8. Simulations - cost implications of identified parameters

8.1. Introduction

This section formed part of the scope of Task 6.1.1 of the Deepmine investigation. The main author was Mr. W Marx of CSIR Miningtek (Marx *et al.*, 2000). This section has been included with the permission of Mr. Marx and is given for the purpose of providing detailed background. The results of this investigation by Mr. Marx led to the investigation into the recirculation of air for deep mines, which was done by the author of this dissertation/report.

Based on the information previously gathered, it was necessary to use a simulation model that would include as many as possible of the parameters identified, so as to ensure that the expected working conditions and related costs could be accurately represented. The methodology that was followed to establish the simulation model, and other parameters related to it, was as follows:

- ❖ Determine design criteria and their applicable ranges
- ❖ Construct a representative mine layout to be modelled, stating all assumptions and taking into account the comparability of the model with other alternatives in ultra-deep-mine design
- ❖ Develop a simulation network using the program ENVIRON 2.5 in order to:

  - Establish the air flow and air thermodynamic properties to determine the cooling requirements
  - Determine the cooling requirements for different design criteria by varying the reject wet-bulb temperature and the stope face air-flow velocity
  - Apply the same process for various mining methods and cooling strategies to determine the effect of method on relative costs
  - Assess the results obtained and comment on the comparability of the modelled design with alternative designs for ultra-deep mines
  - Assess the results and comment on the effect of design criteria on mine ventilation and cooling costs
  - Formulate conclusions and recommendations.

8.2. Design criteria identified and range of values assessed

In all, ten different thermal design criteria applicable to ultra-deep mining have been identified and all of these govern either wet-bulb temperature or air velocity, or both to varying extents. From the investigations done, it became obvious that ACP is the most appropriate thermal design criterion for ultra-deep mines. It takes cognizance of most of the relevant thermal criteria and allows various combinations of wet-bulb temperature and air velocity to be applied in providing a specified level of environmental cooling power. This would, in general terms, make allowance for higher wet-bulb temperatures in areas with relatively high air velocities (e.g. main intake airways), creating opportunities to control costs through reducing the amount of air cooling.

The cost of providing a specific thermal environment is governed mainly by the capital and operating costs for the required cooling and ventilation systems. Design and performance specifications for ventilation and cooling systems are, in turn, determined mainly by the total cooling and air-flow volume required to achieve a specific wet-bulb temperature and air velocity.

Due to the specific layouts of underground mines and practical difficulties in controlling their ventilation, it is commonly accepted that actual wet-bulb temperatures over the extent of all working areas will vary by as much as 2°C above and below the design temperature. However, during the present study it was assumed that the design wet-bulb temperature is the maximum allowed and that the "real" design temperature will therefore be 2°C lower than the design value. The design wet-bulb temperatures considered were: 25, 27, 29 and 31°C (23, 25, 27 and 29 ±2°C, in practical terms), and the stope face air velocities assessed were: 0,5; 0,75; 1,0; 1,25 and 1,5 m/s. The model mine was assumed to be a green field operation, and the refrigeration plant and fan duties were adjusted to obtain the required design conditions.

8.2.1. Representative mine layout for simulation

In constructing the representative mine layout for modelling, all assumptions were carefully evaluated to ensure the comparability of the model with other design alternatives potentially applicable to ultra-deep mining. The comparability of the chosen simulation model will be discussed, together with the results for each simulation or set of simulations, as appropriate.

ENVIRON 2.5 was used to determine mine ventilation and cooling requirements, the details affecting those requirements, and the details of the ventilation and cooling systems themselves. In addition, electrical power requirements for various mine cooling and ventilation strategies were determined. The major design parameters used for the model mine were:

- Average mining depth of 4 900 m below surface
- Total monthly production of 100 000 tons reef and 15 000 tons waste
- Three production levels
- Summer ambient air intake conditions
- Geographical location in the Carletonville area.

The following are the more important assumptions and conditions applied to the simulation model:

- Refrigeration plants are all situated either on surface or underground, and are of the conventional type. Ice systems and water vapour refrigeration were not considered, and it was assumed that the same comparative results would have been obtained had these types of plants been used. Similarly, various combinations of surface and underground installations were not considered.

