.

APPENDIX B

UNCERTAINTY ANALYSIS

B. UNCERTAINTY ANALYSIS

In Chapter 3 data is presented for laboratory tests on the prototype ACU. Measurement equipment used to collect the data included an electrical power meter, temperature sensors, a relative humidity sensor, pressure transmitters, and a water flow meter. In this appendix the standard uncertainty contribution of each measuring instrument is given. Based on this the combined standard uncertainty of the parameters calculated using the measurements is presented. All terms and equations used here are based on the guide to the expression of uncertainty in measurement compiled by ISO (1993).

B.1. Specified measurands

B.1.1. Electrical measurement

The electrical input to the compressor was measured using an AC Watt-hour transducer connected to a data logger using a pulse counting module.

B.1.2. Temperature measurement

All temperature measurements were performed using bead type thermistor temperature sensors. For air measurements the sensor is enclosed in an aluminium casing, and for water measurements in a stainless steel casing. All temperature sensors are connected to a data logger using temperature modules.

B.1.3. Relative humidity measurement

The relative humidity of the air was measured using a humidity sensor enclosed in an ABS plastic probe with a duct mounting flange. The humidity sensor is connected to a data logger using a humidity module.

B.1.4. Air flow measurement

The air volume flow rate through the ACU was determined by conducting air velocity pressure measurements. This was done by using low range capacitance type pressure transducers combined with a pitot tube to measure both the static pressure and dynamic pressure.

B.1.5. Water flow measurement

The water volume flow rate was measured by a turbine type flow meter with a 1 litre pulse output. The flow meter was connected to a data logger using a pulse counting module.

B.1.6. Barometric pressure measurement

The barometric pressure was measured with a silicon piezoresistive pressure transducer with integrated signal conditioning electronics and power supply all enclosed within the module connected to the data logger.

B.1.7. Pressure measurement

The water pressure, evaporation pressure and condensation pressure were measured using silicon strain gauge type pressure transducers of a stainless steel construction. The transducers are connected to the data logger using DC input modules.

B.2. Specified parameters

B.2.1. Thermal energy

The thermal energy on the water side is calculated as follows:

$$Q_{water} = \dot{q} \cdot \rho \cdot c_p \cdot (T_{outlet} - T_{inlet}) \quad (\text{B-1})$$

with \dot{q} the measured water volume flow rate through the condenser during the time step with T_{inlet} and T_{outlet} the measured average water temperatures for the time step.

The thermal energy on the air side is calculated as follows:

$$q_{air} = \dot{q} \cdot \rho \cdot (h_{inlet} - h_{outlet}) \quad (\text{B-2})$$

with \dot{q} the measured air volume flow rate through the evaporator during the time step and h_{inlet} and h_{outlet} the calculated average air enthalpies for the time step based on the measured temperatures and relative humidity.

B.2.2. Coefficient of performance

The heating coefficient of performance on the condenser side is calculated as:

$$COP_{heating} = \frac{Q_{water}}{E} \quad (\text{B-3})$$

with Q_{water} the total amount of thermal energy on the water side, and E the total amount of electrical energy consumed by the ACU.

The cooling coefficient of performance on the evaporator side is calculated as:

$$COP_{cooling} = \frac{Q_{air}}{E} \quad (\text{B-4})$$

with $Q_{cooling}$ the total amount of thermal energy on the air side.

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B.3. Identify and quantify uncertainty sources

B.3.1. Instrumentation effects

Table B-1 provides a list of the measuring equipment used and the standard uncertainty related to each component. The uncertainties were determined based on limits ($\pm\alpha$) obtained from the manufacturers supplied without a confidence level. An assumption of a rectangular distribution with a standard deviation of $\alpha/\sqrt{3}$ was used. This assumption is widely applied in industry.

Table B-1: Specified standard uncertainties.

Instrument	Product code	Standard uncertainty
Logging equipment	Architectural Energy Corporation: MicroDataLogger MDL-202	$\pm 0.06\%$
Power measurement	WattNode: WNA-3D-480-P with Magnelab CT's: SCT-0750-050	$\pm 0.29\%$
Temperature measurement	Architectural Energy Corporation: TEMP-02-T4	$\pm 0.12^\circ\text{C}$
	Architectural Energy Corporation: TEMP-02-T5	$\pm 0.12^\circ\text{C}$
Relative humidity measurement	Vaisala: HMD50U	$\pm 1.73\%$
Air flow measurement	Modus Instruments, Inc.: Pressure transmitter T10-06E5	$\pm 0.58\%$
	Modus Instruments, Inc.: Pressure transmitter T10-07E5	$\pm 0.58\%$
Water flow measurement	Kent: M150(KMM) warm water meter. 1 pulse per litre	$\pm 1.16\%$
Barometric pressure measurement	Architectural Energy Corporation: TEMP-02-T5	$\pm 0.87\%$
Pressure measurement	Ashcroft: ASH-K1-7-M02-15-F2 100psig	$\pm 0.58\%$
	Ashcroft: ASH-K1-7-M02-15-F2 150psig	$\pm 0.58\%$
	Ashcroft: ASH-K1-7-M02-15-F2 300psig	$\pm 0.58\%$

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B.4. Combined standard deviation

The most important parameters to consider in all the measured and calculated results presented in Chapter 3 are thermal energy on the condenser side, thermal energy on the evaporator side and electrical energy input to the ACU.

