Development and Testing of an Underground Remote Refrigeration Plant

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ABSTRACT

The objective of this study was to develop, install, and test a small underground mobile refrigeration plant (M.R.P.) to deal with some of the real problems associated with mine cooling in an operating mine.

The requirement for cooling the underground environment is discussed with particular emphasis on the need for this method of cooling, with the concomitant benefits.

The research investigated current methods of cooling and reasons for previous failures in (M.R.P.). Both static and dynamic simulations were conducted to increase the confidence level under operating conditions.

Implementation and testing, resulted in "lessons learnt" requiring modifications, which are documented. Actual results have been recorded. These results have proved that significant cooling via (M.R.P) is feasible.

Main benefits include positional efficiency, cost per kilowatt of cooling and cooling opportunities for remote area's of a mine.

Finally, a proven technology is now available for large-scale implementation into the mining industry. Now the ventilation engineer has another system of cooling, which can be utilized in the quest to create an occupational environment, which meets the physical and mental health requirements of the worker.

SAMEVATTING

Die doel van die studie was om 'n klein mobiele verkoelingsaanleg (M.V.A.) te ontwikkel, te installeer en te toets om van die praktiese probleme verbonde aan myn verkoeling ondergrond in 'n werende myn aan te spreek.

Die behoefte vir verkoeling van die ondergrondse omgewing word bespreek met spesifieke verwysing na die behoefte vir sodanige verkoeling en die voortspruitende voordele.

Die navorsing het bestaande verkoelings metodes en redes waarom sodanige metodes onsuksesvol was ondersoek. Beide statiese en dinamiese simulasies is uitgevoer om die vertroue vlak onder bedrys kondisies te vehoog.

Implimentering en toetsing het waardevolle inligting aan die dag gebring wat wysigings tot gevolg gehad het wat gedokumenteer is. Werklike resultate is aangeteken. Hierdie resultate toon dat noemenswaardige verkoeling met (M.V.A.) haalbaar is.

Van die noemenswaardige voordele sluit posisionele doeltreffendheid, koste per kilowatt van verkoeling en vekoelings moontlikhede vir afgeleë warm gebiede van myne in.

Uiteindelik is bewysde tegnologie beskikbaar vir grootskaalse implimentering in die mynbedryf. Die ventilasie ingenieur het nou 'n addisionele verkoelingsstelsel tot sy beskikking wat aangewend kan word in die strewe om werksomstandighede te skep wat beide die fisiese en geestes behoeftes van die werker bevredig.

ACKNOWLEDGEMENTS

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Finally, to God the Father of our Lord Jesus Christ be all the glory, honor and praise, for the wisdom and knowledge bestowed upon me. For without Him I can do nothing.

NOMENCLATURE

cp	Specific heat capacity [kJ/kg]
m	Mass flow [kg/s]
q	Heat or power [kW]
T	Temperature [°C]
Q	Quantity [m/s]

The following subscripts are also used:

a	Air
c	Condenser
cin	Entering condenser
comp	Compressor
cout	Leaving condenser
e	Evaporator
ein	Entering evaporator
eout	Leaving evaporator
p	Constant press
dt	Delta temperature
wb	Wet bulb temperature
db	Dry bulb temperature
kPa	Kilopascals
m^3/s	Cubic meters per second
kg/s	Kilograms per second
kg/m^3	Kilograms per cubic meter
kW/°C	Kilowatt per degree centigrade
ℓ/s	Liters per second
°C	Degree centigrade
pa	Per annum
%	Percentage

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Chapter 1
INTRODUCTION

In this chapter, a brief background of this study, the problem definition and the structure of the thesis are described.

1.1 Background

Mining has been the backbone to the development of South Africa. This was the core initiator of industry, which was initially aimed at meeting the needs of the Mining Environment. Even though the importance of mining has diminished somewhat, due to the establishment of "other" industries mining still contributes about 11% of GDP and provides employment for nearly 750 000 men. These men support 3 million independents. Mining output accounts for 55% of foreign exchange earnings [1].

The mining at deeper levels over time, with the correlating increase in temperatures due to the inherent virgin rock temperatures (VRT), has necessitated cooling the occupational environment. This cooling is necessary to ensure the environment meets the mental and physical needs of the workers, thereby ensuring a safe productive working environment[2,3,4]. The expanding nature of mines requires a constant need for upgrading of cooling systems. The cooling is usually done by using chilled water to cool the air through the use of cooling cars, large bulk air coolers and spray chambers[5]. The chilled water is normally produced by large surface cooling plants. The use of large thermal storage dams normally ensures that there is spare capacity of chilled water[6]. The total infrastructure cost required for a new or extension installation is very high. This results in under cooling in many instances.

1.2 Objectives of Project

1.2.1 Problem Statement (For Mining Industry)

One of the major problems associated with deep level mining is the high ambient temperature. Existing cooling systems and infrastructure normally cater for global cooling requirements. Individual and mostly remote requirements have remained the challenge to the engineer.

The main objective of this project was to prove the concept of remote cooling, as expounded below. Remote cooling in this instance means a small self-contained underground installation, which can be installed in outlying (remote) areas of a mine. Industry ventilation engineers, although fully aware of the benefits, were of the opinion that it was not feasible due to the previous problem of rejecting heat via drains or water spray systems.

1.3 Contributions of this Study

The main contribution to this study was the decision to develop, install, and test a small underground mobile refrigeration plant (MRP) to deal with some of the real problems associated with mine cooling in an operating mine.

This entailed analyzing

- why these units failed in the past
- scoping the work required
- motivating for the funds
- and project managing the engineering, design and installation.

During this process, a static design was carried out. This was vetted and supported by a dynamic design, thereby ensuring the integrity and decreasing the risk of the design.

This entire report is focused on the practical resolution of a "real-life" problem. It is also purposely written in layman's terms, for the benefit of aspirant ventilation practitioners.

1.4 Structure of Thesis

The overall objective statement for this project is the following:

Establish if a mobile underground cooling plant is technically feasible. If positive, design, build and commission.

From the above objective statement the following project steps were established:

- Derive a suitable system of heat rejection.
- Find a suitable mobile refrigeration unit.
- Provide a conceptual design for the cooling unit for the specific application.
- Review the static design information to check for errors.
- Simulate the system's performance by means of a dynamic approach.
- Check which plant configuration will give the best performance.
- Design, build, commission and test.
- Modify for improvements.
- Report.

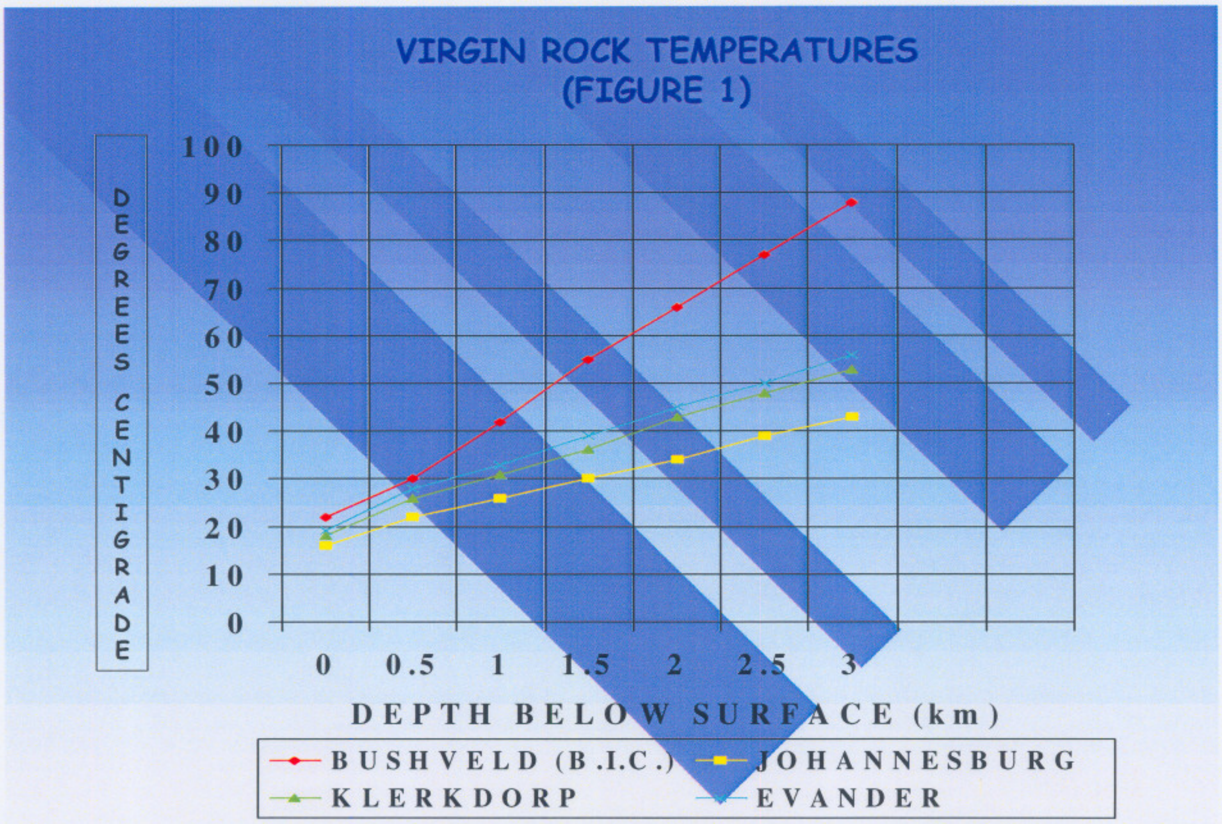
Chapter 2
ANALYSIS OF THE NEED FOR UNDERGROUND COOLING

In this chapter the following topics will be expounded on which will demonstrate the need for underground cooling, namely virgin rock temperatures, human tolerances, union and associations demands, corporate reporting and governance, the business case, motivation of people, and HIV Aids.

2.1 Geothermal Gradient

The exploitation of the reef body was initially achieved by developing footwall inclines from surface outcrops. These inclines were later replaced by first-generation vertical shafts and finally by second-generation shafts. Later, sub-inclines or sub-shafts[7] augmented the inclines. Naturally, with the increase in depth, there is a correlated increase in the VRT[8]. This correlation is depicted in Figure 2.1 where the effect of the specific strata can also be clearly seen.

Figure 2.1: Virgin rock temperature



The challenges presented by the gradual increase in average working depths are especially significant in the area of mine environmental control. The single dominant challenge in this context is the heat problem, which is aggravated at depth, mostly, but not exclusively, due to increased virgin rock temperatures. The ultimate future of mining at great depth will increasingly depend on the industry's ability to contend in an acceptable and cost-effective manner with the environmental control problems related to the provision of satisfactory ventilation and cooling[9].

2.2 Human Tolerances

Circumstantial evidence suggests that human beings evolved as tropical animals: we possess a well-developed sweat mechanism and our skin is practically devoid of insulative hair. In addition, our thermoregulatory system has a greater reserve of heat elimination than of heat conservation capacity indicating that heat presented a greater threat to our ancestors than cold[10].

Air-cooling and ventilation are needed in deep underground mines to minimise the stress associated with heat[11,12]. One of the primary aetiologies of heat stroke is excessive environmental heat loads[13]. Table 2.1 summarises actions required for different ranges of temperature used within the underground mining environment.

The general conclusion with regard to the management of heat stress is that there is no precedent that dictates a uniform approach to the problem[14]. The absolute solution would however be to cool the underground environment to below 27,5°C wb.

Table 2.1: Actions required at various temperatures

Temperature Range	**Interpretation**	**Action**
wb > 32.5°C db > 37.0°C	Unacceptable risk of heat disorders for routine work	No routine work allowed
wb > 25.0°C < 32.5°C db < 37.0°C	Environment conducive to heat disorders	Routine work may take place if a mandatory Heat Stress Management system is in place

Source: Guild R. (2000)[15]

2.3 Unions

The occupational environment in which members of the various unions work, is fast becoming a major area of contention and debate. The occupational environment has a direct bearing on the future long-term health of employees, and as such, employees are not willing to sacrifice a future quality of life if they are not compensated accordingly.

With specific regard to temperatures, the heat tolerance screening (HTS) procedures are seen as inhumane, thus driving the change to cool down the occupational environment to a level that poses no risk (wb < 27.5°C). This would negate the need for HTS and acclimatisation and follow the universal principles of Zero Tolerance Target Zero (OTTO).

2.4 Corporate Reporting

With most mining conglomerates now listed on international stock exchanges, the importance of corporate governance has become increasingly important. Furthermore, in keeping with the vision, mission and values of most responsible companies, there is a direct connotation to the health and safety of the workforce's working environment and contaminant exposures. This has put further pressure on the need for ensuring an occupational working environment, which meets the expectations of first-world standards. These standards, however, exceed the productivity requirement, thereby demanding a move to supply comfort conditions. Hence, the demand for innovative ventilation and cooling strategies, which are cost effective, will remain the order of the day.

2.5 Business Case

There is a clear business case to ensure an occupational environment that meets the physical and mental health requirements of the workers. Responses to heat stress and the outcome of such exposures vary from individual to individual. Moreover, the reasons why people act and perform adversely in heat stress zones are often not understood.

Studies have revealed that at low heat stress levels the main signs are behavioural changes, such as depression, aggression, irritability and numerous psychological problems. This results in a loss of concentration, and a decrease in efficiency in both mental and skilled tasks.

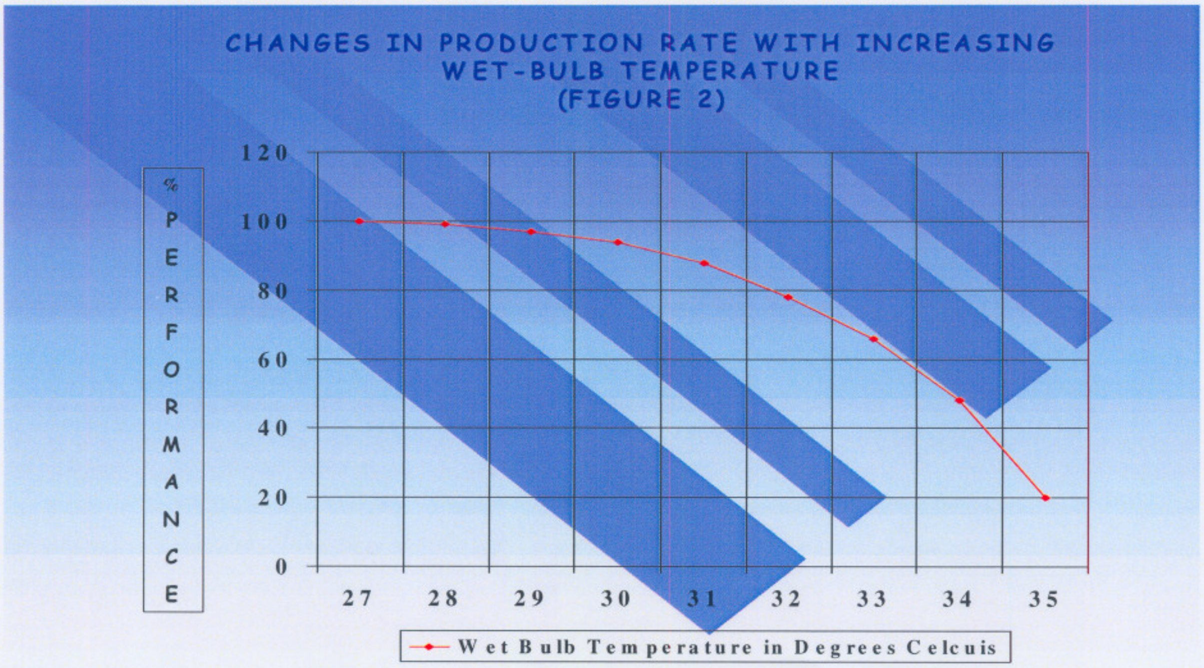
At mild to moderate heat stress, the Index of Heat Stress includes "subtle to substantial decrements" in tasks involving intellectual input, dexterity and alertness.

At higher levels of heat stress, the impact appears to become "physiological" in nature, with a progressive decrease in physical working capacity and, ultimately, the development of heat disorders[16].

Of importance is that heat stress adversely affects mental performance much sooner than any deterioration occurs in physical working capacity.