- The physical locations of underground bulk and spot air coolers in the simulation model were optimised to ensure that the design reject temperature was not exceeded anywhere in the intake airways or workings.

- Outlet air temperatures at underground coolers and service water temperatures/quantities applied during the simulations were determined to be as practicable and realistic as possible, based on practical knowledge and experience.

- The assumed efficiencies of the refrigeration plants, energy-recovery turbines, fans and pumps used to calculate operating costs were based on practical knowledge of such installations.

- The percentages for air leakage simulated in the intake airways and stopes were based on practical knowledge and experience and, hence, can be regarded as realistic.

- Intake airway sizes were optimised to ensure realistic air velocities and pressure losses.

- The ventilation system for the model mine did not include recirculation circuits.

- Other miscellaneous assumptions were based on knowledge of the mining methods currently being considered for ultra-deep operations, on practical experience and on measured data; they include stope face advance rate, stoping width, artificial heat loads and stope layouts.

Two mining methods were assessed to determine the comparability of the simulation model, viz. scattered and longwall mining. Table 8.2.1 summarises the cost value inputs used in the calculations for the categories of capital, maintenance and power costs. The cost calculations include all capital and operating costs for the ventilation and cooling systems, but exclude costs for the monitoring and control of ventilation and cooling. In other words, the costs associated with auxiliary ventilation systems and control measures such as doors, brattices and seals are not included. Electricity costs were based on a rate of R0,13 kWh, and capital costs were calculated on an annual basis at an interest rate of 12% over a 20-year period.

Table 8.2.1
Inputs to simulation model for estimated annual capital and operating costs of ventilation and cooling systems

Equipment	**Capital cost**	**Maintenance and operating cost**
Conventional surface refrigeration plant	R3 300/kW cooling	R100/kW cooling
Conventional U/G refrigeration plant	R4 500/kW cooling	R190/kW cooling
Main surface fans	R2 500/kW electricity	R80/kW electricity
Pelton energy recovery turbines	R2 300/kW electricity	R80/kW electricity
U/G bulk air coolers	R850/kW cooling	R10/kW cooling
U/G spot air coolers	R1 200/kW cooling	R15/kW cooling
Pumps	R2 500/kW electricity	R80/kW electricity
Pipes: 100 mm (Shaft / U/G reticulation)	R350 / R180/m	Included in refrigeration maintenance cost
Pipes: 200 mm (Shaft / U/G reticulation)	R940 / R242/m	
Pipes: 300 mm (Shaft / U/G reticulation)	R1 650 / R350/m	
Pipes: 400 mm (Shaft / U/G reticulation)	R2 100 / R500/m	
Pipes: 600 mm (Shaft / U/G reticulation)	R3 200 / R800/m	

8.3. Costs: Results and discussion

It was confirmed at an early stage of the investigation that heat would be the critical environmental factor in ultra-deep mining and that most other aspects of the environment would remain essentially unchanged in comparison with the situation prevailing at current mining depths. One obvious exception is barometric pressure, aspects of which did not form part of this investigation. It is therefore not considered further in this dissertation/report, except as a factor in the design of ventilation networks. The results from the findings of the various simulations are presented and discussed below.

8.3.1. Cost implications of thermal standards and limits

Analyses were performed to determine the financial impact of varying certain aspects of the thermal environment, including air velocity for a given wet-bulb temperature, wet-bulb temperature for a given air velocity, as well as the effect of various mining methods and cooling strategies.

8.3.1.1 Effect of design reject wet-bulb temperatures vs. constant stope face air velocity

Two sets of simulations were conducted during this part of the investigation. The purpose of the first simulation was to determine the impact of various design reject wet-bulb temperatures on environmental control costs, while that of the second was to confirm that the percentage variation in costs attributable to changes in wet-bulb temperature is approximately constant for different stope face air velocities. The costs for the model mine were calculated over the selected range of wet-bulb temperature (25 to 31°C) for air velocities of 1,0 and 1,5 m/s. It was assumed that refrigeration plants are situated on surface, that a Pelton wheel energy-recovery system (70 % efficiency) is in place and that a scattered mining method is used. Table 8.3.1.1 summarises the input parameter permutations and the simulation results obtained.