B.4.1. Thermal energy

B.4.1.1. Water side

Thermal energy is calculated using equation (B-1). With specific heat and density fixed, the uncertainty can be obtained from:

$$\frac{u_c(q_{water})}{q_{water}} = \sqrt{\left(\frac{\partial q_{water}}{\partial \dot{q}}\right)^2 u(\dot{q})^2 + \left(\frac{\partial q_{water}}{\partial T_{inlet}}\right)^2 u(T)^2 + \left(\frac{\partial q_{water}}{\partial T_{outlet}}\right)^2 u(T)^2}. \quad (B-5)$$

Calculating the uncertainty of each experimental data set resulted in the following average and maximum uncertainties for the test range.

$$\left. \frac{u_c(q_{water})}{q_{water}} \right|_{avg} = 1.351\%$$

$$\left. \frac{u_c(q_{water})}{q_{water}} \right|_{max} = 1.450\%$$

B.4.1.2. Air side

Thermal energy is calculated using equation (B-2). The uncertainty can be obtained from:

$$\frac{u_c(q_{air})}{q_{air}} = \sqrt{\left(\frac{\partial q_{air}}{\partial p_{air,1}}\right)^2 u(p)^2 + \left(\frac{\partial q_{air}}{\partial p_{air,2}}\right)^2 u(p)^2 + \left(\frac{\partial q_{air}}{\partial p_{bar}}\right)^2 u(p_{bar})^2 + \left(\frac{\partial q_{air}}{\partial T_{inlet}}\right)^2 u(T)^2 + \left(\frac{\partial q_{air}}{\partial T_{outlet}}\right)^2 u(T)^2 + \left(\frac{\partial q_{air}}{\partial rh_{inlet}}\right)^2 u(rh)^2}. \quad (B-6)$$

Calculating the uncertainty of each experimental data set resulted in the following average and maximum uncertainties for the test range.

$$\left. \frac{u_c(q_{air})}{q_{air}} \right|_{avg} = 4.129\%$$

$$\left. \frac{u_c(q_{air})}{q_{air}} \right|_{max} = 6.627\%$$

B.4.2. Electrical energy

The uncertainty associated with the measurement of electrical energy is the standard uncertainty of the power meter (WattNode) and current transformers (CT's). From Table B-1, $\frac{u(E)}{E} = 0.29\%$.

APPENDIX C

EES SIMULATION

C. EES SIMULATION

This Appendix contains an example of one of the EES simulation routines used to simulate the different configurations discussed in Chapters 5 and 6.

C.1. Source code

```
" ===== Simulation of deepening with surface chiller and ACUs ===== "
```

```
"Case study mine intake temperature conditions at different depths"
PROCEDURE AirConditions(Depth: T_wb, T_db)
  T_wb = 0.0045*Depth + 12.962
  T_db = 0.0036*Depth + 20.418
END
```

```
"Geothermal gradient equations ( °C) and Barometric pressures"
PROCEDURE VRT_BARO(Depth, Place$: VRT, Baro)
  If Place$ = 'Witwatersrand' then
    VRT = 18 + 9*Depth/1000
    Baro = 84000 "Pa"
  endif

  If Place$ = 'Carletonville' then
    VRT = 16 + 10.5*Depth/1000
    Baro = 84000
  endif

  If Place$ = 'Klerksdorp' then
    VRT = 22 + 10.5*Depth/1000
    Baro = 86000
  endif

  If Place$ = 'Free State' then
    VRT = 20 + 14.6*Depth/1000
    Baro = 86000
  endif

  If Place$ = 'Bushveld Complex' then
    VRT = 21.5 + 22*Depth/1000
    Baro = 89000
  endif
END
```

```
"Calculate the number of ACUs required to do the required cooling"
FUNCTION NOofACU(Q_cooling_required, Q_cooling_ACU)
  dummy = (Q_cooling_required/Q_cooling_ACU) - trunc(Q_cooling_required/Q_cooling_ACU)
  if dummy >0 then
    NOofACU = trunc(Q_cooling_required/Q_cooling_ACU) + 1
  else
    NOofACU = trunc(Q_cooling_required/Q_cooling_ACU)
  endif
END
```

```
"Characteristics of a single air cooling unit (ACU)"
FUNCTION ACU_TWetBulbOut(T_wb_in, m_dot_air)
  if ((m_dot_air >= 2.6) AND (m_dot_air <= 6.0)) then
    ACU_TWetBulbOut = 1.098*T_wb_in - 8.199
  endif

  if ((m_dot_air >= 2.1) AND (m_dot_air < 2.6)) then
    ACU_TWetBulbOut = 0.8065*T_wb_in - 2.877
  endif
END
```

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```

if ((m_dot_air >= 1.85) AND (m_dot_air < 2.1)) then
  ACU_TWetBulbOut = 1.345*T_wb_in - 18.02
endif
END

FUNCTION ACU_COP(Q_cooling, m_dot_air)
if ((m_dot_air >= 2.6) AND (m_dot_air <= 6.0)) then
  ACU_COP = 0.0914*Q_cooling - 2.926
endif

if ((m_dot_air >= 2.1) AND (m_dot_air < 2.6)) then
  ACU_COP = 0.0793*Q_cooling - 1.8048
endif

if ((m_dot_air >= 1.85) AND (m_dot_air < 2.1)) then
  ACU_COP = 0.0189*Q_cooling + 2.5456
endif
END