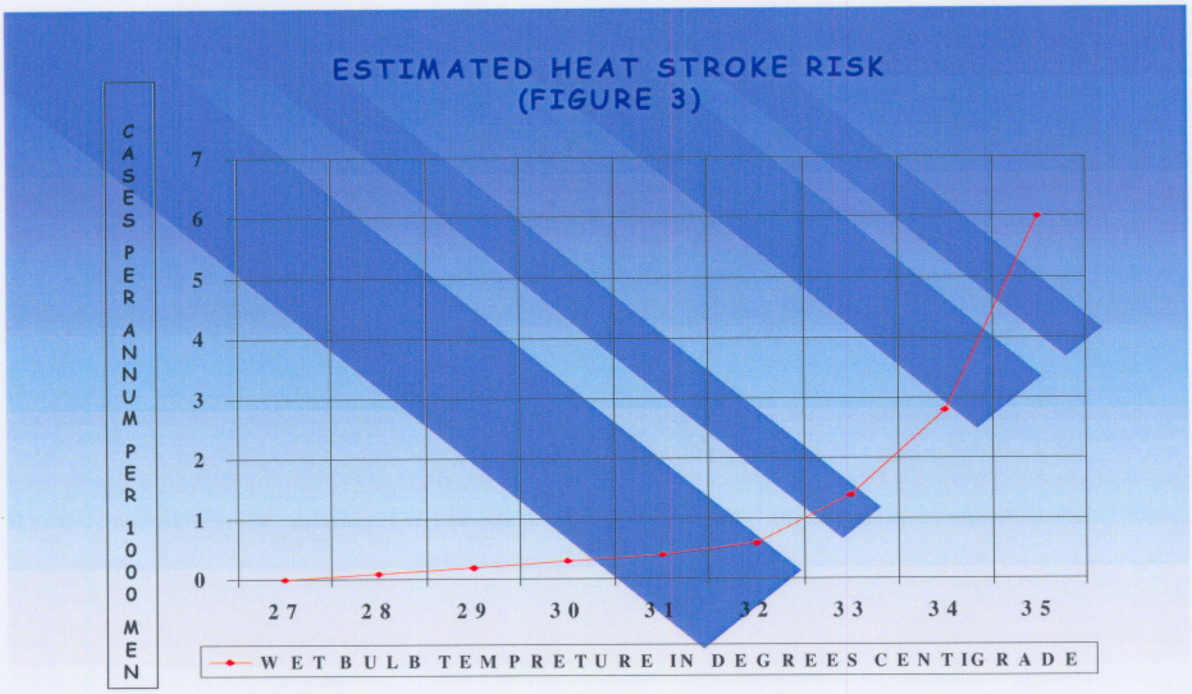
A large number of observations made by the Human Sciences Laboratory of the Chamber of Mines of South Africa have resulted in two graphs that clearly indicate that the thermal environment is directly linked to productivity and safety[17]. (See Figures 2.2 and 2.3)

Figure 2.2: Changes in production rate



Source: Le Roux (1990)[18]

Figure 2.3: Estimated heat stroke risk



Source: Le Roux (1990)[19]

It is therefore deduced that the consequences of heat stress can be expressed in terms of safety, health[20] and production outcomes.

2.6 Motivation of People

Much work has been done on creating a motivating climate in order to unleash the full human potential of the workforce. The mine environment is no different to an office environment. The continual drive to come down on the cost curve is growing in intensity. This has become one of the biggest competitive advantages a mining house can achieve. Therefore, every avenue to realise this potential must be pursued with vigor.

Work carried out by Professor Wyndham at the Chamber of Mines during the 1970s, clearly recognised the deleterious effect of heat on mine labour efficiency[21]. Coupled to this, the harsh occupational environment found underground in terms of light, noise, dust, and cramped surrounds, is a recipe for low morale and motivation. Of all these exposures, heat has the biggest single effect on motivation and performance.

2.8 Benefits

Available ore reserves at certain levels would become "locked up" if cooling were not applied. Figure 2.4 indicates isothermals, determined from the Environ[26] software package, showing the typical cut off dependent on the cooling required. These isothermals were calculated for a sister mine, but used here to indicate the significance of cooling requirements.

The bottom line being that if no cooling is applied, no mining can take place beyond the 29.0°C wet-bulb temperature limit (blue area in this specific mine).

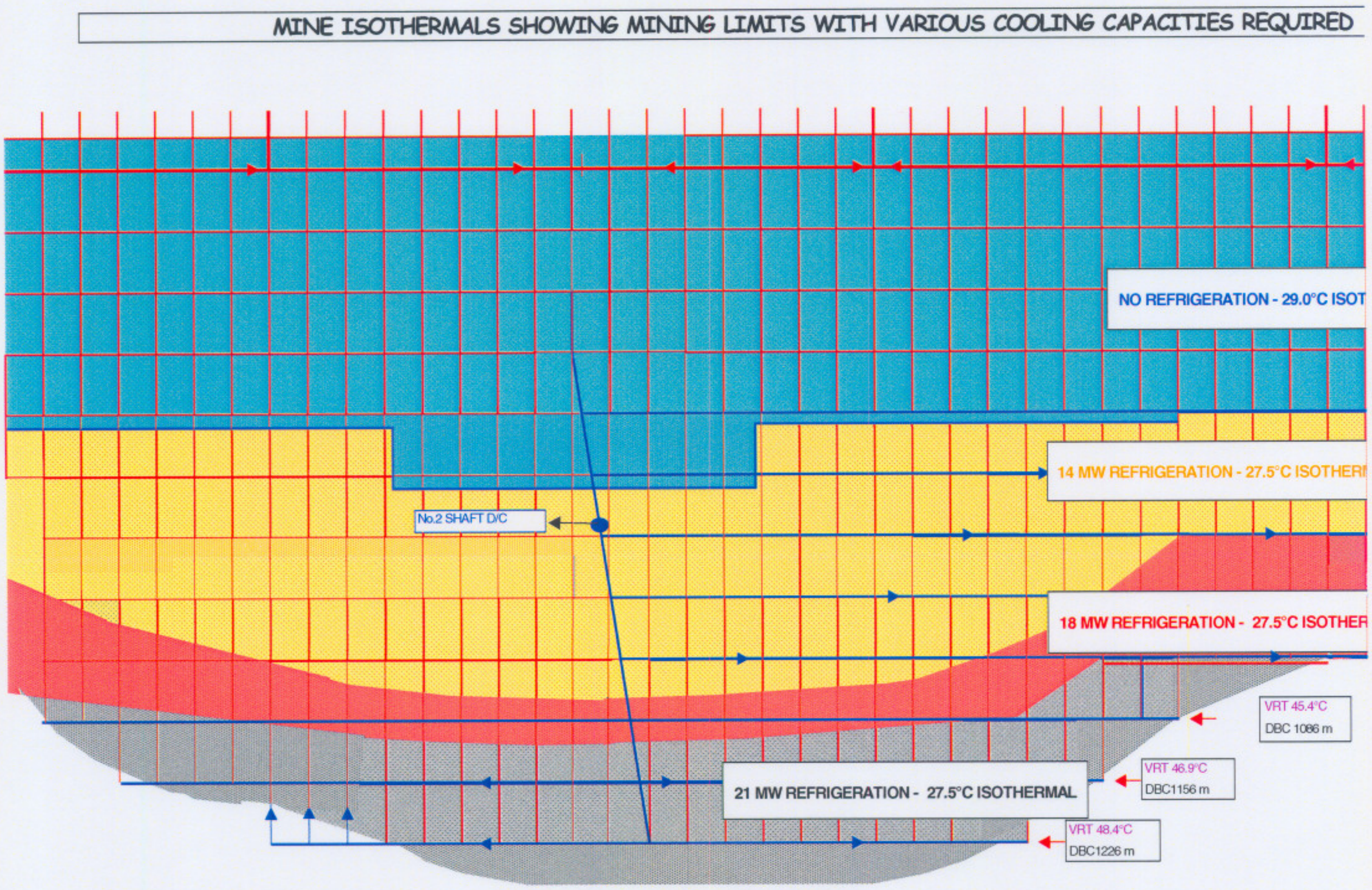


Figure 2.4: Mine isothermals

To further illustrate this benefit, the following financial analysis shows the cost benefit in "unlocking" the ore reserve for mining purposes. The justification of the capital expenditure involves demonstrating whether the refrigeration plant adds sufficient value to cover its own capital costs as well as showing an adequate return on capital employed. This is done by assessing how much of the ore reserve can be mined without cooling, before the maximum thermal exposure to man is reached. This is then compared with the cost of installing cooling with the consequent increase of mineable ore reserve. This is done by doing a discounted cash flow (DCF) analysis of the various options.

Two assumptions with actual calculated results are presented.

Assumption 1

Install a 21 MW refrigeration plant thus allowing mining operations to continue as planned.

Assumption 2

Dispensing with the proposed refrigeration plant and continue mining operations until the 29,0°C isothermal limit is reached.

A (DCF) analysis was done for the options under consideration. The results are tabulated below and are expressed in millions of rands.

Table 2.2: Discounted cash flow analysis

Real discount rate	**8% pa***	**10% pa***	**12% pa***
Assumption 1. Install refrigeration plant as planned	3 526	3 048	2 674
Assumption 2. Continue mining without refrigeration plant	1 982	1 820	1 683
Value added	1 544	1 228	991

* Depicts the Nett Present Value (NPV) at real discount rates

The above tabulation clearly demonstrates the financial viability of installing the refrigeration plant, thereby "unlocking" additional ore reserves for mining purposes.

2.7 HIV Aids

The aids pandemic is another area that is going to impact severely on mine cooling strategies.

Heat stress limits used in the South African mining industry, for unacclimatised as well as acclimatised individuals, are based on the probability to develop hyperthermia (dangerously high body core temperatures). An acceptable risk in this context is that a worker may experience no more than a 10^{-6} (one-in-a-million chance) risk of developing a body temperature above 40ºC[22].

The development of these heat stress limits was based on the physiological responses (body temperature, heart rate and sweat rate) of **medically fit individuals.** (i.e. they have successfully completed a medical examination to assess fitness for work in heat).

Fever is a complex physiologic response to infection or injury[23]. Individuals with elevated body temperatures, for example as a result of fever, will rapidly develop dangerously high body temperatures when working in heat and as such be regarded as being heat intolerant[24]. This heat intolerance could be temporary for cases where fever is associated with ailments like common colds, for instance, but has the potential to become permanent in cases where fever is common and ongoing, such as among HIV-infected individuals[25].

In view of the above, it is obvious that the heat stress limits originally designed for medically fit individuals, **are no longer valid** for individuals with underlying infections resulting in an elevated body temperature. The latter group will be at a higher and unacceptable risk of developing hyperthermia. For known reasons it is difficult to identify individuals with infections associated with fever, and the only way to ensure that such individuals are not exposed to an unacceptable risk of developing heat stroke and related heat disorders, is to **lower the environmental temperature limits.** Thus, the current set limits in terms of reject temperatures (maximum temperature) will have to be lowered. This will put further demand on localised cooling strategies to cater for the increased cooling demand.

Chapter 3
REVIEW OF PRESENT APPROACHES TO MINE COOLING

In this chapter, the various current cooling strategies available for the ventilation engineer will be discussed.

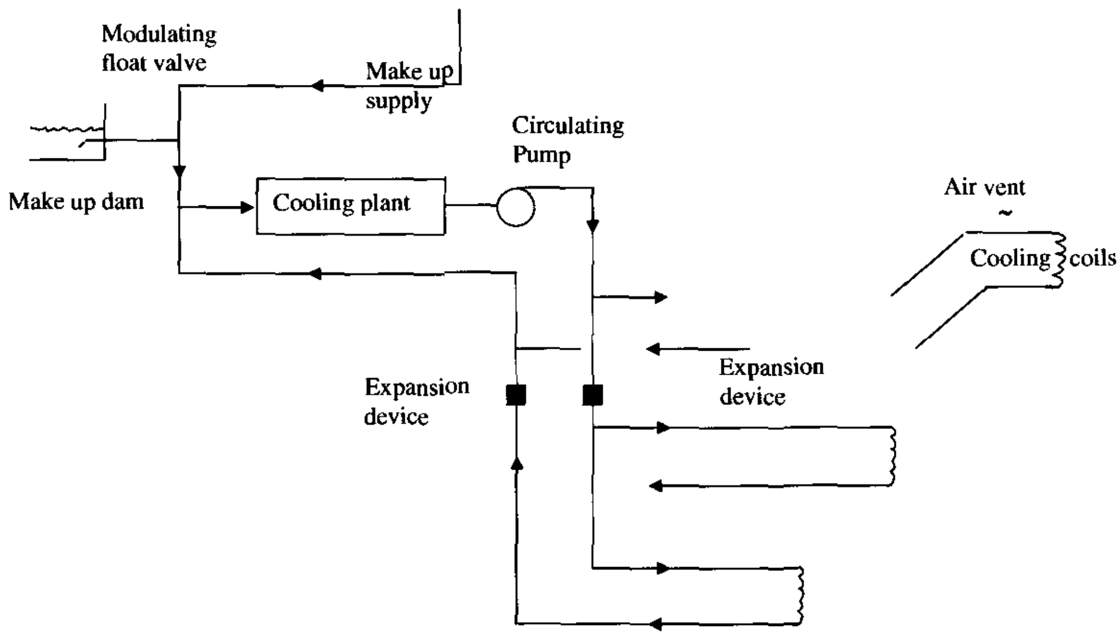
3.1 Cooling Strategies

In order to facilitate greater understanding Calculations used in coil car selection of the various methods of cooling available to the Ventilation Engineer, the following schematics depict some of the generic systems currently in use, and are classified as either open or closed circuits.

Closed Circuit

In this system water is pumped from the refrigeration plant to strategically located cooling coils. The heated water is returned directly from the coils back to the plant. The reticulation is totally sealed (pipe) and theoretically no make-up water is required. A typical layout is illustrated below.

Figure 3.1: Closed circuit schematic

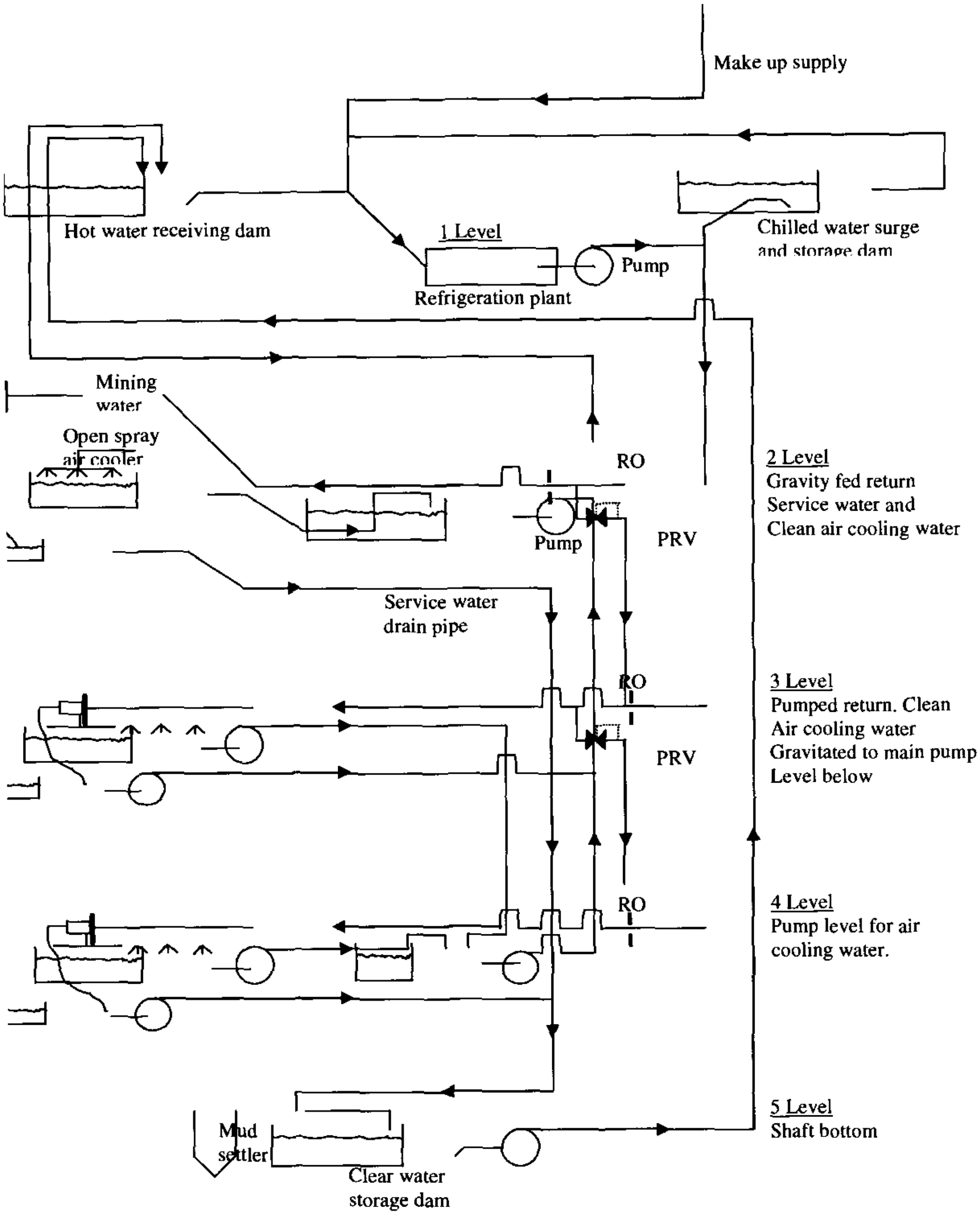


Open Circuit

Similar to the diagram above except that the cooling units are open spray systems and not coils hence water return into drains to a sump or pump station and then pumped back to the plant.

In this system water is gravity fed in a pipe system to the respective open spray type coolers. The hot water then flows down a pipe or annex holes to the settlers and clear water dam. The water is then finally pumped back to surface. A typical layout is illustrated below.

Figure 3.2: Open circuit schematic



The primary source of mine cooling is by normal ventilation air. Temperatures have however increased due to the geothermal gradient, which increases with depth, as well as the auto compression heat added per vertical meter of depth

This has resulted in the generic strategies discussed below, which are used in the following configuration, depending on the actual application/need. The prime reason for this configuration is the cost of cooling, the cost of surface bulk air-cooling being the cheapest[27].

- Surface bulk air-cooling
- Underground station bulk air coolers
- Chilled service water
- Crosscut coolers
- The use of ice

Recently, due to the ever-increasing depth of mining and the correlated temperature rise and flow, the necessity has arisen to add an additional cooling strategy, namely in-stope cooling.