<u>Table 8.3.1.1</u>
Impact of reject wet-bulb temperature on annual costs for two different stope face air velocities

Stope face air flow velocity of 1 m/s						
Design T_{wb} (°C)	**Total cooling (kW)**	**Cooling Capex* (kR)**	**Ventilation Capex (kR)**	**Cooling Opex* (kR)**	**Ventilation Opex (kR)**	**Total cost (kR)**
25	53 967	45 815	5 385	55 803	19 630	126 633
27	48 850	42 284	5 385	50 510	19 630	117 809
29	43 699	38 729	5 385	45 181	19 630	108 925
31		38 251	34 970	5 385	39 545	19 630
Stope face air flow velocity of 1,5 m/s						
25	59 632	49 799	10 437	61 679	38 045	159 960
27	52 774	45 098	10 437	54 584	38 045	148 164
29	45 887	40 344	10 437	47 457	38 045	136 283
31	38 931	35 541	10 437	40 258	38 045	124 281

*Capex = capital expenditure, Opex = operating expenditure

In the comparison of the cost figures there was an indication that the ventilation and cooling costs for the mine modelled varied by approximately 30% for a 6°C change in wet-bulb temperature. This effect can be expected for any ultra-deep mine where stope face air velocities range from 0,5 to 1,5 m/s.

From the results it was also obvious that the absolute magnitude of additional heat flow caused by reducing the ventilation air temperature will be approximately constant at all depths. However, in terms of the proportional increase in total cooling requirements caused by that temperature reduction, the relative impact would be significantly less at greater depths. This can best be illustrated by means of the hypothetical example considered below.

The major contributor to heat load in deep mines is the rock mass, which adds heat to the ventilation air in direct proportion to the difference in temperature (ΔT) between the rock and the air. For a typical mine in the Carletonville area with an average working depth of 2 000 m, ΔT is 14°C (VRT of 40°C minus air temperature of 26°C). For a similar mine with the same air temperature, but having an average working depth of 4 900 m, ΔT would be 43°C, as a result of the VRT being 69°C. A reduction of 4°C in ventilation air temperature will result in a 30% increase in ΔT for an average working depth of 2 000 m, while at 4 900 m the same reduction in air temperature would increase ΔT by less than 10%. Accordingly, the cost of incremental reductions in air wet-bulb temperature at great depth, relative to that of maintaining it at the current limit for routine work (32,5°C) is of secondary significance.

8.3.1.2 Effect of various stope face air velocities for a constant wet-bulb temperature

Two sets of simulations were conducted during this part of the investigation. The purpose of the first simulation was to determine the impact of varying the design stope face air velocity, while that of the second was to confirm that the percentage variation in costs attributable to changes in air velocity is approximately constant for different reject air wet-bulb temperatures. The costs of environmental control were calculated for the selected stope face air velocities and at the maximum or reject wet-bulb temperatures of 27 and 31°C. It was again assumed that refrigeration plants are situated on surface, that an energy-recovery system is in place and that a scattered mining method is used. Table 8.3.1.2 summarises the input parameter permutations and the simulation results obtained.

Table 8.3.1.2

Impact of stope face air velocity on annual costs for two different reject wet-bulb temperatures

Face velocity (m/s)	Total cooling (kW)	Cooling Capex* (kR)	Ventilation Capex (kR)	Cooling Opex* (kR)	Ventilation Opex (kR)	Total cost (kR)
Reject air wet-bulb temperature of 27°C (25°C design)						
0,5	43 855	39 009	1 656	45 367	6 038	92 070
0,75	46 241	40 404	3 617	47 806	13 186	105 013
1,0	48 850	42 284	5 385	50 510	19 630	117 809
1,25	50 845	43 723	8 021	52 580	29 240	133 564
1,5	52 774	45 098	10 437	54 584	38 045	148 164
Reject air wet-bulb temperature of 31°C (29°C design)						
Face velocity (m/s)	**Total cooling (kW)**	**Cooling Capex* (kR)**	**Ventilation Capex (kR)**	**Cooling Opex* (kR)**	**Ventilation Opex (kR)**	**Total cost (kR)**
0,5	36 807	33 811	1 653	38 035	6 024	79 523
0,75	37 264	34 202	3 453	38 514	12 588	88 758
1,0	38 251	34 970	5 385	39 545	19 630	99 530
1,25	38 692	35 331	7 529	40 005	27 444	110 309
1,5	38 931	35 541	10 437	40 258	38 045	124 281