FUNCTION Q_ACU(m_dot_air, h_air_in, h_air_out)
  Q = m_dot_air*(h_air_in - h_air_out)
  if Q > 84500 then
    Q_ACU = 84500
  else
    Q_ACU = Q
  endif
END

PROCEDURE ACU(T_wb_in, T_water_in, q_dot_air, P_air, P_water: Q_cooling, P_compressor, m_dot_water,
T_wb_out, T_water_out)
  "Evaporator side"
  T_db_in = T_wb_in
  rho_air_tests = DENSITY(AirH2O, T=T_db_in, P=86500, R=1)
  q_dot_air_tests = 3.045
  m_dot_air_tests = q_dot_air_tests*rho_air_tests
  h_air_in_tests = ENTHALPY(AirH2O, T=T_db_in, P=86500, R=1)
  T_wb_out_tests = ACU_TWetBulbOut(T_wb_in, m_dot_air_tests)
  T_db_out_tests = T_wb_out_tests
  h_air_out_tests = ENTHALPY(AirH2O, T=T_db_out_tests, P=86500, R=1)
  Q_cooling_tests = Q_ACU(m_dot_air_tests, h_air_in_tests, h_air_out_tests)
  rho_air = DENSITY(AirH2O, T=T_db_in, P=P_air, R=1)
  m_dot_air = q_dot_air*rho_air
  h_air_in = ENTHALPY(AirH2O, T=T_db_in, P=P_air, R=1)
  h_air_out = h_air_in - Q_cooling_tests/m_dot_air
  T_wb_out = TEMPERATURE(AirH2O, H=h_air_out, P=P_air, R=1)
  Q_cooling = Q_cooling_tests/1000

  COP = ACU_COP(Q_cooling, m_dot_air)
  P_compressor = Q_cooling/COP

  "Condenser side"
  Q_heating = Q_cooling + P_compressor*0.9
  T_water_out = 45 "deg C"
  cp_water = Specheat(Water, T=T_water_in, P=P_water)/1000
  m_dot_water = Q_heating/(cp_water*(T_water_out - T_water_in))
END

"Heat transfer coefficient of water inside pipe"
FUNCTION WaterHeatTransferCoeff(Re, Pr, k, D_i)
  If Re > 2300 then
    WaterHeatTransferCoeff = k/D_i*(0.023*Re^0.8*Pr^0.4)
  else
    WaterHeatTransferCoeff = 4.36*k/D_i
  endif
END

```

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"Heat transfer coefficient of air over pipe"

```
FUNCTION AirHeatTransferCoeff(Re, Pr, k, D_i)
  AirHeatTransferCoeff = k/D_i*(0.2*Re^0.6)
END
```

"Heat transfer from rock to air in tunnel"

```
FUNCTION HT_RockAir(VRT, T_s, Wall_ID, Length)
  k = 6 "W/m °C quartzite"
  Age = 0.25 "Age in years"
  F_o = 315*Age/Wall_ID^2
  T = 0.685/F_o^0.146 "Age factor"
  HT_RockAir = 2*PI*k*T*(VRT-T_s)*Length "W"
END
```

"Determine the change in air temperature in a horizontal tunnel"

```
MODULE AirTemperatureChange(Wall_ID, Length, T_wb_in, T_db_in, P_air, m_dot_air, VRT, T_db_out, T_wb_out)
```

"Air properties"

```
mu_air = VISCOSITY(AirH2O,T=T_db_in, P=P_air, B=T_wb_in)
k_air = CONDUCTIVITY(AirH2O,T=T_db_in, P=P_air, B=T_wb_in)
cp_air = CP(AirH2O,T=T_db_in, P=P_air, B=T_wb_in)
```

"Air heat transfer calculation"

```
Re_air = 4*m_dot_air/(PI*Wall_ID*mu_air)
Pr_air = cp_air*mu_air/k_air
h_air = WaterHeatTransferCoeff(Re_air, Pr_air, k_air, Wall_ID)
A_inside = PI*Wall_ID*Length
R_inside = 1/(h_air*A_inside)
```

```
q_heat = HT_RockAir(VRT, T_db_in, Wall_ID, Length)
enthalpy_air_in = ENTHALPY(AirH2O,T=T_db_in, P=P_air, B=T_wb_in)
w_air_in = HumRat(AirH2O,T=T_db_in,B=T_wb_in,P=P_air)
q_heat = m_dot_air*(enthalpy_air_out - enthalpy_air_in)
w_air_out = w_air_in
T_wb_out = WetBulb(AirH2O,h=enthalpy_air_out,w=w_air_out,P=P_air)
T_db_out = Temperature(AirH2O,h=enthalpy_air_out,w=w_air_out,P=P_air)
```

END

"Determine the heat transfer between the water and surroundings in a section of pipe"

```
PROCEDURE PipeElementHeatTransfer(direction, Length, Wall_ID, Pipe_ID, WallThickness, T_air_db_out, m_dot_air, T_water_in, P_water_in, m_dot_water: q_elem)
```

"Water properties"

```
mu_water = VISCOSITY(Water,T=T_water_in, P=P_water_in)
k_water = CONDUCTIVITY(Water,T=T_water_in, P=P_water_in)
cp_water = CP(Water,T=T_water_in, P=P_water_in)
```

"Air properties"