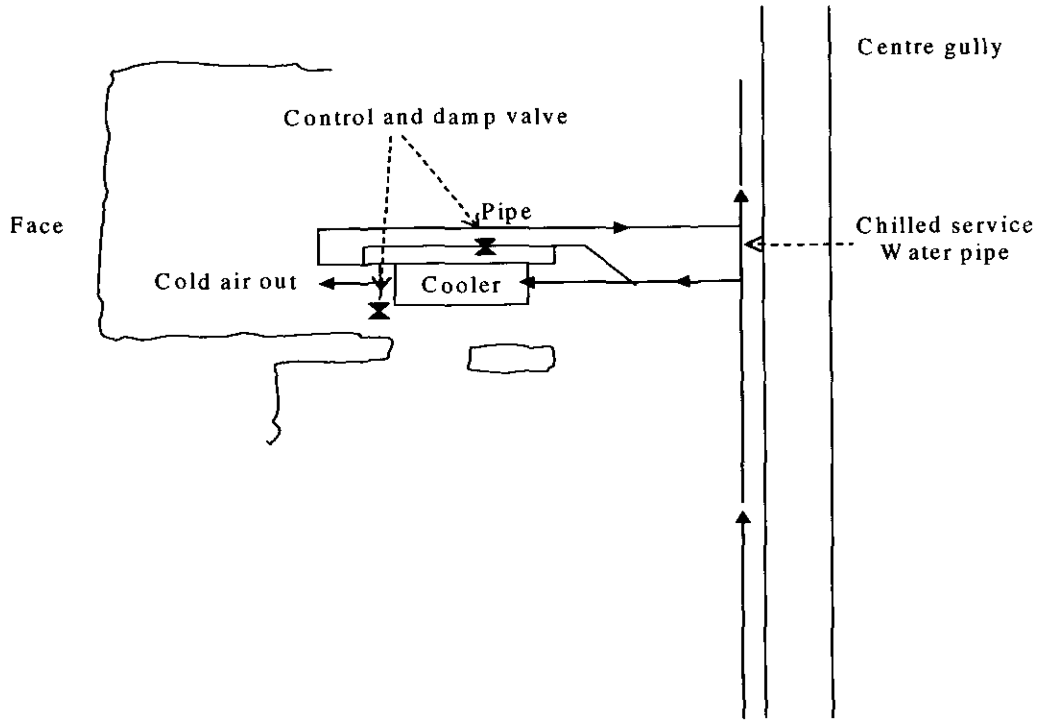
3.2 In-stope Cooling

Personal calculations using Environ[26] (a detailed load calculation computer program developed by the CSIR for mine cooling calculations) show that in high thermal environments, subsequent cooling has to be taken into the working face. This has necessitated the installation of secondary cooling devices within the production environment.

It is evident that refrigeration technology has progressed dramatically over the last decade. When account is taken of the increase in capacities required and the enormous costs of operating systems, then the importance of further improvements is clear[28]. Alternatively, as stated by a refrigeration leader of our time. "There is far too little understanding of the general problem of refrigeration application and distribution."[29]

A typical layout showing a simplified control system is depicted below.

Figure 3.3: In stope distribution schematic



3.3 Surface Bulk Air-Cooling

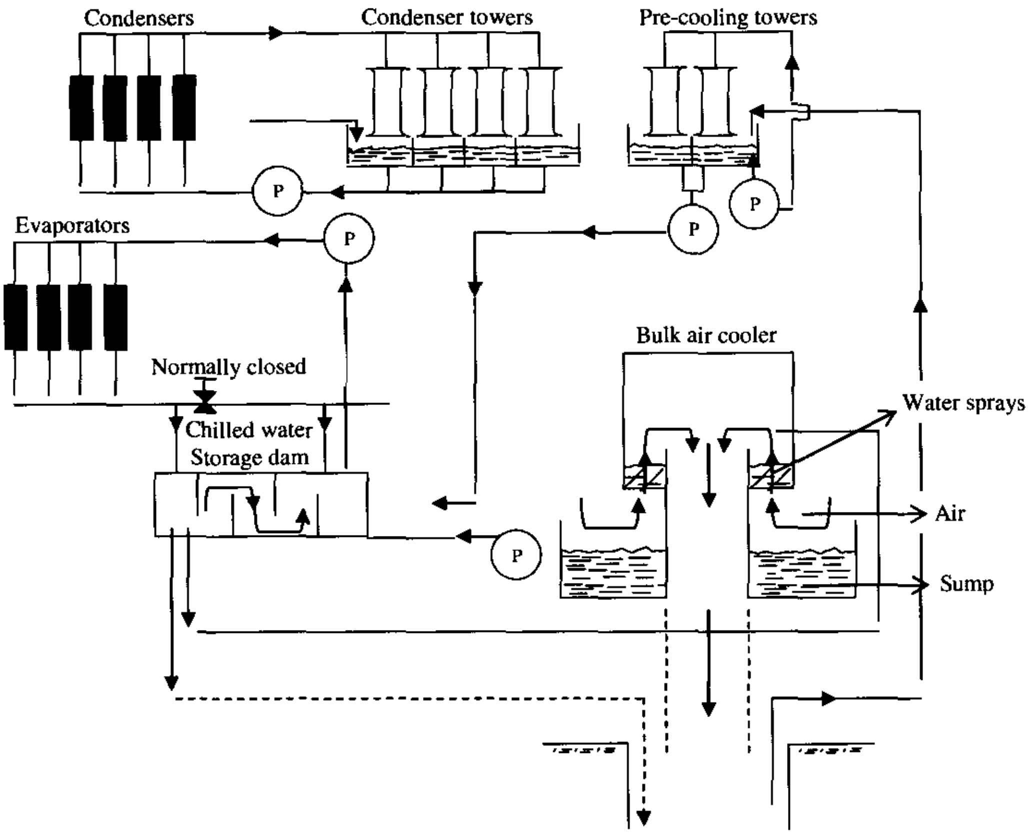
This has traditionally been used solely for cooling the shafts and haulages where people travel or work, to an acceptable limit, and had no direct bearing on the in-stope cooling requirement. It did however assist somewhat in reducing the approaching temperature to the secondary or tertiary cooling[30].

The above statement is only true though for deep level mines where the VRT and auto-compression play a major role in the heat flow equation.

Within the shallower mines with high VRT, like the platinum mines within the Bushveld Igneous Complex, the heat pick-up due to auto-compression plays a lesser role due to the relatively shorter shafts even though steeper VRT gradients are experienced. In these instances, depending on the installed capacity, there remains sufficient residual cooling to be used within the stope horizon. Therefore, once the initial thermal inertia has been overcome, adequate cooling via surface bulk air-cooling, within the reef plane can be achieved as a "first phase" cooling strategy.

This cooling is the cheapest and easiest application of refrigeration, but suffers from poor positional efficiency. A typical layout is depicted below.

Figure 3.4: Surface bulk air cooler schematic



3.4 Underground Bulk Air Coolers

Underground bulk air coolers are introduced when the temperature in the shaft is acceptable but the temperatures in the haulages, which lead to the working places, are not. These coolers at placed at strategic sites to cool the ventilation air whenever the air temperature increases above the maximum design value[31] .

This form of cooling is then designed to ensure that the thermal environment, where people have to travel and work on the infrastructure leading to the working places, is acceptable. The thermal duties of these units are typically between 0,5 and 20 MW[32].

This form of cooling results in a better positional efficiency than surface cooling, and because of the warmer conditions, the plant tends to run at full load throughout the year[27].

3.5 Cross-cut Coolers

Cross-cut coolers augment the previous cooling strategies to ensure that the air entering the actual working places are in line with accepted norms.

This results in even a better positional efficiency, as the water passes through a heat exchanger (which resembles the radiator of a car). Air is blown through the cooler by means of a small fan. The air is cooled and delivered directly to the working face[27].

3.6 Chilled Service Water

The cooling of service water plays an important role in the overall cooling strategy of mines, since the chilled service water is distributed to the working faces where it is traditionally difficult to install and maintain air coolers[27].

This form of cooling results in the best positional efficiency, but suffers from not being continuous[33]. (Cooling only occurs when water is used, and this is intermittent.)

3.7 The Use of Ice

Ice in this context is used to reduce the temperature of the underground service water in mines, and has some distinct advantages[34,35]. Experimental results from a pilot installation for conveying ice in pipelines down a mine is well documented and constitutes no problem[36]. In terms of its usefulness underground, the primary feature of this system is the heat exchange during the latent heat of fusion of ice, which is 335 kJ/kg. Simply put, if ice at –5°C is used to chill water underground to 4°C, the water involved is about one quarter of the total circulating underground[37]. This relates to further savings in terms of the consequent reduction in pumping requirements[38].

3.8 Tabular summary of cooling strategies

The following table is a summary of the cooling strategies discussed, showing the main advantages and disadvantages of each system, which must be taken into consideration when designing a cooling system.

Table 3.9 Summary of cooling strategies

Cooling Strategies	Advantages	Disadvantages
Closed Circuit	• Low pumping cost • Water returns to plant in sealed pipe (U-tube effect)	• High capital cost of installing high pressure pipes • Safety sensitive due to high pressure
Open Circuit	• Cheaper capital cost of installation	• High pumping cost • Water returning to pump station in drains
Surface Bulk Air Cooling	• Cheapest and easiest application of refrigeration • Ease of maintenance • Heat rejection not a problem	• Poor positional efficiency
Underground Bulk Air Cooling	• Increased positional efficiency to the above	• Maintenance more difficult • Capital cost of excavation • Increased pumping cost
Cross Cut Coolers	• Increased positional efficiency to the above	• Limited cooling capacity • Maintenance intensive
In Stope Cooling	• Increased positional efficiency to the above	• Limited cooling capacity • Requires frequent moving
Chilled Service Water	• Best positional efficiency - Cooling right at working face	• Cooling not continuous
The Use of Ice	• Reduction in pumping cost	• High capital cost

Chapter 4
MOTIVATING THE NEED FOR THE MOBILE REFRIGERATION PLANT

In this chapter, the motivation in terms of the required application will be discussed.

4.1 Contribution of this Study

The contribution of the remote underground cooler is manifold, in terms of costs and positional efficiency. However, the overriding benefits within an underground mining environment can be stated as:

- End-of-mine-life scenario
- Delay major installations
- Effective control of hot spots

4.2 End-of-Mine-Life Scenario

As mining progresses, the resident VRT will determine the extent to which one could mine prior to installing cooling. The remote underground cooler can be effectively utilised to cool that fraction of the working face, thereby negating the installation of a major surface-cooling infrastructure. This in effect increases the mine-able ore reserve.

4.3 Delay Major Installations

Traditionally, the installation of cooling takes place when the underground thermal parameters can no longer be maintained through conventional means, namely removal of heat by air volume circulation.

This necessitates major capital expenditure for the installation of cooling. Unfortunately, the cooling must be introduced to cater for the next 10 to 15 years, requiring over-installation for the initial years. In today's terms, this cost for surface bulk-air cooler and underground-refrigerated service water is approximately R7 000.00 per kW of cooling.

Any delay in this expenditure has a direct positive financial benefit to the bottom line of the organisation. Thus, refrigeration is always installed "rather later than sooner". In this instance the MRP can be used to cool those working areas requiring immediate cooling and in many instances can delay major installations for 2 to 3 years.

4.4 Effective Control of Hot Spots

The tendency in all mines is to have a proportion of underground working places which are too hot[39]. A typical example will be a cluster of development ends that requires **series**-type ventilation **(air from one development end is used in series to ventilate subsequent ends).**

This poses a major problem to the ventilation engineer and results in limiting the number of working ends. With the availability of the MRP, these areas can be adequately cooled to meet the production demand without incurring excessive costs.

From the above, the benefits of the MRP are clearly distilled, resulting in a strong motivation for the system.

Chapter 5
TECHNICAL STUDIES

In this chapter, the reasons for past failures are addressed and the technical studies to increase the level of confidence in terms of the coil heat exchanger duties, are discussed.

5.1 Background to Study

MRP is not a totally new concept, as the idea was attempted over two decades ago albeit with inferior equipment and poor design capabilities.

In order not to "re-invent" the wheel or make the same mistakes as in the past, an analysis was done which highlighted the following main causes of failure:

The original design was based on using spray-type chambers to cool the high condensing temperatures. One of the main problems with this configuration was the introduction of contamination into the system with the correlating high maintenance requirements of the spray chambers. A further problem was the reliability and poor technical abilities of the compressors to handle the high operating temperatures they were subjected to.

The efficiency of chillers is limited by the dissipation from their principle components: compressor and heat exchanges at the condenser and evaporator[40]. Simply put, the previous attempts in MRP failures can primarily be attributed to high condensing temperatures.

This was caused due to failure to provide adequate heat rejection facilities. This anomaly resulted in high oil temperatures causing separation of the oil molecules, which contributed to premature compressor failure. The high condensing temperatures, which previously caused separation of the oil molecules, can now be easily accommodated by state-of-the-art single screw compressors. These compressors offer advantages under certain conditions[41] and can operate at a higher upper limit of their temperature envelope, namely 50°C.

If the above reasons for the demise of such a system are analysed, the re-implementation of MRPs can easily be justified. This is due to the technical advancement of compressors and the ability to utilise coil heat exchangers on the condenser circuit, instead of the problematic spray heat rejection system.

5.2 Technical Studies

Technical studies and simulations were conducted to ensure the integrity of the duty of coil heat exchangers. This was deemed necessary to ensure adequate heat transfer capacity in a closed loop layout. These cooling coils were to be used for both the evaporator and condenser circuit in closed loop format. These results were then modified to reflect the duty at site densities.

This data supplied below is a summary of performance tests carried out in the Heat Exchanger Test Centre of the CSIR (Miningtek) on a coil heat exchanger manufactured by Joules Technology (Pty) Ltd.

The objective of these tests was to determine the heat rejection capacity of the coil heat exchanger at the design conditions. The tests were conducted on surface at temperatures and flow rates that would normally be experienced underground. By using accepted methods, it was possible to predict the performance at underground barometric pressures. The design specifications are depicted in Table 5.1.

Table 5.1: Design specifications of the heat exchanger

Air inlet temperature	29,9 (°C)
Water inlet temperature	50,0 (°C)
Barometric pressure	103,0 kPa
Airflow rate	10, 9 m³/s
Water flow rate	8,0 ℓ/s

Results from the CSIR (Miningtek) laboratory are tabulated below. The resultant values have been expressed in terms of K[(1)]. Table 5.2 also shows a summary of the expected performance of the cooling car at the underground barometric pressure of 103 kPa.

Table 5.2: Results of the heat transfer tests

Water Flow rate (l/s)	Water Temp in (°C)	Water Temp out (°C)	Air Flow rate (m³/s)	Air mass flow rate (kg/s)	Dry bulb temperature in (°C)	Wet-bulb temperature out* (°C)	Dry bulb temperature out (°C)	Duty (kW)	K-factor (kW/°C)
6,0	39.5	36.9	7,0	6,7	25.5	20,5	36.7	65.6	4.5
6,0	39.6	36.8	9,0	8,6	26,2	20.0	36.0	70.8	5.3
6.1	40.2	36.9	11.4	10.8	25.7	20.0	35.8	83.4	5.8
8,1	39.2	37.3	7,0	6,7	25.1	20.3	36.7	65.9	4.7
8,0	39.6	37.7	9,0	8,6	27,2	21.2	36.7	66.7	5.4
8,2	39.9	37.5	11.4	10.8	25.9	20.1	35.7	84.0	6.0
10,3	39.0	37.4	7,0	6.7	25.0	20.4	36.6	69.2	4.9
10,2	39.6	38.1	9,0	8.6	27,8	22.2	37.0	63.5	5.5
10,0	39.7	38.0	10.8	10.3	28.1	21,9	36.6	70.1	6.1

Source: CSIR (Miningtek)

The duty of the cooling car depends on the air mass flow rate rather than directly on the volume flow rate.

Table 5.3 signifies the expected performance at underground conditions.

Table 5.3: Performance at underground conditions

Air volume flow	10,9 m^3/s
Air mass flow	12,4 kg/s
Air density	1,1 kg/m^3
K value	6,4 kW/°C
Inlet air temperature	29,9 °C (wb)
Barometric pressure	103 kPa
Water flow	8,0 ℓ/s
Inlet water temperature	50,0 °C
Duty*	128,0 kW

*Measured data: modified for underground barometric pressure.

Chapter 6
STATIC DESIGN

In this chapter, the static design will be discussed.

6.1 Design Brief

The design brief was to provide a means of producing 500 kW of cooling at a location situated in close proximity to the workplace. The advantages of this are self-evident as positional efficiency is high when measured between duty produced and duty in the workplace.

6.2 Variations

During the initial feasibility study, a number of sites were identified for the pilot mobile underground refrigeration plant (MRP) project. Cooling of the environment was to be achieved conventionally, i.e. using closed loop air/water coil type heat exchangers. Heat rejection could be achieved by three means:

a) Closed loop water-cooled plate type heat exchangers
b) Closed loop air-cooled coil type heat exchangers
c) Horizontal or vertical spray type cooling towers

The last option (c) was not considered since the idea was to make the unit mobile in the sense that it could be relocated later if required. Open circuit cooling also poses the problem of contamination of the water circulating through the plant.

Closed loop water-cooled plate type heat exchangers

This option requires the availability of a source of water of suitable quality and temperature for heat rejection. Hot water from the plant condenser is circulated through a plate-type heat exchanger in a closed loop. Heat is transferred into cooler water, also circulating through the heat exchanger in an open loop, supplied from an alternate source for this purpose. An ideal situation is a site in close proximity to a clean water, settling dam of large enough volume to ensure there is no heat pick-up from the return water from the heat exchangers, and also to ensure that sufficient time is allowed for settling. This suggests that an amount of water at least equal to or greater than that required for heat rejection, continuously replaces the water pumped from the dam up to surface. The treatment of the water is almost a prerequisite since sediment content is usually high in this environment, and the temperature and chemical content of the water will cause scaling in the heat exchanger – on the open circuit side. It must be noted that the cooling water could be circulated directly through the plant's condenser to achieve the same result. This does however present a high risk of failure of the overall system since it is easier to protect (filtration and treatment), maintain and/or replace a relatively inexpensive heat exchanger than it is for a refrigeration plant.