*Capex = Capital expenditure, Opex = Operating expenditure

The results highlighted the effects of auto-compression, air density and the increased resistance associated with higher stope face air velocities. The increased enthalpy of downcast air due to auto-compression is 9,8 kJ/kg per kilometre of depth. Therefore, the increase in cooling requirements attributable to auto-compression is directly related to the mass flow of air circulated through a mine and the depth of the workings. For the model mine under consideration, the air-flow volumes on surface required to yield stope face velocities of 0,5 and 1,5 m/s are 270 and 810 m^3/s, respectively. The increase in air density from approximately 1 kg/m^3 on surface to 1,5 kg/m^3 at the average working depth, combined with an increase in air leakage, is a significant factor in the greater air-flow requirement. The resultant increase in air mass flow contributes approximately 30 MW of additional heat from auto-compression that must be removed by the mine's cooling system.

However, an increased air mass flow rate in the intake airways will result in a smaller temperature difference across an air cooler. (1 kW or 1 kJ/s of cooling will yield a smaller difference if the total mass air flow across the cooler is increased.) This will reduce ΔT and, thus, cooling requirements. It is evident from the results, particularly for a reject temperature of 31°C, that the increased heat load from auto-compression is virtually cancelled out by the reduced heat load in the intake airways, as a result of the increased air mass flow rate. These effects explain the very significant increase in ventilation costs associated with a greater quantity of ventilation air and, ultimately, any increase in stope face air velocity.

From the results, it can also be deduced that the percentage increase in environmental costs due to changes in stope face air velocity is approximately constant for different reject wet-bulb temperatures, and that the percentage variation can be used as a general basis for environmental design studies. The findings are also indicative of the model's comparability.

8.3.1.3 Effect of various reject wet-bulb temperatures for different stoping methods

The simulations for this part of the study were conducted by means of a global heat load estimation program and not with ENVIRON 2.5. The main purpose was to confirm the comparability of the model and, thus, to determine whether the overall phenomenon identified during the study is applicable to different mining methods. No specific results are provided for this part of the study. However, it was found that a change in mining method does not have any significant influence on the cost trends demonstrated for variations in design reject wet-bulb temperature.

8.3.1.4 Varying design stope face air velocity for different cooling strategies

The simulations conducted during this part of the study were intended to confirm that the percentage variation in costs attributable to changes in reject air wet-bulb temperature is approximately constant for different refrigeration plant locations (surface or underground) and for various cooling strategies. Table 8.3.1.4 summarises the simulation results obtained when surface plant systems are compared with underground plant systems. It was assumed that sufficient heat-rejection capacity is available for underground plant, and the same assumptions as made previously were applied for energy-recovery systems and mining method.

Table 8.3.1.4
Impact of reject wet-bulb temperature on annual costs for various cooling strategies

Refrigeration plants located on surface (stope face velocity of 1 m/s)						
Design T_{wb} (°C)	**Total cooling (kW)**	**Cooling Capex* (kR)**	**Ventilation Capex (kR)**	**Cooling Opex* (kR)**	**Ventilation Opex (kR)**	**Total cost (kR)**
25	53 967	45 815	5 385	55 803	19 630	126 633
27	48 850	42 284	5 385	50 510	19 630	117 809
29	43 699	38 729	5 385	45 181	19 630	108 925
31	38 251	34 970	5 385	39 545	19 630	99 530
Refrigeration plants located underground (stope face velocity of 1 m/s)						
25	53 967	47 251	5 385	32 176	19 630	104 442
27	48 850	43 583	5 385	29 094	19 630	97 692
29	43 699	39 891	5 385	25 934	19 630	90 840
31	38 251	35 986	5 385	22 677	19 630	83 678

*Capex = Capital expenditure, Opex - Operating expenditure

An evaluation of the results indicated that the two cooling strategies considered do not differ significantly in their impact on relative cooling and ventilation cost trends for a given design reject wet-bulb temperature. It also indicated the comparability of the simulation model, and there is no reason to expect that cost trends for other cooling-generation systems would differ significantly from these results.