```
mu_air = VISCOSITY(Air,T=T_air_db_out)
k_air = CONDUCTIVITY(Air,T=T_air_db_out)
cp_air = CP(Air,T=T_air_db_out)
```

"Inside heat transfer calculation"

```
Re_water = 4*m_dot_water/(PI*Pipe_ID*mu_water)
Pr_water = cp_water*mu_water/k_water
h_inside = WaterHeatTransferCoeff(Re_water, Pr_water, k_water, Pipe_ID)
A_inside = PI*Pipe_ID*Length
R_inside = 1/(h_inside*A_inside)
```

"Conduction components"

```
Pipe_OD = Pipe_ID + 2*WallThickness
k_wall = 45 "W/m °C"
R_wall = ln(Pipe_OD/Pipe_ID)/(2*PI*k_wall*Length)
```

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```

"Outside heat transfer coefficient"
D_h = Wall_ID - Pipe_OD
Re_air = 4*m_dot_air/(PI*D_h*mu_air)
Pr_air = cp_air*mu_air/k_air
h_outside = AirHeatTransferCoeff(Re_air, Pr_air, k_air, D_h)
A_outside = PI*Pipe_OD*Length
R_outside = 1/(h_outside*A_outside)

```

```

UA = 1/(R_inside + R_wall + R_outside)
mcp_min = min(m_dot_water*cp_water, m_dot_air*cp_air)
mcp_max = max(m_dot_water*cp_water, m_dot_air*cp_air)
q_max = mcp_min*(T_air_db_out - T_water_in)
NTU = UA/mcp_min
epsilon = 1-exp(-NTU)
q_element = epsilon * q_max

```

```

if (direction = 0) then
  q_elem = -q_element
else
  q_elem = q_element
endif

```

END

"Energy conservation in a section of pipe"

```

MODULE PipeElementEnergy(config, m_dot_in, enthalpy_in, Depth, W_dot, Q_dot, m_dot_out, Length,
enthalpy_out)

```

"config = 1 for vertical pipe and 0 for horizontal pipe"

```

  m_dot_out*(enthalpy_out + 9.79*(-(Depth+Length))*config) + W_dot = m_dot_in*(enthalpy_in + 9.79*(-
Depth))*config) + Q_dot
END

```

"Solving the outlet conditions of a section of pipe with energy conservation"

```

MODULE PipeElement(config, direction, Depth, Length, Wall_ID, Pipe_ID, PipeThickness, T_air_db_out,
m_dot_air, T_water_in, P_water_in, m_dot_water, T_water_out, P_water_out, Q_heat, W_pump)

```

"Determine heat transfer for pipe element"

```

CALL PipeElementHeatTransfer(direction, Length, Wall_ID, Pipe_ID, PipeThickness, T_air_db_out, m_dot_air,
T_water_in, P_water_in, m_dot_water: Q_heat)

```

```

  enthalpy_water_in = cp_water*T_water_in
  rho_water_in = Density(Water, T=T_water_in, P=P_water_in)
  m_dot_water = m_dot_water_in
  m_dot_water_in = m_dot_water_out
  rho_water_out = Density(Water, T=T_water_out, P=P_water_out)
  CALL PipeElementEnergy(config, m_dot_water_in, enthalpy_water_in, Depth, W_pump, Q_heat,
m_dot_water_out, Length, enthalpy_water_out)
  P_water_out = P_water_in + (rho_water_in+rho_water_out)/2*9.79*Length*config
  cp_water = 4180
  cp_water*T_water_out = enthalpy_water_out
END

```

"Main Program"

```

R$ = 'Carletonville'
Depth[0] = 0 "m"
T_water_in[1] = 5 "deg C"
P_water_in[1] = P_air[0] "Pa"
m_dot_water[1] = m_dot_water_total_supply "kg/s"
Call VRT_BARO(Depth[0], R$:VRT, Baro)
P_air[0] = Baro

```

ACU_CapitalCost = 260000 "R - Purchase cost of one ACU"

ACU_Q_nominal = 80 "kW"

COP_Q_chiller = 5.25

h_fg = 2400 "kJ/kg"

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```

"Shaft info to existing depths"
"Pipes down to chilled water dam"
duplicate i=1,9
  config[i] = Lookup('Shaft Pipe Data', i, 1) "1 for vertical, 0 for horizontal"
  Length[i] = Lookup('Shaft Pipe Data', i, 2) "Length of pipe section [m]"
  Wall_ID[i] = Lookup('Shaft Pipe Data', i, 3) "Inside diameter of shaft/haulage [m]"
  Pipe_ID[i] = Lookup('Shaft Pipe Data', i, 4) "Inside diameter of water pipe section [m]"
  PipeThickness[i] = Lookup('Shaft Pipe Data', i, 5) "Wall thickness of pipe section [m]"
  m_dot_watertolevel[i] = Lookup('Shaft Pipe Data', i, 6) "Water mass flow to each level [kg/s]"
  q_dot_air_shaft[i] = Lookup('Shaft Pipe Data', i, 7) "Air flow down shaft between levels [m^3/s]"
  m_dot_air[i] = q_dot_air_shaft[i]*rho_air[i]
  Depth[i] = Depth[i-1] + config[i]*Length[i] "Depth of outlet of pipe section [m]"
  Call AirConditions(Depth[i]: T_air_wb[i], T_air_db[i])
  rho_air[i] = DENSITY(AirH2O, T=T_air_db[i], P=P_air[i], B=T_air_wb[i])
  P_air[i] = P_air[i-1] + 9.79*rho_air[i]*config[i]*Length[i]
end