Closed loop air-cooled coil type heat exchangers

This option uses air drawn over the coil to cool hot water circulating inside the tubes – similar to a car radiator. The main consideration is the temperature and volume of air available. The advantage of this, and any closed loop system, is that the water circulating through the unit is not contaminated and hence only requires initial treatment when charging the system. The primary downside to this method is fouling on the airside due to dust load, and external corrosion due to blasting fumes.

After due consideration option (b) was chosen as:

- It would make the system mobile.
- It would pose the least operating problem.
- No large source of water was available.

6.3 Site Selection

The final site was selected based on a number of considerations:

- Its availability; no additional excavation required as an existing battery bay was to be utilised for plant and evaporator coils
- Proximity to disused raise through which the hot water piping could be installed for the heat rejection coils
- No suitable site was located that allowed the water-cooled option.

6.4 Heat Rejection Constraints

From the outset, there were certain constraints that had to be contended with namely:

- Volume of air available for heat rejection - approximately 40 kg/s

The air temperatures in the return airway (RAW) were already high (26,8/29,9°C) which meant that to achieve a suitable dT for heat rejection, condensing temperatures had to be high. Physical constraints were also placed on the size of the coils, which had to fit into the cages for transportation – this in turn affected the available heat transfer area as well as water and airflow rates through each coil.

6.5 Calculations used in determining the solution

Calculations used in coil car selection:

In mining air cooler equations are based on sigma energies.

$Q = Ma\ (Sai - Sao) = Mw\ Cpw\ (Two - Twi)$

Q = Duty (kW)

Ma = mass flow rate kg/sec

S = specific sigma energy kJ / kg

Cpw specific heat at a constant pressure kJ / Kg Deg C

Subscripts for air and water; a w

Subscripts for inlet and outlet: i o

In cases of mine coolers the duty equation is often simplified to:

$Q = K(T\ wbi - T\ wi)$

In which K is a characteristic of a given coil unit and is a function of the mass flow rates of air and water.

This is the calculation that is used when testing a coil to ascertain the range of duties the coil is capable of at the air and water flow rates the coil was tested at.
Once the K value has been determined this value can be extrapolated into a varying degree of differing temperatures to ascertain its duty.

If the air cooler is calculated on a mass enthalpy basis the formulae used would be:

$Q = Ma\ (Hai - Hao) = M\ wi\ Cpw\ (Two - Twi) - J1$

Where $J1$ is $= Mc\ Cpw\ tc$

Based on mass flow rates and sigma energies, the duty is:

$Q = Ma\ (Sai - Sao) = Mw\ Cpw\ (Two - Twi) - J2$

Where $J2 = Mc\ Cpw\ Tc + Ma\ Cpw\ (Ri\ Twbo - Ro\ Twbi)$

Terms J1 & J2 relate to the secondary effect of condensation and can usually be ignored.

R moisture content of air, kg/kg

If first principals are used to evaluate the overall heat transfer coefficient for a new coil material and design the following formulae would be utilised as a base line calculation for all future coils designed of the same material.
Thereafter a computer model would be designed from the original calculation verified from the results achieved from subsequent laboratory testing.

$U = 1 / (1/ h1 + (Ai / Ao) (1/ ho) + hf)$

Notes

1. thermal resistance of the material of the tube itself is usually small enough to be neglected, as has been done in this equation.

2. fouling on both inside and outside is usually combined into one term (hf) based on the inside area.

A typical fouling factor used for cooling water: 0,0005 – 0,0008 (m2 degC/W)

Other Formulas Used to calculate coil performance:

Water side:

$Q = m.Cp.dt$

Q	= duty (kW)
m	= mass flow rate (kg/s)
Cp	= Heat capacity of water (kJ/KgK)

dT	= Tin - Tout (°C)

$v = m/a$

v	= Velocity (m/s)
m	= mass flow rate (kg/s)
a	= area of flow (m^2)

$dP = 4flv^2/2gd$

dP	= Pressure drop (kPa)
f	= Friction factor
l	= length of tube (m)
v	= velocity (m/s)
g	= gravitational acceleration (m^2/s)
d	= inside diameter of tube (m)

Psychometric:

$Q = G.(Hai - Hao)$

Q	= duty (kW)
G	= mass flow (kg/s)
Hai	= Enthalpy Vair in (kJ/kg)
Hao	= Enthalpy Vair out (kJ/kg)

$Pv = Ps - A(Tdb-Twb).Patm$

Pv	= Vapour Pressure (kPa)
Ps	= Vapour pressure saturated
A	= Psychrometric Factor
Tdb	= Dry bulb Temperature (°C)
Twb	= Wet bulb temperature (°C)
Patm	= Atmospheric pressure

$Ps = 0.615.\exp[17.27.T/(237.3 + T)]$

$H = 1.005.Tdb + (0.0018\ Tdb) + 2.501).ASH$

ASH = Apparent specific humidity

ASH = 622.Pv / (Patm – Pv)

Vv = 0.287(273.15 + Tdb) / (Patm – 0.378.Pv)

Vv = True specific volume of air

∅ = Pv/Psdb . 100

∅ = Relative Humidity (%)

Heat exchanger Design calculations:

All coils are based on the fin and tube heat exchanger. The coil model is therefore the dimensions of the fin and tube, irrespective of the coil type. For example, the model of a chilled water coil would be:

4 CWC 21 x 720 x 4r x 12f – 7c

The dimensions are interpreted as follows:

- 4 is the nominal tube diameter in eighths of an inch. Why? Because the common tube sizes are still based on imperial measurement.
- 21 is the fin height in inches. Once again, the inch measurement is the result of the fin press having an imperial dimension.
- 720 is the finned length of the coil in millimetres.
- 4 row deep
- 12 fins per inch fin spacing
- 7 parallel fluid circuits

The performance of all coils is based on the simultaneous solution of the following 3 heat transfer equations:

1. Air side duty: Q = ma . (hai-hao)
2. Fluid side duty: Q = mw . Cpw . (two-twi)

3. Heat Transfer: Q = e . Qmax

The above is the basis of the effectiveness method where clearly, the performance is controlled by the effectiveness.

Qmax is the maximum possible duty that could be achieved for the coil and this is different for the various coil types. In a hot water coil, For example, this would be Ca (twi - dbi).
In the coil calculation programs, we have used the counterflow effectiveness
e = (1-exp(-Ntu(1-Cr)))/(1-Cr*exp(-Ntu(1-Cr)))

where
Ntu = Number of transfer units
Cr = Capacity Ratio = Air Capacity Rate / Water Capacity Rate
Ntu = Ntuo/(1+Cr*Nr)
Nr = Ntuo/Ntui
Ntuo = outside transfer units = ho*Ao/Ca
Ntui = inside transfer units = Ui*Ai/Cw
ho = eo*Outside film coefficient
eo = fin effectiveness = 1-(Asec/Ao)*(1-ef)
Asec = coil secondary area
Ao = Coil outside surface area
ef = fin efficiency = tanh(mL)/mL
mL = Hf*Sqrt(2*hc/(Kf*tf))
Hf = Fin height
Kf = Fin conductivity
tf = Fin thickness
hi = Film heat transfer for liquid in tube
Ui = Inside Coefficient = hi*(Kw/(Kw+hi*Wt))
Kw = Wall thermal conductivity
Wt = Wall thickness
B = Ratio of coil outside to inside surface areas

Furthermore, we see that the two unknowns are the outside and inside film coefficients and these are determined by the types of fluids and the characteristics of the flow.

On the air side, this is common to all coils. Here, we have used the film coefficient for airflow over finned coil from Dr AH Elmahdy Dr RC Biggs, "Finned Tube Heat Exchanger: Correlation of dry surface heat transfer data", ASHRAE Paper No 2544. valid for 200 < Re < 2000.

For water coils, both hot and cold, we use the well known Dittus-Boelter equation
Nu = 0.026 Re^0.8 Pr^n where n=0.4 for heating and 0.3 for cooling
In the refrigeration coils, we have adopted the boiling and condensing equations as published in the ASHRAE 2001 Fundamentals.

Allowance is made for moist air condensation where the surface is below the air dew point by calculating an equivalent moist air capacity rate.

6.6 Calculations and Discussion

The refrigeration unit chosen for the project was a Daikin EUW, 200KX with a capacity range of 174–580 kW. The Daikin capacity tables suggested a target leaving water evaporator condition of 7°C and condensing temperature of 45–50°C, this equating to a cooling capacity of 484–516 kW.

All calculations were done via an Excel program. The results were then verified by an independent third party namely the CSIR (Miningtek). The results were excellent and only showed and average deviation of 4.3 %

After initial calculations, it was determined that four heat rejection coils, as detailed below, would be capable of **just** rejecting the required heat load at the lower end of the cooling capacity targeted. Consideration was made of the length of piping for the hot water reticulation, and subsequent heat rejection through the pipes – this was estimated at 21 kW.

Heat Rejection (Condenser Coils)

4 HWC-54 x 1500 x 8r x 6f - 36 c

DESIGN

Tube diameter	1/2 inch
Height	54.0 inches
Length	1500 mm
Rows deep	8
Fin spacing	6 fpi
Circuits	36.0
Serpentine	1

AIR

Barometer

Atmospheric pressure	101.325 kPa
Altitude	0 m (asl)

Properties

Inlet temperature	29.9/26.8 °C wb/db	
Outlet temperature	45.1/30.2 °C wb/db	
Inlet enthalpy	83.9 kJ/kg	
Outlet enthalpy	99.7 kJ/kg	
Inlet abs. humidity	21.1 g/kg	
Outlet abs. humidity	21.1 g/kg	
Inlet rel. humidity	79 %	
Outlet rel. humidity	35 %	

Flow

Mass flow	10.641 kg/s	
Inlet volume flow	9.250 m3/s	(3)
Face velocity	4.50 m/s	

Other

Air pressure drop	372.2 Pa	

WATER

Inlet temperature	50.0 °C	
Outlet temperature	45.4 °C	
dT	4.6 °C	
Mass flow	8.867 kg/s	(4)
Tube velocity	2.00 m/s	(5)
Pressure drop	58.925 kPa	
Density	989 kg/m3	
Heat capacity	4.181 kJ/kgK	
Conductivity	640.7 kW/mK x 1e6	

DUTY

Heat rejection	168.7 kW	(6)

Various coil configurations, water temperatures, airflow, etc. were modeled and the above was found to be the most effective. Another advantage was that the same coils could be used for the cooling function as shown later.

Rejecting circa 674 kW through the coils translated into a cooling capacity of 484 kW from the plant – lower than the required 500 kW. This reduced duty was accepted since the aim of the project was to prove the concept of remote, closed loop cooling systems. Site limitations would have to be considered at mine design stage if this system of cooling were to be implemented in the future in order to ensure optimal functionality of the plants. It was understood that the site selected for trial was not ideal in that it could only accommodate three cooling coils.

It must also be noted that a sensitivity study done on the coils suggested that seasonal variance on ambient temperatures, coupled with inevitable external fouling due to the fact that the coils would be dry, would negatively influence the ability to reject the required heat.

To address this, a humidifying ring which would spray service water onto the coils was to be installed to literally wash off the airborne sediment from the fin surfaces. This would also have a small positive effect on heat rejection capacity, as it would also slightly pre-cool the air coming onto the coil as well as facilitate some evaporative cooling. The effect of this was difficult to predict as the water temperature was found to vary considerably, and the sprays would not be continuous.

Original design suggested that the fans for the coolers would be 30 kW axial flow type – situated downstream of the coils, i.e. air was to be sucked through the coolers. This was considered necessary to ensure inspection and maintenance of the humidifying ring. Thermodynamically it would be better to have the fan upstream, thereby making full use of the cooling effect on the heat load. It is also good practice to have the electrical equipment (fans) on the dry end.

After initial calculations and costing, it was decided to re-visit the heat rejection coils to try to reduce the number of coils used to:

- reduce the size of cubby required

- optimize the cost of the project.

It was decided to reject through three coils. (same constraints as before)

4 HWC-57 x 1500 x 8r x 8f - 38 c

DESIGN

Tube diameter	1/2 inch
Height	57.0 inches
Length	1500 mm
Rows deep	8
Fin spacing	8 fpi
Circuits	38.0
Serpentine	1

AIR

Barometer

Atmospheric pressure	101.325 kPa
Altitude	0 m (asl)

Properties

Inlet temperature	29.9/26.8 °C wb/db
Outlet temperature	46.0/30.4 °C wb/db
Inlet enthalpy	83.9 kJ/kg
Outlet enthalpy	100.7 kJ/kg
Inlet abs. humidity	21.1 g/kg
Outlet abs. humidity	21.1 g/kg
Inlet rel. humidity	79 %
Outlet rel. humidity	33 %

Flow

Mass flow	13.8 kg/s	
Inlet volume flow	12.0 m3/s	(7)

Face velocity	5.53 m/s	

Other

Air pressure drop	529.5 Pa	

WATER

Inlet temperature	50.0 °C	
Outlet temperature	43.7 °C	
dT	6.3 °C	
Mass flow	8.00 kg/s	(8)
Tube velocity	1.98 m/s	
Pressure drop	59.024 kPa	
Density	989 kg/m3	
Heat capacity	4.180 kJ/kgK	
Conductivity	640.2 kW/mK x 1e6	

DUTY

Heat rejection	211.67 kW	(9)

1) Daikin Hydronic Systems #6, Pg 204
2) Pipe Diameter, D = 114mm
   Air Velocity in tunnel, v = 1 m/s (assumed)
   Air Temperature in Tunnel, tai = 34 deg C (assumed)
   Water Temperature at pipe inlet, twi = 50 deg C
   Dynamic Viscosity air, u = 1.846e-5 kg/ms
   Density air, r = 1.177 kg/m3
   Conductivity air, k = 2.624e-5 kW/mK
   Prandtl Number air = 0.707
   Reynolds Number = rvD/u = 7268
   Nussult Number = hD/k or 0.023*Re^0.8Pr^0.4 = 24.58
   Film Coefficient =, h = 5.65 W/m2C (U)
   A = pi D L = 0.359 m2/m length
   dT to coils = 16
   dT from coils = 11
   Q = U A dT = 32 W/m length (supply to coils)
   = 22 W/m length (from coils)
   say 27 W/m mean
   approximately 800m piping = 800 Q = 21kW
3) 9.250 x 4 = 37 m3/s (Max volume through RAW)

4) 8.867 x 4 = 35.468 l/s (Daikin Hydronic Systems #6, Pg 200, Condenser Water Flow Range 17.4 – 45.56 l/s)
5) In-house design limitation i.e. 1.8 - 2.1 m/s water flow velocity to prevent erosion of tubes.
6) 168.7 x 4 = 674.8 kW plus heat rejected through pipes (21kW) = 695.8kW total.
7) 12 x 3 = 36 m3/s
8) 8.0 x 3 = 24 l/s (Within chiller specification)
9) 211.67 x 3 = 635.01 kW plus heat rejected through pipes (21kW) = 656 kW total.

Note: All calculations done using Std Psychrometric Equations as per "Mine Ventilation Society" Data Book Pgs TP-P 4 and TP-P 5
Coils are manufactured of Cu tubes with Aluminium Fin.

The total heat rejection capacity would be circa 656 kW, which obviously reduced the cooling capacity – this was determined at approximately 460 kW. It was also highlighted that the external fouling of the coils would be compounded, as tighter fin spacing was required to achieve the duty. Again, the 30 kW fans should be used. This did not happen in practice as the mines supplied their own fans with an IP (Internationally Protected; i.e. the protection of the moving components inside the motor from ingress of contaminants – dust and water) rating, which would not allow downstream use as specified.

Heat Transfer (Evaporator Coils

In terms of the evaporator cooling coils, it was decided to use two coils identical to the heat rejection coils selected. The calculated cooling duty amounted to 242 kW.