P_air[10] = P_air[9]
Depth[10] = Depth[9]

"Pipes from chilled water dam to lower levels"
duplicate i=1,23
  config[i] = Lookup('Shaft Pipe Data', i, 1) "1 for vertical, 0 for horizontal"
  Length[i] = Lookup('Shaft Pipe Data', i, 2) "Length of pipe section [m]"
  Wall_ID[i] = Lookup('Shaft Pipe Data', i, 3) "Inside diameter of shaft/haulage [m]"
  Pipe_ID[i] = Lookup('Shaft Pipe Data', i, 4) "Inside diameter of water pipe section [m]"
  PipeThickness[i] = Lookup('Shaft Pipe Data', i, 5) "Wall thickness of pipe section [m]"
  m_dot_watertolevel[i] = Lookup('Shaft Pipe Data', i, 6) "Water mass flow to each level [kg/s]"
  q_dot_air_shaft[i] = Lookup('Shaft Pipe Data', i, 7) "Air flow down shaft between levels [m^3/s]"
  m_dot_air[i] = q_dot_air_shaft[i]*rho_air[i]
  Depth[i] = Depth[i-1] + config[i]*Length[i] "Depth of inlet of pipe section [m]"
  Call AirConditions(Depth[i]: T_air_wb[i], T_air_db[i])
  rho_air[i] = DENSITY(AirH2O, T=T_air_db[i], P=P_air[i], B=T_air_wb[i])
  P_air[i] = P_air[i-1] + 9.79*rho_air[i]*config[i]*Length[i]
end

m_dot_water_total_supply = sum(m_dot_watertolevel[1..9]) + sum(m_dot_watertolevel[11..23]) +
m_dot_watertolevel[49] + m_dot_water_ACUtolevel[49]

"Shaft pipes down to chilled water dam"
duplicate i=1,9
  Call PipeElement(config[i], 1, Depth[i], Length[i], Wall_ID[i], Pipe_ID[i], PipeThickness[i], T_air_db[i],
  m_dot_air[i], T_water_in[i], P_water_in[i], m_dot_water[i], T_water_out[i], P_water_out[i], Q_heat[i], 0)
  T_water_out[i] = T_water_in[i+1]
  P_water_out[i] = P_water_in[i+1]
  m_dot_water[i+1] = m_dot_water[i] - m_dot_watertolevel[i]
end

"Chilled water dam"
m_dot_in_chilldam = m_dot_water[10]
T_in_chilldam = T_water_in[10]
T_out_chilldam = T_in_chilldam
m_dot_out_chilldam = m_dot_in_chilldam
m_dot_out_chilldam = m_dot_water[11] + m_dot_water[49]
T_out_chilldam = T_water_in[11]
T_out_chilldam = T_water_in[49]
P_out_chilldam = P_air[10]
P_out_chilldam = P_water_in[11]
P_out_chilldam = P_water_in[49]

"Shaft pipes from chilled water dam to lower levels"
duplicate i=1,23
  Call PipeElement(config[i], 1, Depth[i], Length[i], Wall_ID[i], Pipe_ID[i], PipeThickness[i], T_air_db[i],
  m_dot_air[i], T_water_in[i], P_water_in[i], m_dot_water[i], T_water_out[i], P_water_out[i], Q_heat[i], 0)

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T_water_out[i] = T_water_in[i+1]
P_water_out[i] = P_water_in[i+1]
m_dot_water[i+1] = m_dot_water[i] - m_dot_watertolevel[i]
end

"Heat load per level to hot dam"
duplicate i=11,23
  Q_load[i] = m_dot_watertolevel[i]*4180*(T_in_hotdam - T_water_out[i])
end

Q_total_load = sum(Q_load[11..23])

"Shaft pipes from chilled water dam to deepening"
config[49] = Lookup('Shaft Pipe Data', 49, 1) "1 for vertical, 0 for horizontal"
Length[49] = Lookup('Shaft Pipe Data', 49, 2) "Length of pipe section [m]"
Wall_ID[49] = Lookup('Shaft Pipe Data', 49, 3) "Inside diameter of shaft/haulage [m]"
Pipe_ID[49] = Lookup('Shaft Pipe Data', 49, 4) "Inside diameter of water pipe section [m]"
PipeThickness[49] = Lookup('Shaft Pipe Data', 49, 5) "Wall thickness of pipe section [m]"
m_dot_watertolevel[49] = Lookup('Shaft Pipe Data', 49, 6) "Water mass flow to each level [kg/s]"
q_dot_air_shaft[49] = Lookup('Shaft Pipe Data', 49, 7) "Air flow down shaft between levels [m^3/s]"
m_dot_air[49] = q_dot_air_shaft[49]*rho_air[49]
Depth[49] = Depth[10] + config[49]*Length[49] "Depth of outlet of pipe section [m]"
Call AirConditions(Depth[49]: T_air_wb[49], T_air_db[49])
rho_air[49] = DENSITY(AirH2O,T=T_air_db[49],P=P_air[49],B=T_air_wb[49])
P_air[49] = P_air[9] + 9.79*rho_air[49]*config[49]*Length[49]

m_dot_water[49] = m_dot_watertolevel[49] + m_dot_water_ACUtolevel[49]

Call PipeElement(config[49], 1, Depth[49], Length[49], Wall_ID[49], Pipe_ID[49], PipeThickness[49],
T_air_db[49], m_dot_air[49], T_water_in[49], P_water_in[49], m_dot_water[49], T_water_out[49],
P_water_out[49], Q_heat[49], 0)

Q_chiltdown_deepening = Q_heat[49]

"Tunnel on deepening to cooling cars"
"First distance into each level"

n = 51
config[n] = Lookup('Shaft Pipe Data', n, 1) "1 for vertical, 0 for horizontal"
Length[n] = Lookup('Shaft Pipe Data', n, 2) "Length of pipe section [m]"
Wall_ID[n] = Lookup('Shaft Pipe Data', n, 3) "Inside diameter of shaft/haulage [m]"
Pipe_ID[n] = Lookup('Shaft Pipe Data', n, 4) "Inside diameter of water pipe section [m]"
PipeThickness[n] = Lookup('Shaft Pipe Data', n, 5) "Wall thickness of pipe section [m]"
m_dot_water[n] = m_dot_water[n-2]
T_water_in[n] = T_water_out[n-2]
P_water_in[n] = P_water_out[n-2]
m_dot_air[n] = m_dot_air[n-2]
P_air[n] = P_air[n-2]
Depth[n] = Depth[n-2]
Call AirConditions(Depth[n]: T_air_wb[n], T_air_db[n])
rho_air[n] = DENSITY(AirH2O,T=T_air_db[n],P=P_air[n],B=T_air_wb[n])
Call VRT_BARO(Depth[n], R$:VRT[n], dummy[n])
Call AirTemperatureChange(Wall_ID[n], Length[n], T_air_wb[n], T_air_db[n], P_air[n], m_dot_air[n], VRT[n],
T_db_out[n], T_wb_out[n])
Call PipeElement(config[n], 1, Depth[n], Length[n], Wall_ID[n], Pipe_ID[n], PipeThickness[n], T_air_db[n],
m_dot_air[n], T_water_in[n], P_water_in[n], m_dot_water[n], T_water_out[n], P_water_out[n], Q_heat[n], 0)

"Extended distances on each level from end of first distance"
duplicate n = 51, 52
  m_dot_water[n+1] = m_dot_water_ACUtolevel[49]/2
  T_water_in[n+1] = T_water_out[n]
  P_water_in[n+1] = P_water_out[n]
  m_dot_air[n+1] = m_dot_air[n]
  P_air[n+1] = P_air[n]
  Depth[n+1] = Depth[n]

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T_air_wb[n+1] = T_wb_out[n]
T_air_db[n+1] = T_db_out[n]
rho_air[n+1] = DENSITY(AirH2O, T=T_air_db[n+1], P=P_air[n+1], B=T_air_wb[n+1])
VRT[n+1] = VRT[n]
Call AirTemperatureChange(Wall_ID[51], Length[51], T_air_wb[n+1], T_air_db[n+1], P_air[n+1],
m_dot_air[n+1], VRT[n+1], T_db_out[n+1], T_wb_out[n+1])
Call PipeElement(config[51], 1, Depth[n+1], Length[51], Wall_ID[51], Pipe_ID[51], PipeThickness[51],
T_air_db[n+1], m_dot_air[n+1], T_water_in[n+1], P_water_in[n+1], m_dot_water[n+1], T_water_out[n+1],
P_water_out[n+1], Q_heat[n+1], 0)
end

Q_leveltoACU = sum(Q_heat[51..53])

"Cooling requirements per level"
"Deepening"
Q_cooling_required[49] = 1000 "kW"

duplicate n=51,51
Call ACU(T_wb_out[n], T_water_out[n], 2.6, P_air[n], P_water_out[n]: Q_cooling_ACU[n],
P_compressor_ACU[n], m_dot_water_ACU[n], T_wb_out_ACU[n], T_water_out_ACU[n])
Call ACU(T_wb_out[n+1], T_water_out[n+1], 2.6, P_air[n+1], P_water_out[n+1]: Q_cooling_ACU[n+1],
P_compressor_ACU[n+1], m_dot_water_ACU[n+1], T_wb_out_ACU[n+1], T_water_out_ACU[n+1])
Call ACU(T_wb_out[n+2], T_water_out[n+2], 2.6, P_air[n+2], P_water_out[n+2]: Q_cooling_ACU[n+2],
P_compressor_ACU[n+2], m_dot_water_ACU[n+2], T_wb_out_ACU[n+2], T_water_out_ACU[n+2])

m_dot_water_ACUtolevel[n-2] = Q_cooling_required[49]*massflowPerQcooling[n]

No_ACU[n] = NOofACU(Q_cooling_required[n-2]/3, Q_cooling_ACU[n])
No_ACU[n+1] = NOofACU(Q_cooling_required[n-2]/3, Q_cooling_ACU[n+1])
No_ACU[n+2] = NOofACU(Q_cooling_required[n-2]/3, Q_cooling_ACU[n+2])