4 CWC-54 x 1500 x 8r x 6f - 36 c

DESIGN

Tube diameter	1/2 inch
Height	54.0 inches
Length	1500 mm
Rows deep	8
Fin spacing	6 fpi
Circuits	36.0
Serpentine	1

AIR

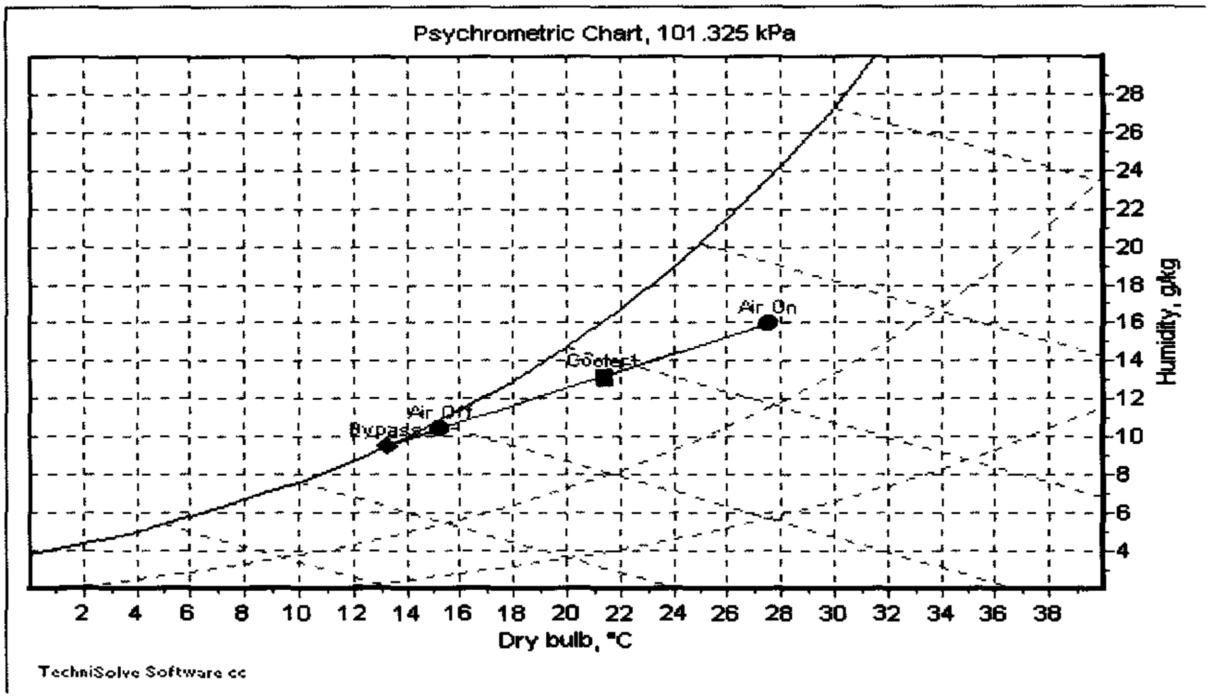
Barometer		
Atmospheric pressure	101.325 kPa	
Altitude	0 m (asl)	
Properties		
Inlet temperature	27.5/23.0 °C wb/db	
Outlet temperature	15.4/14.9 °C wb/db	
Inlet enthalpy	68.0 kJ/kg	
Outlet enthalpy	41.8 kJ/kg	
Inlet abs. humidity	15.8 g/kg	
Outlet abs. humidity	10.4 g/kg	
Inlet rel. humidity	69 %	
Outlet rel. humidity	96 %	
Flow		
Mass flow	9.305 kg/s	
Inlet volume flow	8.0 m3/s	
Face velocity	3.89 m/s	(1)
Other		
Air pressure drop	398.6 Pa	

WATER

Inlet temperature	7.0 °C	
Outlet temperature	15.2 °C	
dT	8.2 °C	(2)
Mass flow	7.000 kg/s	(3)
Tube velocity	1.59 m/s	
Pressure drop	46.2 kPa	
Density	999 kg/m3	
Heat capacity	4.192 kJ/kgK	
Conductivity	589.1 kW/mK x 1e6	

DUTY

Total cooling	242 kW	(4)



The two coolers would achieve the 484 kW produced by the plant. For the air volume required 18.5kW fans would be used. The decision to use coolers of the size above was based on a number of considerations: it is a standard size manufactured by ourselves, there is capacity for duty improvement if ambient conditions become more unfavourable, the heat rejection and cooling coils were identical so from a redundancy point of view it was more practical.

The above was adjusted after the heat rejection design was amended:

4 CWC-54 x 1500 x 8r x 6f - 36 c

DESIGN

Tube diameter	1/2 inch

Height	54.0 inches
Length	1500 mm
Rows deep	8
Fin spacing	6 fpi
Circuits	36.0
Serpentine	1

AIR

Barometer

Atmospheric pressure	101.325 kPa
Altitude	0 m (asl)

Properties

Inlet temperature	27.5/23.0 °C wb/db
Outlet temperature	14.8/14.4 °C wb/db
Inlet enthalpy	68.0 kJ/kg
Outlet enthalpy	40.4 kJ/kg
Inlet abs. humidity	15.8 g/kg
Outlet abs. humidity	10.1 g/kg
Inlet rel. humidity	69 %
Outlet rel. humidity	96 %

Flow

Mass flow	8.374 kg/s
Inlet volume flow	7.200 m3/s
Face velocity	3.50 m/s

Other

Air pressure drop	315.2 Pa

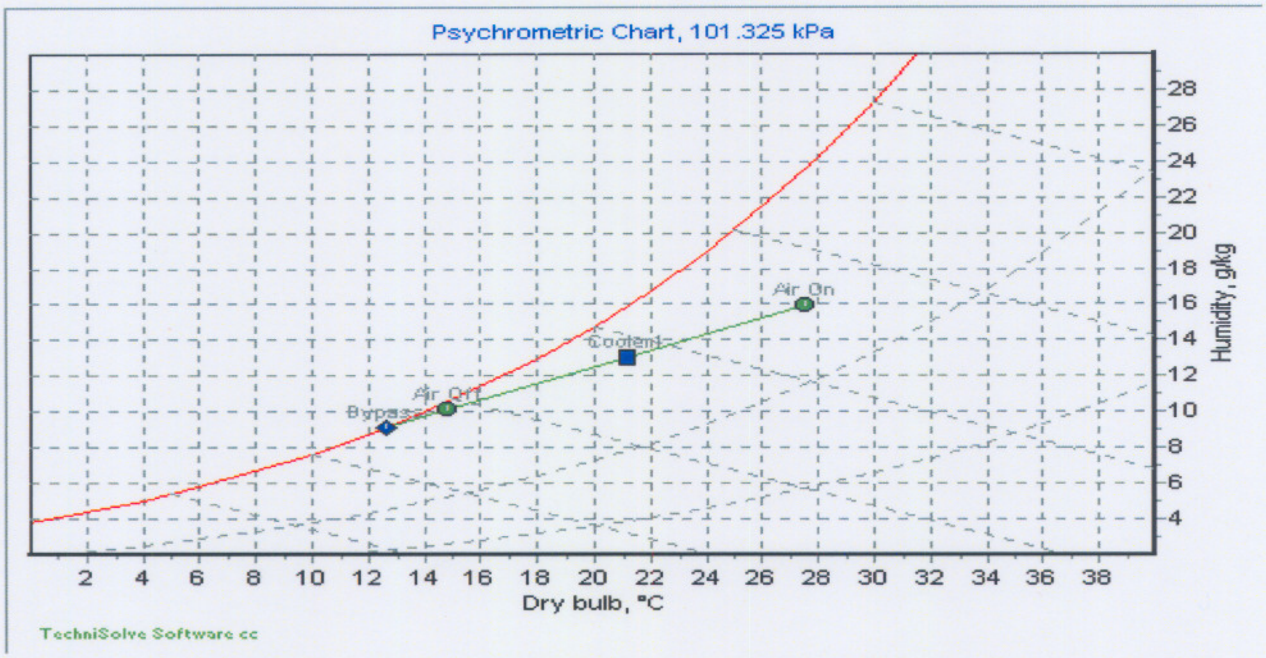
WATER

Inlet temperature	7.0 °C
Outlet temperature	14.9 °C

dT	7.9 °C	
Mass flow	7.0 kg/s	
Tube velocity	1.58 m/s	
Pressure drop	46.9 kPa	
Density	999 kg/m3	
Heat capacity	4.192 kJ/kgK	
Conductivity	588.4 kW/mK x 1e6	
DUTY		
Total cooling	230.9 kW	(5)

1) Air volume kept low to produce duty – some capacity for duty increase if ambient conditions change.
2) Higher than Daikin specification of 2 – 5 deg C, well within the Plant's capacities though.
3) Within Plant Capacity (13.8 – 42.8 l/s)
4) Total Capacity = 242 X 2 = 484 kW
5) Cooling Duty = 231 kW

Total Capacity = 230 X 2 = 460kW



The units would achieve the 460kW produced – the only change made was the reduction of air through the coolers. Again 18.5kW axial flow fans should be utilised.

The two coolers would achieve the 484 kW produced by the plant. For the air volume required, 18,5 kW fans would be used. The decision to use coolers of the size above was based on a number of considerations namely:

- It is a standard off-the-shelf unit.
- There is capacity for duty improvement if ambient conditions become more unfavorable.
- The heat rejection and cooling coils were identical, so from a standardisation point of view it was more practical.

Figures 6.1 and 6.2 are photographs showing the refrigeration machine and cooling coil heat exchangers used in the project.

Figure 6.1: Refrigeration machine

Figure 6.2: Coil heat exchangers

Chapter 7
DYNAMIC DESIGN

In this chapter, the dynamic/active design will be discussed.

7.1 Overview

This chapter shows the operating conditions relative to the design by using dynamic (referring to the re-iterative process where actual conditions are taken into account, and where feedback is given into the system) integrated computer simulations and ultimately whether this system will produce the cooling required. These simulations were done by Temmi using the Quick Control program which is a widely used, well verified simulation program.

7.2 System Configuration

The system consists of a chiller, two evaporator-cooling cars and three condenser-cooling cars. The cooling cars consist of a cooling coil and fan. The chiller consists of a semi-hermetic reciprocating compressor, a shell, and finned tube heat exchangers for the evaporator and condenser respectively.

In addition, the system comprises of two closed water loops. The first loop works as follows: The chiller cools the water at the evaporator. This water is then pumped through the two evaporator-cooling car coils where the water is used to cool the air going to the work area that needs to be cooled. This water is then passed to the evaporator of the chiller where it is cooled again. The second loop starts at the condenser where the refrigerant heat is rejected to the water. The water is then pumped through the three condenser-cooling car coils where it is cooled by the ventilation air exiting the mine. The cooled water then returns to the condenser where it is heated again. (This is diagrammatically depicted on the next page.)

Each of these loops can entail different configurations with respect to the air and water flow.

7.3 Different Options

Each of the loops can be used in four different configurations with respect to the air and water flow. Only the evaporator loop will be discussed. The only difference between the two loops with respect to the configurations is the number of cooling cars.

Figure 7.1: Configuration A

In configuration A, the two cooling cars are in parallel with respect to both the air and water flows.

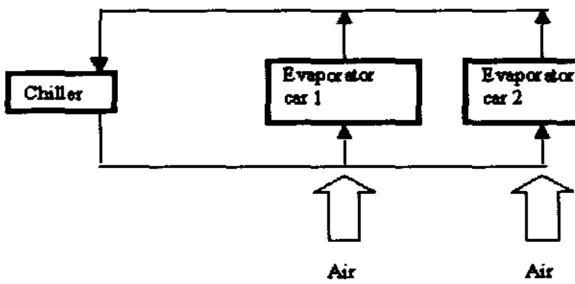


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Figure 7.2: Configuration B

In configuration B, the two cooling cars are in series with respect to both the air and water flows.

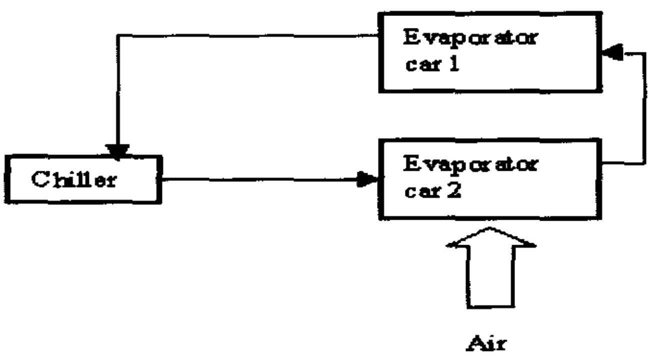


Figure 7.3: Configuration C

In configuration C, the two cooling cars are in series with respect to the airflow but in parallel with respect to the water flow.

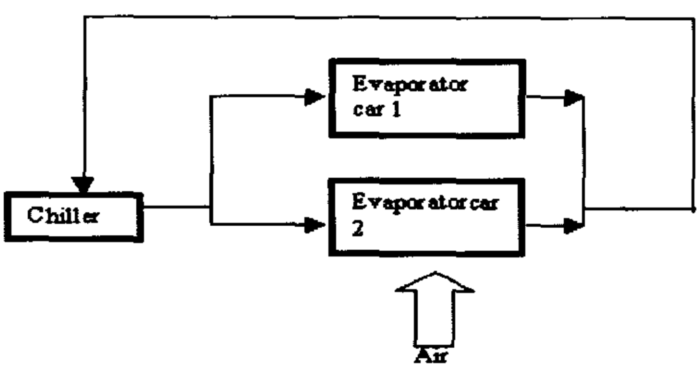
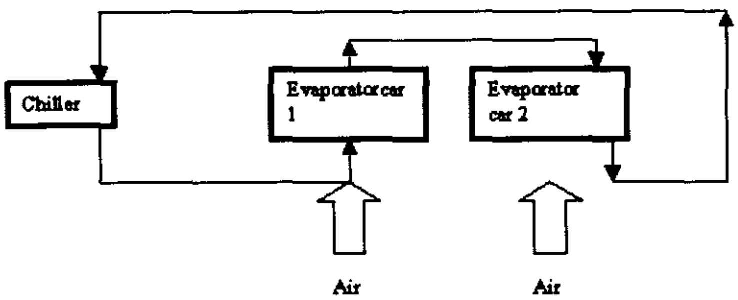


Figure 7.4: Configuration D

Configuration D is just the inverse of configuration C. Here the two cooling cars are in series with respect to the water flow and in parallel with respect to the airflow.



These different configurations on both the evaporator and condenser loops lead to 16 different configurations in which this system can be used. All these configurations were compared to one another to decide on the best configuration.

7.4 Condenser Coils

Assuming the entering coil conditions are correct, all other calculations in this chapter are based on these figures, then:

Cooling capacity	=	509 kW
Compressor power	=	180 kW

According to the manufacturers, these figures will not vary as long as the flow stays within specified limits and the exiting water temperatures stay the same. The assumptions used to determine the chiller's performance lead to very conservative values for cooling capacity and compressor power being used. This will lead to the condenser coils appearing to be less underdesigned as shown later in this chapter. The evaporator coils on the other hand would appear more overdesigned than shown.

The flow through the evaporator is within limits. The flow through the condenser is too low.

Energy balance over chiller:

$Q_c = Q_e + Q_{comp}$

This leads to $Q_c = 689$ kW.

Using the equation for the heat gained by the water at the condenser we have:

$Q_c = m^*c_p^*(T_{cin} - T_{cout})$

Then Qc = 24*4.19*(50-43.7) = 633.528 kW

Nevertheless, 689 kW must be rejected at the chiller's condenser. This means that the condenser-cooling car coils are 8% underdesigned. This will cause the entering condenser temperature to be higher than expected. The effect of this will be investigated using integrated simulations.

For the evaporator-cooling car coils the cooling capacity can be calculated as follows:

$Q_e = m^*c_p^*(T_{ein} - T_{eout})$

Then Q_e = 15.2*4.19*(15-5) = 636.88 kW

The chiller has a cooling capacity of only 509 kW. This means that the evaporator-cooling car coils were 25% overdesigned. This will cause the entering evaporator water temperature of the chiller to be higher than expected which inevitably will cause the exiting temperature to be higher as well which, on its part, will influence the performance of the evaporator-cooling cars.

7.5 Verification

The condenser coil, evaporator coil and chiller models were each simulated on its own. Design values were used to configure the coil models. The results of the verification study can be seen in Table 7.1.

Table 7.1: Verification results

	Design Value	**Simulated Value**	**% Error**
Chiller			
Leaving evaporator water temperature (°C)	**7**	**7.02**	**0.41**
Leaving evaporator water temperature (°C)	**50**	**50.67**	**1.14**
Condenser coiling coil			
Leaving water temperature (°C)	**43.7**	**45.84**	**0.32**

Evaporator cooling coil			
Leaving water temperature (°C)	**15**	**15.2**	**1.30**

As can be seen, the models are very accurate. Due to these satisfactory results, the 16 configurations were modeled and simulated.

7.6 Simulation Results

The different configurations on both the evaporator and condenser loops lead to 16 different configurations in which this system can be used. All these configurations were compared to one another to decide on the best configuration.

The results are summarised in bar charts, which compare the configurations with respect to one temperature at a time.

Figures 7.5 and 7.6: The best performances with respect to air temperatures are configurations 1 to 4.