Q_cooling_actual[n] = No_ACU[n]*Q_cooling_ACU[n] + No_ACU[n+1]*Q_cooling_ACU[n+1]
+ No_ACU[n+2]*Q_cooling_ACU[n+2]
P_comp_actual[n] = No_ACU[n]*P_compressor_ACU[n] + No_ACU[n+1]*P_compressor_ACU[n+1]
+ No_ACU[n+2]*P_compressor_ACU[n+2]
QcoolPerQnominal[n] = Q_cooling_actual[n]/(ACU_Q_nominal*(No_ACU[n] + No_ACU[n+1] + No_ACU[n+2]))
massflowPerQcooling[n] = (No_ACU[n]*m_dot_water_ACU[n] + No_ACU[n+1]*m_dot_water_ACU[n+1] +
No_ACU[n+2]*m_dot_water_ACU[n+2])/Q_cooling_actual[n]

end

Q_cooling_ACUs = Q_cooling_required[49]*1000
P_comp_ACUs = P_comp_actual[51]*1000/Q_cooling_actual[51]*Q_cooling_required[49]

"Additional heat load of deepening"
Q_load_deepening = m_dot_watertolevel[49]*4180*(28 - T_water_out[49])

"Deepening dam"
m_dot_out_deepdam = m_dot_watertolevel[49] + m_dot_water_ACUtolevel[49]
T_out_deepdam*m_dot_out_deepdam = 45*m_dot_water_ACUtolevel[49] + 28*m_dot_watertolevel[49]

"Pipeline between deepening dam and hot water dam"
m_dot_water[55] = m_dot_out_deepdam
W_deeppumping = length[55]*m_dot_water[55]*9.79/0.75
W_deeppumping = m_dot_water[55]*4180*(T_out_deeppump - T_out_deepdam)
T_water_out[55] = T_out_deeppump
P_deeppump_in = P_air[55]
P_deeppump_out = P_deeppump_in + (W_deeppumping*0.75)/(m_dot_water[55]/1000)
P_water_out[55] = P_deeppump_out

config[55] = Lookup('Shaft Pipe Data', 55, 1) "1 for vertical, 0 for horizontal"
Length[55] = Lookup('Shaft Pipe Data', 55, 2) "Length of pipe section [m]"
Wall_ID[55] = Lookup('Shaft Pipe Data', 55, 3) "inside diameter of shaft/haulage [m]"
Pipe_ID[55] = Lookup('Shaft Pipe Data', 55, 4) "inside diameter of water pipe section [m]"
PipeThickness[55] = Lookup('Shaft Pipe Data', 55, 5) "Wall thickness of pipe section [m]"

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q_dot_air_shaft[55] = Lookup('Shaft Pipe Data', 55, 7) "Air flow down shaft between levels [m^3/s]"
m_dot_air[55] = q_dot_air_shaft[55]*rho_air[55]
Depth[55] = Depth[47] + config[55]*Length[55] "Depth of inlet of pipe section [m]"
Call AirConditions(Depth[55]: T_air_wb[55], T_air_db[55])
rho_air[55] = DENSITY(AirH2O, T=T_air_db[55], P=P_air[47], B=T_air_wb[55])
P_air[55] = P_air[47] + 9.79*rho_air[55]*config[55]*Length[55]

Call PipeElement(config[55], 0, Depth[55], Length[55], Wall_ID[55], Pipe_ID[55], PipeThickness[55], T_air_db[55],
m_dot_air[55], T_water_in[55], P_water_in[55], m_dot_water[55], T_water_out[55], P_water_out[55],
Q_up_deepening, 0)

T_in_deeptodam = T_water_in[55]

"Hot water dam"
T_in_hotdam = 28 "°C"
m_dot_in_hotdam = sum(m_dot_watertolevel[11..23])
m_dot_out_hotdam = m_dot_in_hotdam + m_dot_water[55]
m_dot_in_hotdam*T_in_hotdam + m_dot_water[55]*T_in_deeptodam = m_dot_out_hotdam*T_out_hotdam

"Return pipeline to surface from hot dam"
P_air[24] = P_air[0] "Pa"
Depth[24] = Depth[0] "m"

duplicate i=25,47
  config[i] = Lookup('Shaft Pipe Data', i, 1) "1 for vertical, 0 for horizontal"
  Length[i] = Lookup('Shaft Pipe Data', i, 2) "Length of pipe section [m]"
  Wall_ID[i] = Lookup('Shaft Pipe Data', i, 3) "Inside diameter of shaft/haulage [m]"
  Pipe_ID[i] = Lookup('Shaft Pipe Data', i, 4) "Inside diameter of water pipe section [m]"
  PipeThickness[i] = Lookup('Shaft Pipe Data', i, 5) "Wall thickness of pipe section [m]"
  q_dot_air_shaft[i] = Lookup('Shaft Pipe Data', i, 7) "Air flow down shaft between levels [m^3/s]"
  m_dot_air[i] = q_dot_air_shaft[i]*rho_air[i]
  Depth[i] = Depth[i-1] + config[i]*Length[i] "Depth of inlet of pipe section [m]"
  Call AirConditions(Depth[i]: T_air_wb[i], T_air_db[i])
  rho_air[i] = DENSITY(AirH2O, T=T_air_db[i], P=P_air[i-1], B=T_air_wb[i])
  P_air[i] = P_air[i-1] + 9.79*rho_air[i]*config[i]*Length[i]
end

duplicate i=25,46
  Call PipeElement(config[i], 0, Depth[i], Length[i], Wall_ID[i], Pipe_ID[i], PipeThickness[i], T_air_db[i],
m_dot_air[i], T_water_in[i], P_water_in[i], m_dot_water[i], T_water_out[i], P_water_out[i], Q_heat[i], 0)
  P_water_out[i] = P_water_in[i+1]
  T_water_out[i] = T_water_in[i+1]
  m_dot_water[i] = m_dot_water[i+1]
end

i = 47
m_dot_water[i] = m_dot_out_hotdam
W_pumping = (sum(length[25..47]) - length[32])*m_dot_water[25]*9.79/0.75
W_pumping = m_dot_water[25]*4180*(T_out_pump - T_out_hotdam)
T_water_out[i] = T_out_pump
P_pump_in = P_air[47] - 38370
P_pump_out = P_pump_in + (W_pumping*0.75)/(m_dot_water[25]/1000)
P_water_out[i] = P_pump_out