Figure 7.5: Evaporator exiting dry bulb temperature

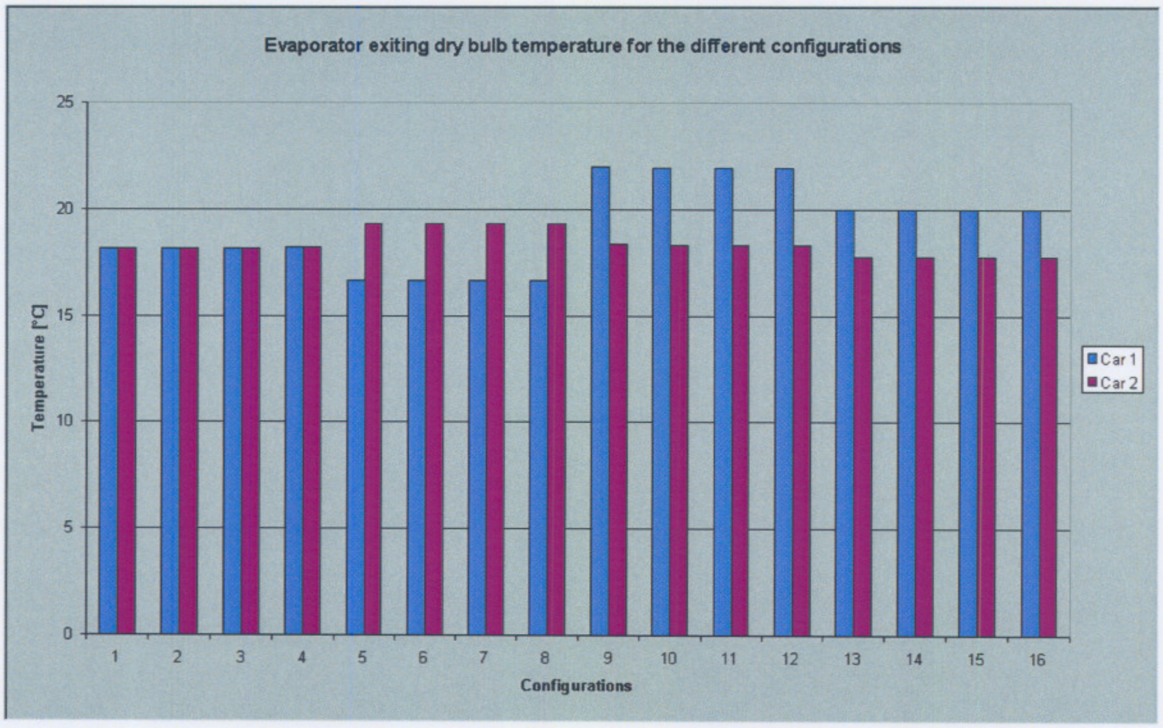
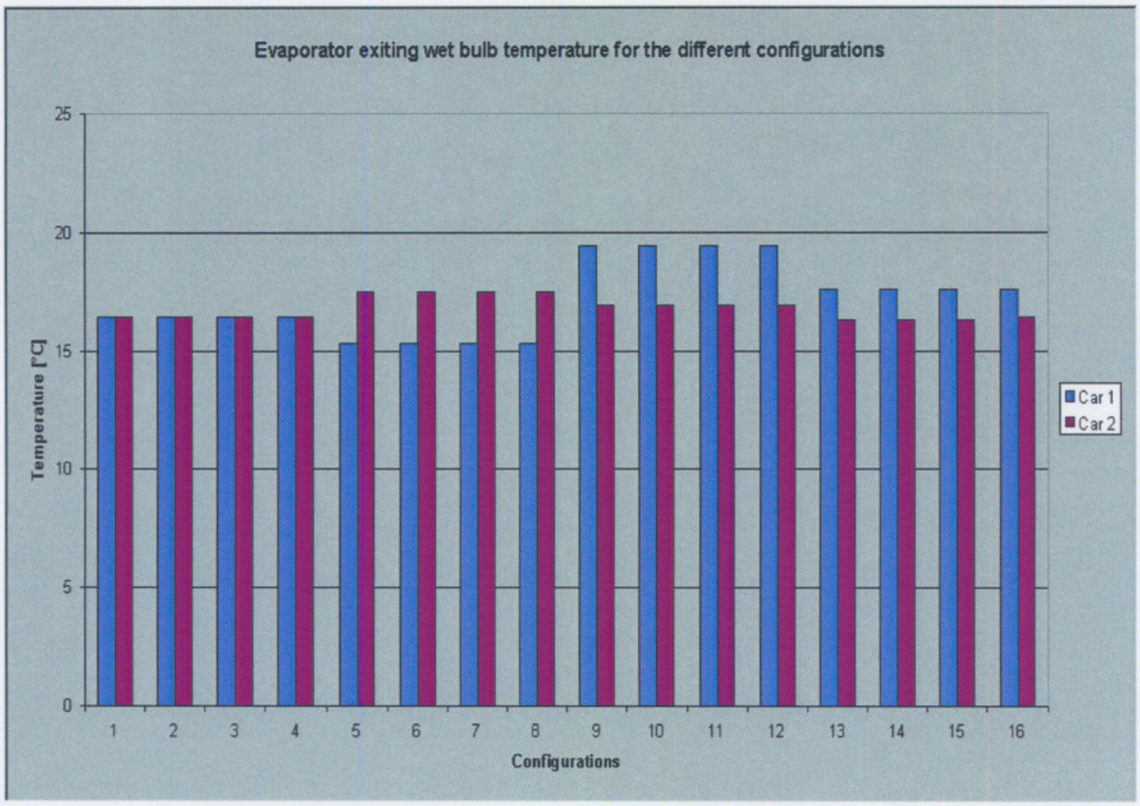


Figure 7.6: Evaporator exiting wet bulb temperature



Figures 7.7 and 7.8: These graphs give the exiting condenser dry bulb and wet bulb temperatures. The best configurations, with respect to the air temperatures, are 3, 7, 11 and 15. The second important observation is that the exiting air temperatures are very high (45°C). This means that the placement of the condenser cooling cars would be very important.

Figure 7.7: Condenser exiting dry bulb temperature

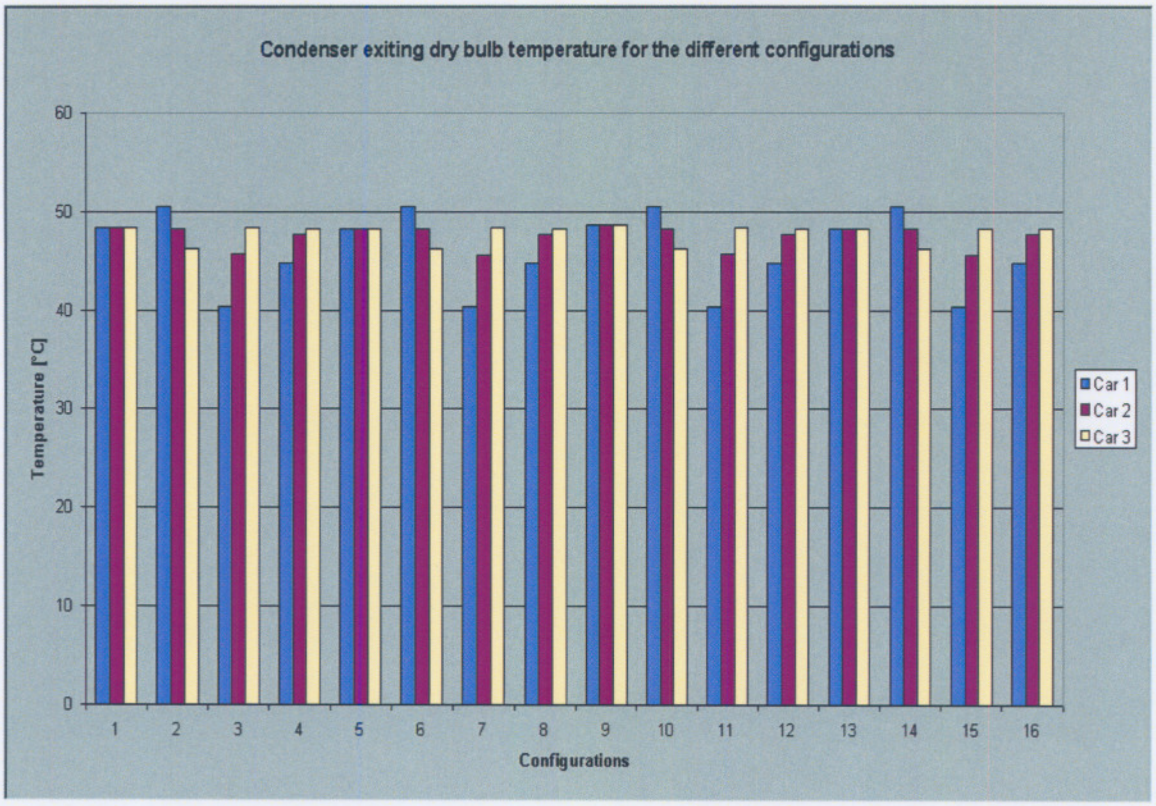
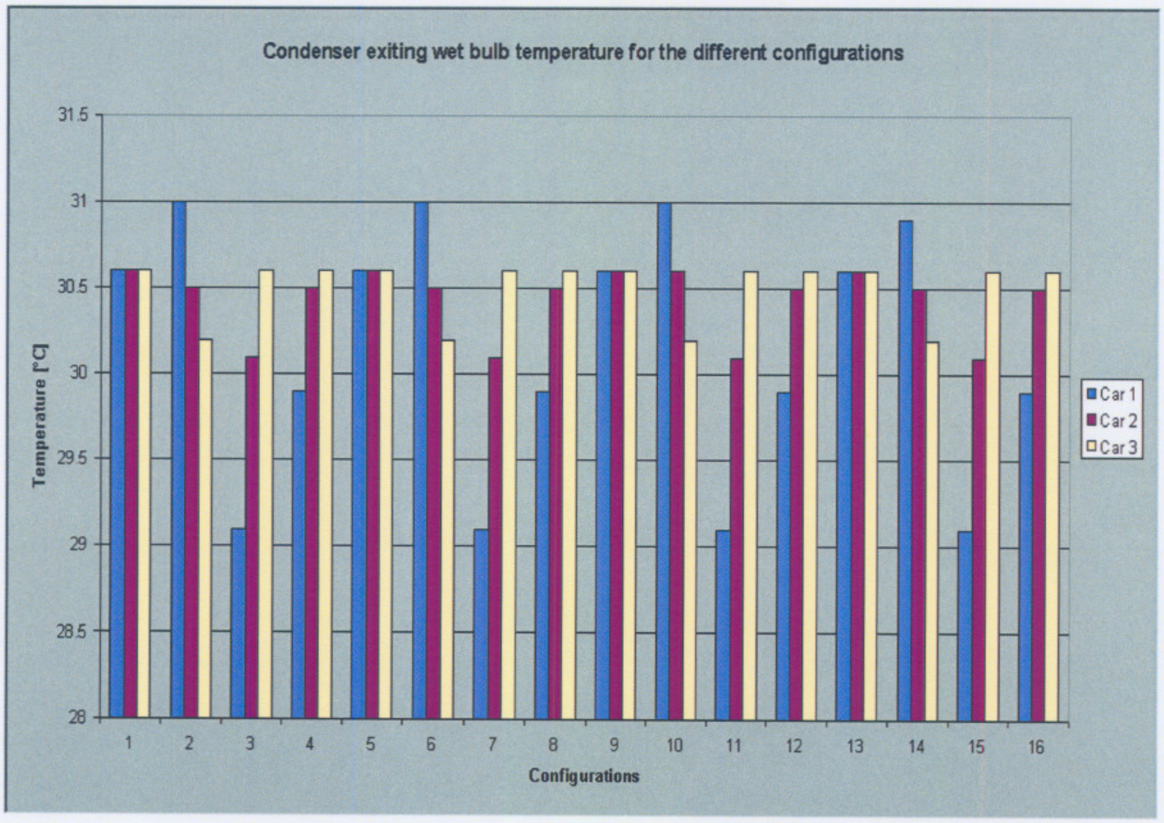


Figure 7.8: Condenser exiting wet bulb temperature



Figures 7.9 and 7.10: The following figures show the effects of the over- and underdesign of the evaporator and condenser coils respectively. In Figures 7.6 and 7.7 it can be seen that the exiting evaporator coil water temperature is on average about 2,5°C higher than the design values. This leads to the entering evaporator coil temperature being between 3 and 6°C higher than expected. The best configurations, with respect to the water temperatures, are configurations 1 to 4.

Figure 7.9: Evaporator exiting water temperature

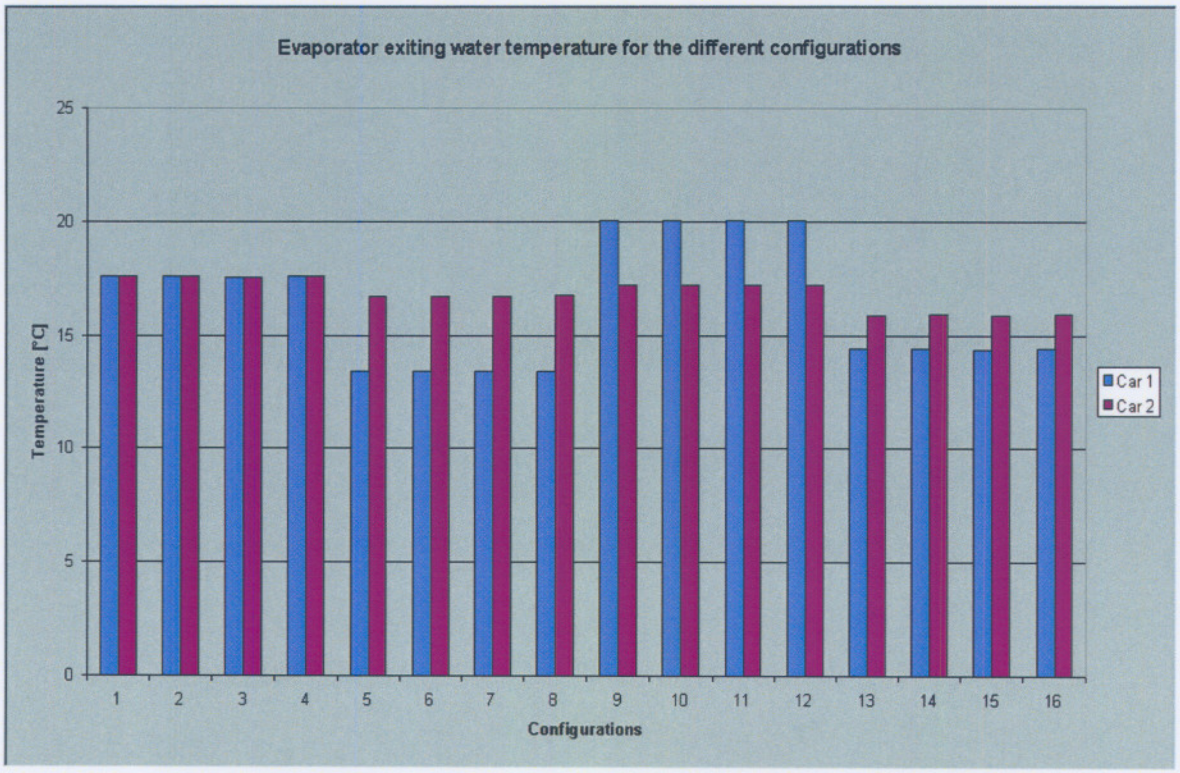
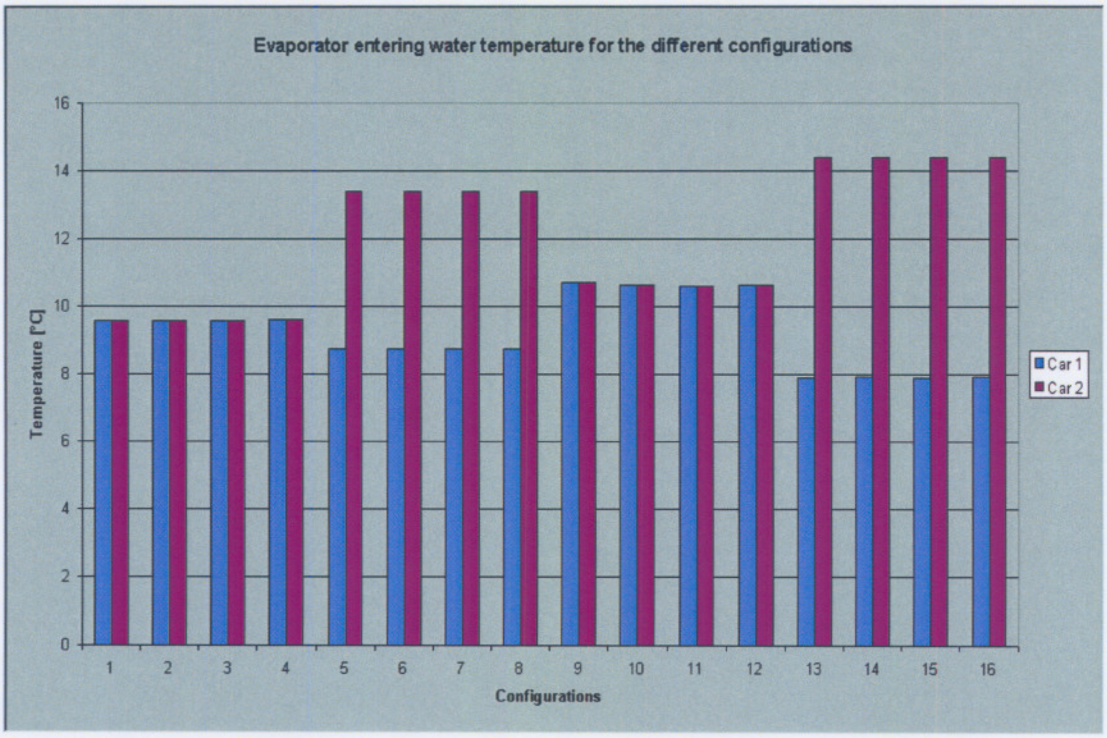


Figure 7.10: Evaporator entering water temperature



Figures 7.11 and 7.12: The coil exiting temperatures are between 1,5 and 5°C higher than the design expectations. The entering coil temperatures are also higher than expected by the same margin. The chiller's condenser exiting temperatures are getting very close to the maximum permissible temperature of 55°C; in some of the configurations this temperature is surpassed. The best configurations, with respect to the condenser water temperatures, are configurations 3, 7, 11 and 15.

Figure 7.11: Condenser exiting water temperature

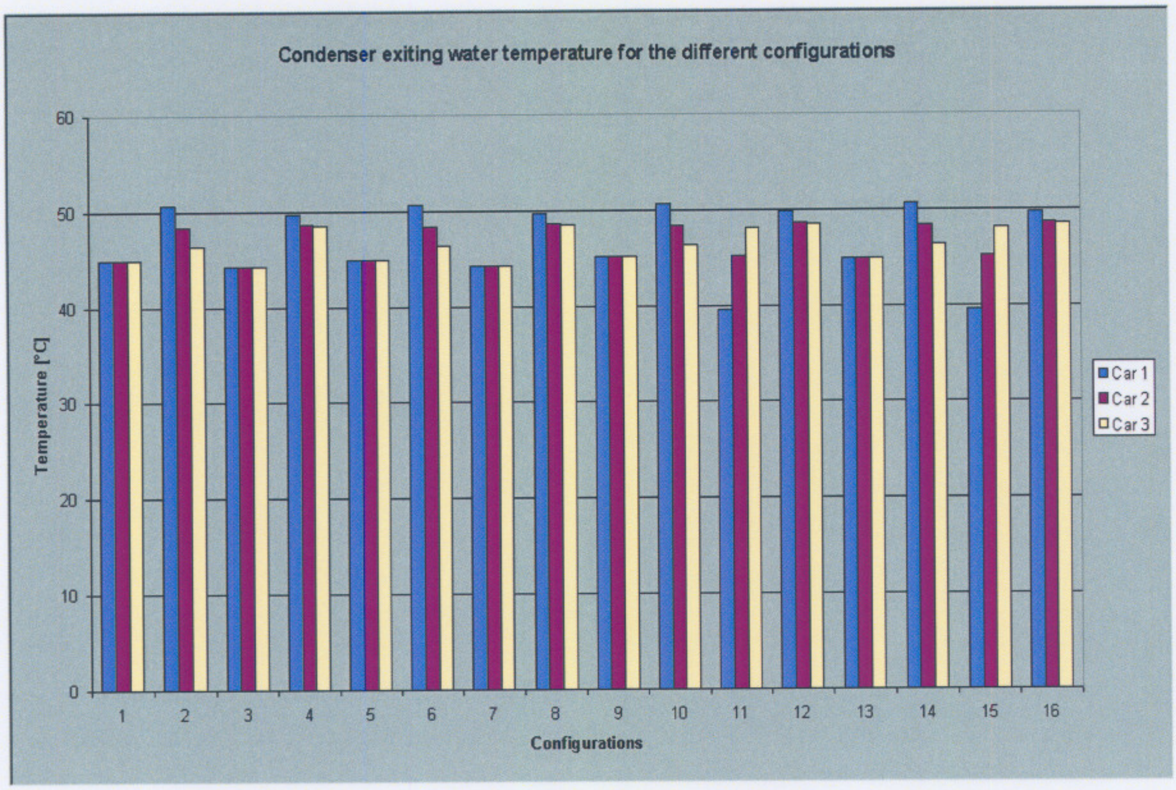
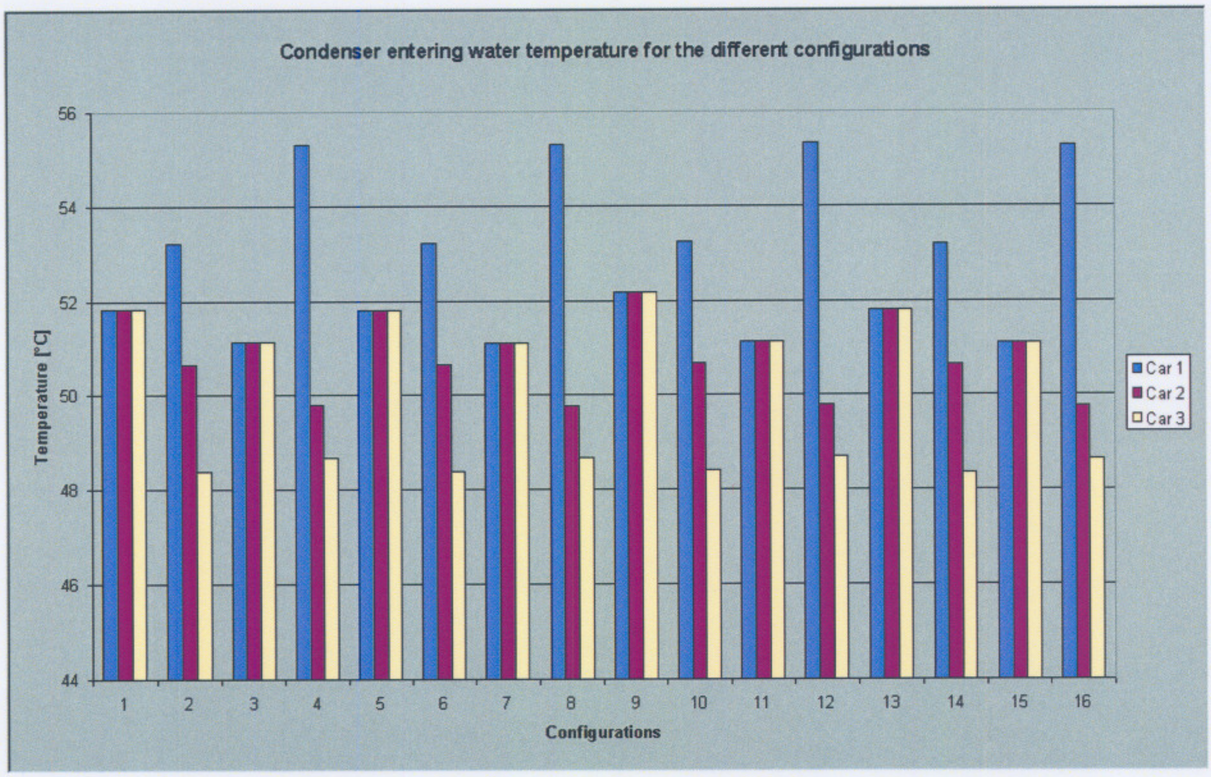


Figure 7.12: Condenser entering water temperature



7.7 Best Configuration

The configuration with the best performance is configuration 3. This means that the best results will be obtained if the evaporator-cooling cars are used in parallel to each other with respect to both airflow and water flow. The condenser-cooling cars on the other hand should be connected in parallel with respect to the water flow and in series with respect to the airflow.

7.8 Discussion

The review of the design data showed that the evaporator-cooling coils were 25% overdesigned and the condenser-cooling coils 8% underdesigned. Due to this fact, computer simulations were done to see how this would affect the system. The cooling system has a capacity of only 509 kW. The coefficient of performance (COP) of the chiller is only 2.83 where the COP at the design conditions equals 3.3. This decrease in performance is due to the high condenser temperatures.

Chapter 8
COMPARISON BETWEEN THE STATIC AND DYNAMIC DESIGN APPROACHES

In this chapter, the comparison between the static and dynamic design approaches will be discussed.

8.1 Observation

The following two premises should be taken into consideration in the discussion and analysis below:

- The emphasis of the static design was to prove the engineering feasibility of the design.
- The emphasis of the dynamic design was not only to prove the engineering feasibility, but also to optimise.

8.2 Discussion

Qualifying statement: The following comments are all relative to the Dynamic simulation.

The comment that the evaporator coils are 25% overdesigned is very clinical and calculated on an exact duty. As can be seen from the preceding pages, the static design approach was based on how much heat rejection capacity could be achieved at a specific site. A cooling duty was then calculated and the cooling coils selected to suit. The fact that the coils have the capacity for more (or less) duty is of no consequence since this can be controlled by airflow/water flow rates, and would in fact need to be set up since the environmental conditions would invariably not be as used during the design.

Additionally, the cost attributed to this "overdesign" is based on the assumption that all twenty of the units would have similar environmental conditions, and would subsequently all be overdesigned. It is standard practice that one would design a system to suit a particular site/application. In this case a certain amount of flexibility in terms of operating range would be incorporated (10 to 15%). This is necessary as the underground thermal environment is not static but varies according to diurnal and seasonal changes. Finally, the selected refrigeration unit must have sufficient flexibility in its performance characteristics to accommodate this.

Configuration A most closely resembles the static design (the counter flow recommendation aside) for both heat rejection and cooling. The comment that the heat rejection coils must be in series with respect to airflow and parallel with respect to water flow begs debate. Static calculations of the heat rejection coils show that the air-off temperature is extremely high – within 4-5°C of the water on condition (50°C). When this off condition is used as an on condition for the second coil in series on the air side and recalculated using the original four coils recommended, the second coil can only reject a fraction of the heat load as opposed to a parallel configuration. This would suggest that the total heat rejection required is unattainable – even if a large number of coils were utilised.

From the above it is clear that this configuration requires further study, as the static calculations do not correspond to the dynamic simulations in this instance. This phenomenon is most interesting and could bring a new perspective to thermodynamic heat flows.

The comment that the chiller is operating at the limit of its operational domain is valid, as this design is based on high condensing temperatures. If this system were to be implemented in some form or another, it is recommended that the use of the system be factored into the design phase of the mining operation, which would facilitate the provision of heat rejection capacity – either by means of water-cooled or air-cooled heat exchangers. By addressing this, one would achieve the duty required from the plant quite easily.

8.3 Summary

Since the objective was to prove the concept of remote cooling via MRP, and not to optimise the performance of the unit, the system has a high probability of success.

8.4 Conclusion

Armed with both the dynamic and static assimilations, which produced a close relation in outputs, the decision to proceed with the layout as depicted in Annexure A and B was made, taking full cognizance of the indicated shortfall in condenser-cooling capacity.

Chapter 9
IMPLEMENTATION AND RESULTS

In this chapter, the implementation of the pilot system and actual measured results will be presented.

9.1 Plant Configuration

The plant and evaporator cooling coils were installed on 25 level at Spud shaft, with the heat rejection coils installed on 24 level in the RAW. The diagram (Figure 9.1) depicts the actual underground installation layout:

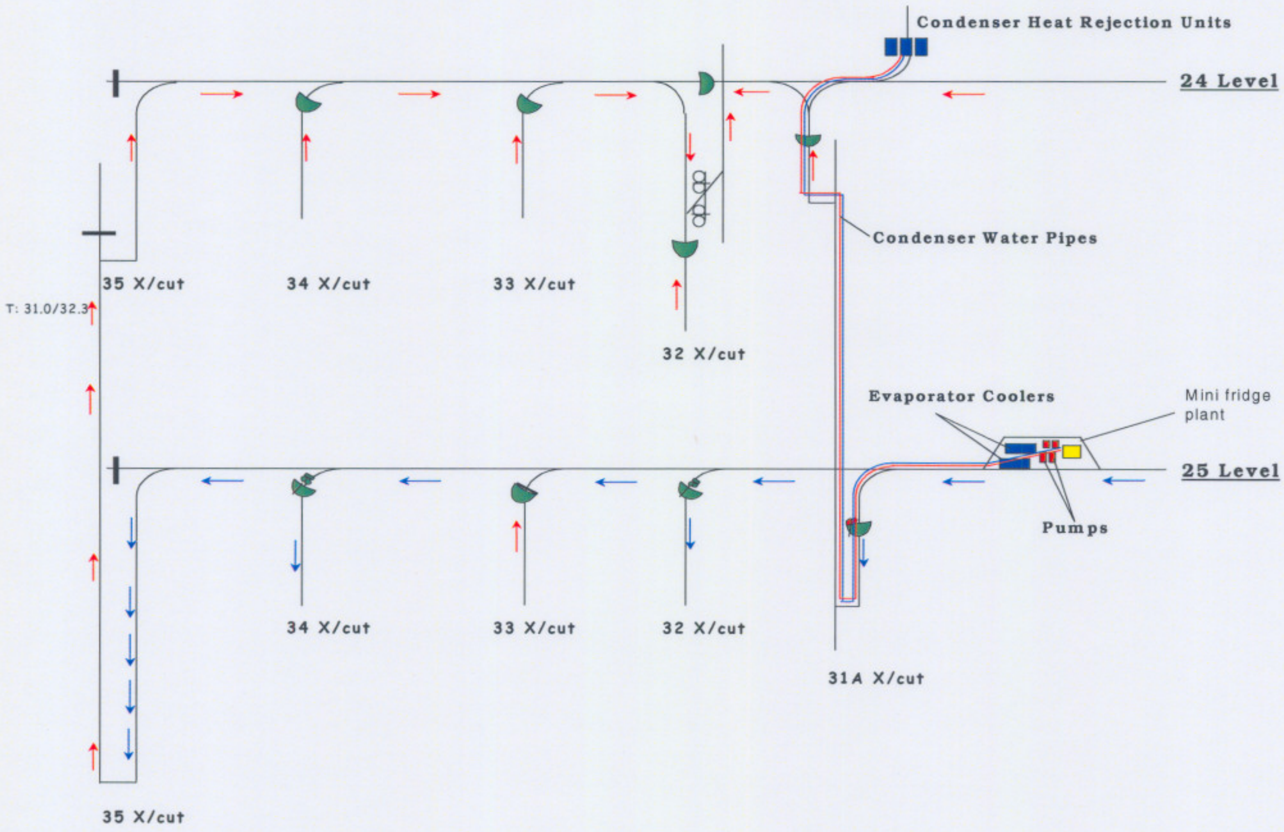


Figure 9.1: Schematic of the underground installation

Of significance, and as can be seen, is the remote installation of the condenser coils on 24 level. This concept now allows the ventilation engineer flexibility in terms of installation.

9.2 Measurement techniques

EQUIPMENT USED:

Airflow Instrument

Calibrated Vane Anemometer Serial No. 140035 with calibration certificate R16 – measuring range of anemometer 0,5 to 30 m/s

Area measuring Instrument
Disto Lazer Meter

Temperature Instrument
Standard whirling hygrometer with calibrated mercury thermometers – range 0 to 50°C gradients.

Time Instrument
Digital Casio Stopwatch with <3 seconds error

Measuring Period

Summer Season	-	January 2002
Time	-	Morning Shift between 10H00 and 12H00
Ambient Conditions		
Wind speed	-	0,18m/s
Temperature	-	24,2/37,6°C

9.3 Results

Actual measurements on the water side (evaporator) are as follows:

Water temperature in	9°C
Water temperature out	18°C
Water mass flow	13 ℓ/s

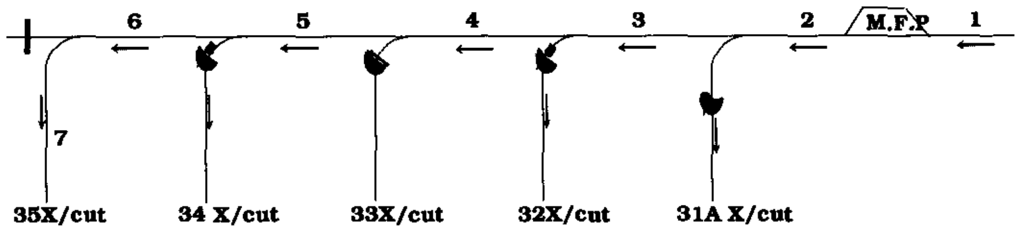
From these measurements, and by using the equation q = m x cp x dt the plant duty is calculated as: q = 13 x 4.187 x 9

= <u>490 kW</u>

This duty correlates exceptionally well with the rated capacity of 506 kW. This proves that the system has met its objective and that the modifications as detailed in the next chapter have achieved the desired result.

The actual measured results within the underground stoping environment, with and without the MRP running, is depicted in Figure 9.2 below.

SPUD SHAFT
25 Level Intake Air



Without Mini Fridge Plant Operating

	7	6		5		4		3		2		1	
Quantity (kg/s)	21.2	21.2		31.4		30.8		41.2		42.6		42.6	
Temp. (wb/db°C	29.0/31.6	27.7/31	0	27.5/30	9	26.5/30	0	26.0/29	0	25.5/28	5	25.5/28	5
Remarks	Mini fridge plant not operational.												

With Mini Fridge Plant Operational

Quantity (kg/s)	21.2	21.2		31.4		30.8		41.2		42.6		42.6	
Temp. (wb/db°C)	25.2/26.6	24.7/25	9	24.4/25	5	23.6/24	7	23.1/24	2	22.4/23	6	26.7/27	7
Remarks	Mini fridge plant operational.												

Figure 9.2: Schematic of system results

The numbers in the diagram correspond to the conditions measured in the table and are indicated vertically below the respective positions.

From the above tabulation, the reduction of the temperature by 4.3°C (26.7 to 22.4) is significant, and is proof of the MRP functionality in terms of performance.

Further calculations and on two measured conditions show the following:-

Water Efficiency (Cooling Tower Efficiency)

Condenser Coils

$$\frac{TWi - TWo}{TWi - TWBi} \times 100 = \text{Efficiency}$$

TWi = Temp. water in

TWo = Temp. water out

TWBi = Temp. wet bulb of air in

Condition 1

$$\frac{48 - 42}{48 - 30{,}4} \times 100 = 34\%$$

Condition 2

$$\frac{48 - 43}{48 - 30{,}9} \times 100 = 29{,}2\%$$

This refers to the effectiveness at which heat is removed from the outlet condenser water.

Condenser duty

Condition 1

$:q = M \times Cp \times \Delta t$

$= 24 \times 4{,}187 \times 48 - 43$

$= 502{,}4$ kW

Condition 2

$:q = 24 \times 4{,}187 \times 48 - 42$

$= 602{,}9$ kW

M = Mass flow kg/s

Cp = Thermal capacity kJ/kg

Δt = Delta temp °C

Evaporator duty

$:q = M \times CP \times \Delta t$

$= 16{,}6 \times 4{,}187 \times 17{,}1 - 11{,}6$

$= 382{,}2$ kW

$:q = 16 \times 4{,}187 \times 15 - 7$

$= 535{,}9$ kW

M = Mass flow kg/s

Cp = Thermal capacity kJ/kg

Δt = Delta temp °C

Heat Balance

Condenser duty = evaporator duty + Total compressor input power

502,4 = 382,2 + 150	602,9 = 535,9 + 158
502,4 = 532,2	602,9 = 693,9
= 5,6%	= 13%

A typical figure for this is 5%.