Call PipeElement(config[i], 0, Depth[i], Length[i], Wall_ID[i], Pipe_ID[i], PipeThickness[i], T_air_db[i], m_dot_air[i],
T_water_in[i], P_water_in[i], m_dot_water[i], T_water_out[i], P_water_out[i], Q_heat[i], 0)

"Heat gain down"
Q_gain_chilldown = sum(Q_heat[1..9])
Q_gain_damdown = sum(Q_heat[11..23])

"Heat loss up"
Q_losses_up = sum(Q_heat[25..47])

"Pumping to surface"

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$$P_{\text{pump_total}} = W_{\text{pumping}} + W_{\text{deeppumping}}$$

"Surface refrigeration installation"

$$Q_{\text{surface}} = m_{\text{dot_water_total_supply}} * 4180 * (T_{\text{water_in}[1]} - T_{\text{water_in}[25]})$$

$$P_{\text{surface}} = Q_{\text{surface}} / \text{COP}_{\text{Q_chiller}}$$

$$Q_{\text{heat_surface}} = Q_{\text{surface}} + P_{\text{surface}} * 0.9$$

C.2. Simulation results

APPENDIX C: EES SIMULATION

C-10

configID	Depth[m]	Length[m]	m_dot_water[m_dot_water]	Pipe_ID[m]	P_water_in[m]	T_water_in[T]	Wm_ID[m]	P_Air[m]	T_Air[m]	rho_Air[kg/m^3]	m_dot_water[m_dot_water]	q_dot_water[q_dot_water]	P_water_out[P]	Q_west[Q_west]	T_water_out[T]	O_load[O_load]
0	0	0	[kg/s]	[m]	[m]	[Pa]	[m]	[Pa]	[degC]	[kg/m^3]	[kg/s]	[kg/s]	[Pa]	[kW]	[degC]	[W]
1	6.75	676	1766	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	1578	702	1880	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	1920	342	1945	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	1988	78	1921	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2118	120	1936	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2166	48	1756	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2238	72	1551	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2238	400	1181	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2274	36	1146	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2274	66	1110	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2340	66	1051	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2472	66	898	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2564	96	943.8	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2684	96	870.3	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2760	96	727.7	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2856	96	596.3	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2952	96	501.5	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3018	66	436.3	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3072	54	312.5	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3192	120	225.5	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3312	120	176.6	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3432	120	125.1	0.008	0.4	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
0	0	0	0	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	676	876	1587	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	1578	702	1720	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	1920	342	1838	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	1988	78	1869	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2118	120	1844	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2166	48	1714	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2238	72	1510	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2238	400	1159	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2274	36	1129	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2274	66	1094	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2406	66	1024	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2472	66	920.8	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2568	96	844.5	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2664	96	766.1	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2856	96	589.8	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	2952	96	486.5	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3018	66	424.5	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3072	54	318.6	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3192	120	218.1	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3312	120	170.8	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3432	120	121.8	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	3474	42	122.5	0.008	0.6	84000	8.35	33235	23.57	16.9	1.077	0	8.85E+06	424730	7.281	833759
1	4000	1726	66.98	0.008	0.4	109396	11.22	10	134269	34.82	30.96	1.472	1.71E+07	70866	16.83	528E+06
0	4000	150	66.98	0.008	0.4	109396	11.22	10	134269	34.82	30.96	1.472	1.71E+07	70866	16.83	528E+06
0	4000	54	66.98	0.008	0.4	109396	11.22	10	134269	34.82	30.96	1.472	1.71E+07	70866	16.83	528E+06
0	4000	5.4	66.98	0.008	0.4	109396	11.22	10	134269	34.82	30.96	1.472	1.71E+07	70866	16.83	528E+06
1	4000	526	60.83	0.008	0.4	175626	45.21	10	129255	34.82	30.96	1.337	5.28E+06	46.64	1.71E+07	70866

configID	T_db_out[degC]	T_wb_out[degC]	VRIT[degC]	Q_cooling[kW]	m_dot_water[m_dot_water]	P_compressor[kW]	Q_cooling_Actual[kW]	T_water_out[degC]	T_wb_out[degC]	m_dot_water[m_dot_water]	m_dot_water_Actual[m_dot_water]	Q_cooling_Actual[kW]	Q_cooling_Actual[kW]	mass_flow_ParCooling[kg/s]
0	35.64	31.22	58	1080	83.19	17.79	17.79	45	25.66	0.8079	5	10.67	1.045	
0	36.81	31.48	58	1080	83.19	17.72	17.72	45	25.92	0.8042	4	10.67	1.045	
0	37.74	31.59	58	1080	84.11	17.68	17.68	45	26.17	0.8032	4	10.67	1.045	

An investigation into the DSM and energy efficiency potential of a modular underground air cooling unit applied in the South African mining industry.
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