Overall Compressor COP

Condition 1	**Condition 2**
$= \dfrac{\text{Evaporator duty}}{\text{Compressor input power}}$	
$= \dfrac{382,2}{150}$	$= \dfrac{535,9}{158}$
= 2,5	= 3,39

This indicates that for every 1 kW of input power to the compressor only 2,5 kW and 3,39 kW of cooling respectively is obtained at the evaporator. A typical figure for an underground plant would range between 3 and 3,5.

Actual Compressor COP

Condition 1	**Condition 2**
$= \dfrac{\text{Evaporator duty}}{\text{Compressor output power}}$	

=	$\frac{382,2}{127,5}$	=	$\frac{535,9}{134,3}$
=	2,99	=	3,99

This means that for every 1kW of power delivered by the compressor motor 2,99 kW and 3,99 kW of cooling respectively is obtained at the evaporator. Typical figures for this analysis would range between 5 and 6.

Above figures considered low.

Overall Compressor PCR

$$= \frac{\text{Compressor motor input power}}{\text{Cooling at evaporation}}$$

Condition 1		**Condition 2**	
=	$\frac{150}{382,2}$	=	$\frac{158}{535,9}$
=	0,39	=	0,29

This means that for every 1 kW of cooling obtained at the evaporator 0,39 kW and 0,29 kW of input power respectively must be supplied to the compressor motor.
0,31 is considered a typical value.

Line Losses

Losses between the chiller and the coils are negligible as they are within 10m.

Duty of Cooling Tower / Heat Rejection Coils: Air Circuit

BP = 102,5 kPa

V = 30 m^3/s

M = Q x W = 33,6 kg/s

Tai = 29,5 / 33,9°C

Tao = 32,7 / 44,9°C

From 102,5 kPa Psychometric chart:

Sigma heat of air entering = 93 kJ/kg

Sigma heat of air leaving = 109,2 kJ/kg

Apparent specific volume = 0,893 m^3/kg $w = \frac{1}{0,893} = 1,12\ kg/m^3$

Duty q = m x Δs

= 33,6 x 16,2

= 544,3 kW

This figure compares well with the condenser duties considering that line losses do occur.

Finally, this analysis once again indicates that the condenser is not capable of removing the heat created in the evaporator, due to insufficient condenser cooling capacity.

Chapter 10
MODIFICATIONS

In this chapter, the modifications required through lessons learnt is discussed.

10.1 Modifications

As with all new projects, during commissioning one needs to "iron out" the problem areas and optimise the performance. The installation of the MRP was no exception to this rule. Before one modifies one needs to know the problems. This was recorded as follows:

10.2 Main Problems

The main problems that initially created poor reliability can be attributed to:

- Poor installation practices
- Inferior make of pipe and coupling arrangements.

10.3 Lessons Learnt

The lesson learnt from the piping problem is that galvanised flanged pipes instead of high-density polyethylene (HDPE) should be installed in all instances to prevent the poor reliability of couplings coming apart and excessive leakage through these joints.

A further problem attributed to poor piping installation is the intake of air through leaking joints, specifically at the top of the raise. The entrance of air was exacerbated due to the high velocity of the water following the air pocket which was formed. This acceleration caused a negative internal pressure creating an intake of air on each pass of the air column.

This problem was overcome by introducing modified air bleed valves which had non-return valves installed to prevent any possibility of air intake.

Fortunately, the evaporator reticulation did not cause any of the above problems, and this is due to its close proximity to the plant.

10.4 Additional Observations

Due to the difficult piping layout through the non-operational raise, it was apparent that if a problem arose on the condenser line, no person would investigate the cause of the problem due to the difficult climb up the raise.

In order to overcome this situation and to assist in fault finding, a monitoring system to record the condenser and evaporator coils was recently installed. The recordings on both condenser and evaporator cars are for water volume, pressure and temperature.

The condenser also has a spray system operated by solenoid valves as previously explained.

The air temperature is also recorded and airflow switch is incorporated to depict that air is actually flowing across the coil.

This was deemed necessary rather than a fan motor indication due to a previous incident of a fan being operational albeit that the impeller had disintegrated due to poor repair procedures.

The total monitoring system will be able to communicate to surface by the telemetry system, which will provide the valuable information required to the environmental department.

10.5 Further Improvements

A further modification that will undoubtedly improve the reliability of the system is the installation of water boxes fitted at the top of the raise.

The box is a simplistic reservoir, which will be constantly fed by the service water via a check valve. The control of the water intake will be by a float level arrangement, which will only activate if the water level drops due to air ingress.

The resultant effect will be that no ingress of air can take place, as constant make-up water will be available which will automatically ensure "topping up".

Chapter 11
CONCLUSIONS

In this chapter, the conclusions of this project are summarised.

11.1 Conclusion

Both the static and dynamic assimilations predicted a shortfall in the planned condenser capacity. However, after due consideration the following constraints led to the installation of only three condenser coils:

- The size of the excavation at the reject point
- The use of standard off-the-shelf heat exchangers in order to minimise the cost
- The limited available air volume.

This decision was made knowing the impact on efficiencies and lower expected duty. The main issue though was to prove the concept. The "tweaking" of the system could take place at a later stage.

With reference to the initial project objectives, it is with confidence that we can affirm that all the objectives were met. A MRP has been built, commissioned and optimised, and is now another means of cooling available for implementation for the ventilation engineer.

As can be deduced, the closed loop underground remote refrigeration plant is well suited for underground installation. This is attributed to the following:

- The flexibility afforded for heat rejection
- Its physical size, requiring no excavation
- The condenser cooling coils can be located kilometers away where there is heat rejection capacity.
- The excellent positional efficiency
- Ease of installation

This project has received much attention locally and abroad, and has resulted in the planned installation of seven identical units within the mining environment.

Finally, a cooling system has been designed and proven, which can be used to supply or supplement the cooling need in remote areas of a mine.

11.2 Additional Benefits

The spray rings proved so effective in terms of keeping the coils clean that it has been incorporated as the standard for all future installations.

The cost of an MRP's total installation in today's terms of money stands at R3 600 per kW (cooling). The current cost of the cheapest form of cooling[27], namely surface bulk air-cooling is R7 000.00 per kW (cooling).

Not only is the MRP nearly half the price, but it also has a positional efficiency ranging from 80 to 100% (depending on configuration), whereas a surface bulk air cooler has a positional efficiency of approximately 30%.

Considering that environmental control in deep hot mines accounts for up to 40% of electricity costs[42], mine cooling techniques must be directed at improving energy efficiency.

11.3 Further Work

Results of this study have highlighted the need to pursue the following initiatives:

- The reasons for the discrepancy between the static and dynamic simulation results pertaining to the installation configuration (series versus parallel): Through better understanding, this anomaly may bring a new perspective to thermodynamic heat flows. At least, it would support current knowledge in this complex scientific area.
- The practicality of taking the evaporator cooling coils into the stope reef horizon: This would result in a 100% positional efficiency and pave the way for the inevitable.

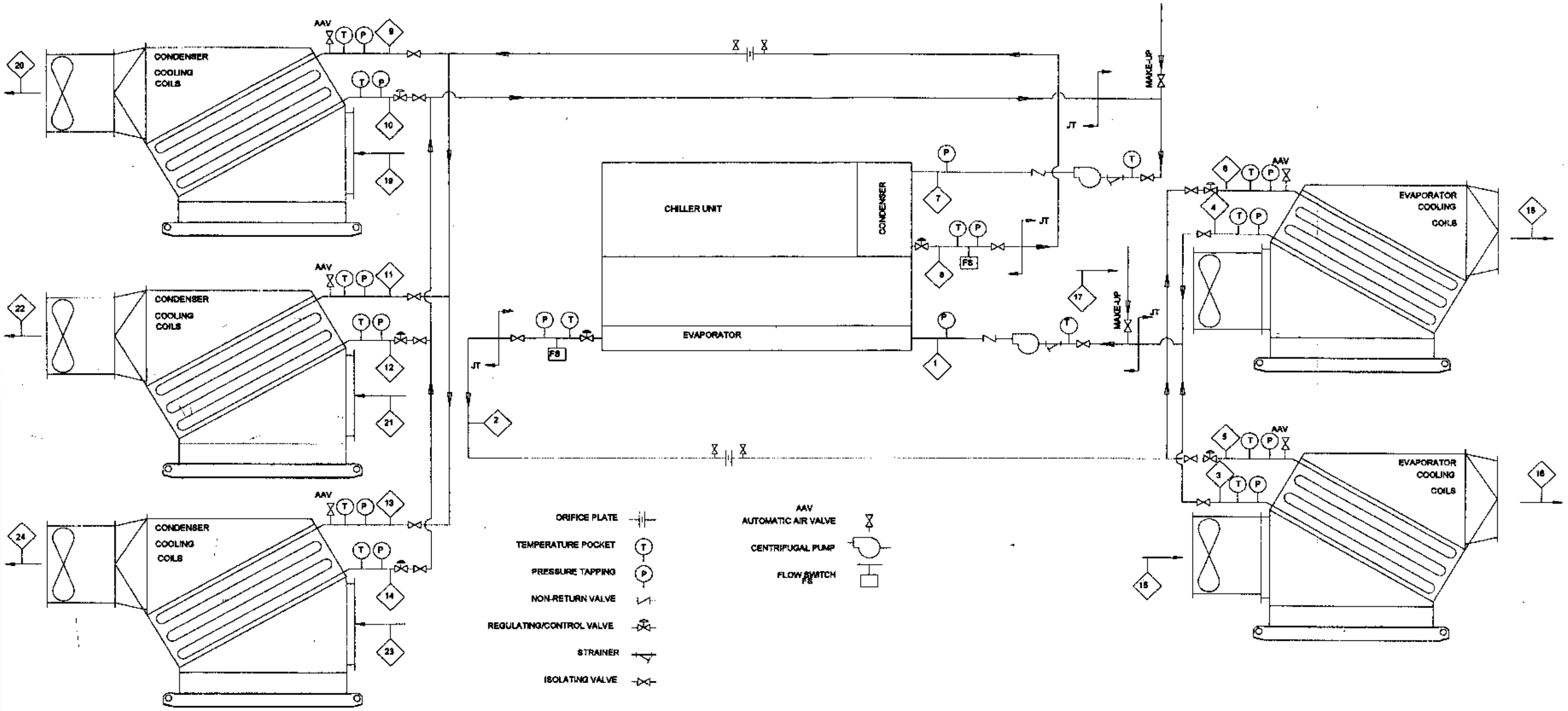
11.4 Closing Remarks

In mine ventilation terms, there is only five degrees difference between "heaven" and "hell". At 27,5°C the environment is heaven with no adverse effects to productivity or human health. At 32,5°C the environment is "hell" with poor output and a high propensity to adverse human health.

From the above statement, it is clear that even a 1°C reduction in temperature is of immense value to the cooling regime employed in a mine. To achieve 4,3°C in close proximity to the workplace, as has been proven, is outstanding.

It is with confidence that we report that this exercise has been a resounding success. Not only did we meet the project's objectives, but also more importantly a cheap system of ventilation refrigeration has been proven, which will greatly assist ventilation engineers in their quest to provide a working environment which meets the physical and mental health requirements of the underground workers.

ANNEXURE B



WATER READINGS	1	2	3	4	5	6	7	8	9	10	11	12	13	14
TEMP(°c)	14.9	7	14.9	14.7	7	7	43.7	50	50	43.7	50	43.7	50	43.7
PIPE SIZE(mm)	100	100	100	100	100	100	100	100	100	100	100	100	100	100
FLOW(Kg/s)	14	14	7	7	7	7	24	24	6	8	8	8	8	8

AIR READINGS	15	16	17	18	19	20	21	22	23	24
WET BULB TEMP(°c)	23	14.4	23	14.4	26.9	30.4	26.9	30.4	26.9	30.4
DRY BULB TEMP(°c)	27.5	14.8	27.5	14.8	29.9	46	29.9	46	29.9	46
FLOW(Kg/s)	8.37	8.37	8.37	8.37	13.81	13.81	13.81	13.81	13.81	13.81

No	PM SHEET No	DATE	No	PM SHEET No	DATE	SYM	DRG. No	SYM	DRG. No

REVISION

JIGS TOOLS & FIXTURES

MATERIAL SPECIFICATION

HEAT TREATMENT SPEC.

FINISH SPECIFICATION

TOLERANCES UNLESS OTHERWISE STATED		
0,5-6	0,1	±
6-30	0,2	±
30-120	0,3	±
120-400	0,5	±
400-1000	0,8	±
1000-2000	1,2	±

ALL DIMENSIONS IN mm UNLESS OTHERWISE STATED

JT

JOULES TECHNOLOGY (PTY) LTD

CLIENT

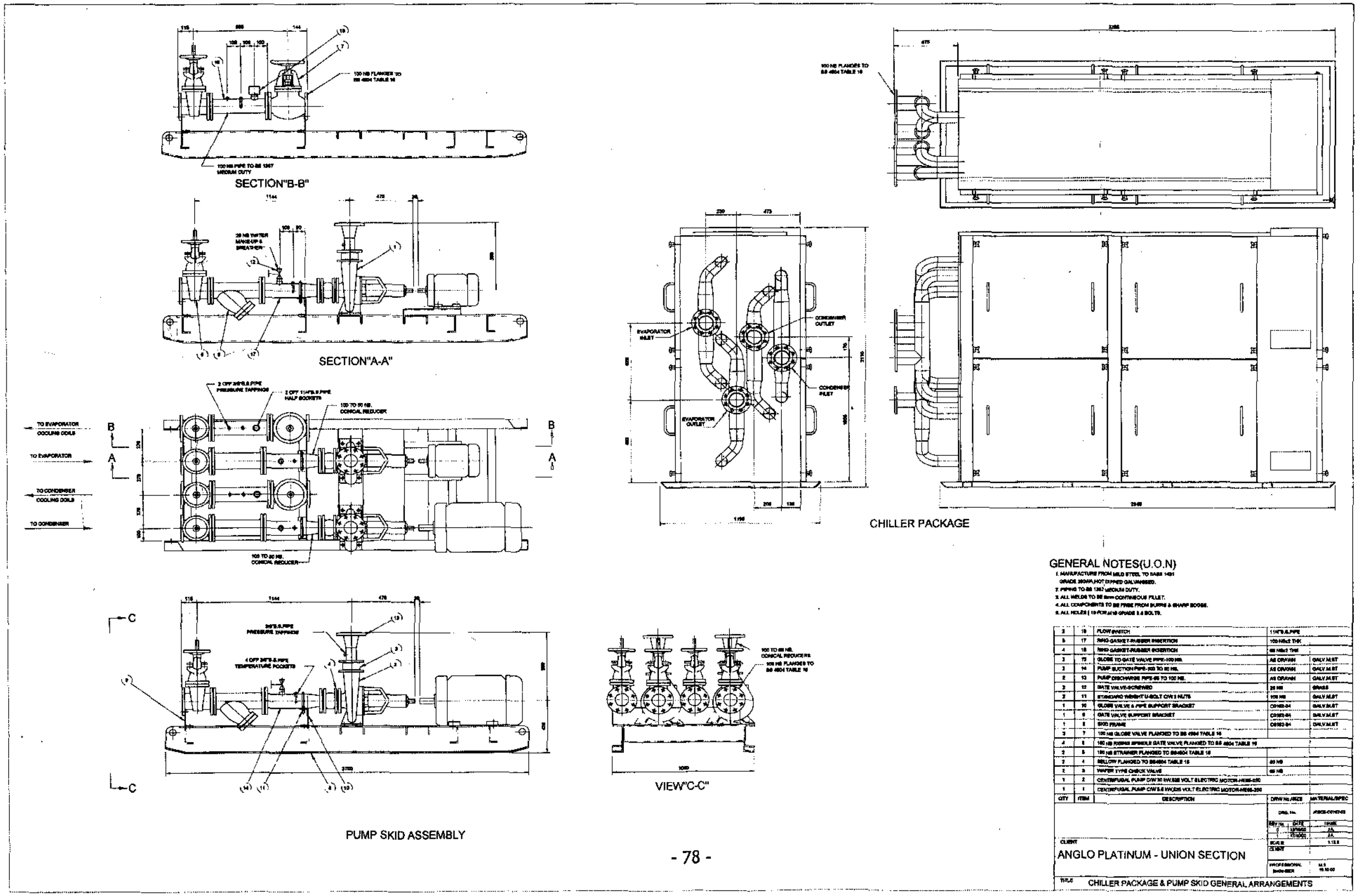
ANGLO PLATINUM - UNION SECTION

TITLE

APPROVED DESIGN CHILLER SCHEMATIC

DRG. No.	JTECE-C0162-01	
REV No.	DATE	NAME
0	13/10/00	J.A.
1	17/10/00	J.A.
SCALE:		1:12.5
CLIENT		
PROFESSIONAL ENGINEER		M.S 19.10.00

ANNEXURE A


SECTION"B-B"
SECTION"A-A"
PUMP SKID ASSEMBLY
VIEW"C-C"
CHILLER PACKAGE
EVAPORATOR INLET
CONDENSER OUTLET
CONDENSER INLET
EVAPORATOR OUTLET
GENERAL NOTES(U.O.N)
FLOW SWITCH
GLOBE TO GATE VALVE PIPE-100 NB.
PUMP SUCTION PIPE
PUMP DISCHARGE PIPE
GATE VALVE-SCREWED
GLOBE VALVE & PIPE SUPPORT BRACKET
GATE VALVE SUPPORT BRACKET
SKID FRAME
WAFER TYPE CHECK VALVE
QTY
ITEM
DESCRIPTION
MATERIAL/SPEC
CLIENT
ANGLO PLATINUM - UNION SECTION
TITLE
CHILLER PACKAGE & PUMP SKID GENERAL ARRANGEMENTS

LIST OF REFERENCES

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