8.3.2. Conclusion on costing for thermal standards and limits

Although much of the knowledge gained in cooling and ventilating mines at depths approaching 4 000 m will certainly be applicable at 5 000 m, changes in the application of cooling and ventilation methods will inevitably be required. Current practice and conventional wisdom are necessarily based on previous research and practical experience in cooling and ventilating mines that are not as deep as those presently being contemplated. The industry's experience thus far has been that small improvements in thermal conditions have been achieved through large increases in ventilation and, particularly, cooling costs. In contrast, the results of the present study indicate that reducing the reject wet-bulb temperature would have less impact on the overall cost of ventilation and cooling than increasing the stope face air velocity. Over the range of reject wet-bulb temperatures considered (25 to 31°C), cooling and ventilation costs were shown to vary by up to 30% from the base case of 31°C. While it is acknowledged that reducing temperatures in ultra-deep mines would increase the absolute cost of environmental control, the increase in relative or percentage terms would be less significant than for current mining depths. In any case, a given level of ACP in an ultra-deep mine can be provided at far lower costs by reducing wet-bulb temperature than by increasing air velocity and air quantity. It follows then that, should health, safety and productivity-related requirements dictate a design reject wet-bulb temperature of, say, 25°C, it should be feasible (in terms of ventilation and cooling costs) to meet that criterion, provided a design temperature of 31°C was economically viable in the first place.

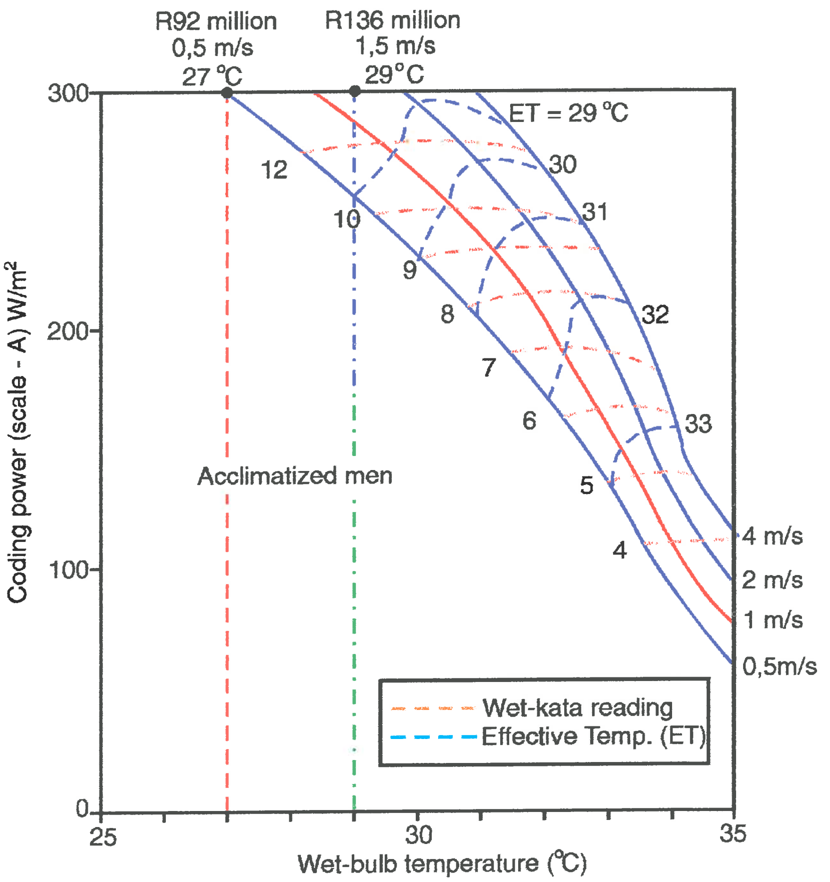

Figure 8.3.2 Annual costs for various combinations of wet-bulb temperature and air velocity that provide 300 W/m² of Air Cooling Power

Previous studies quantifying heat stress in terms of ACP have indicated that a level of 300 W/m^2 can be regarded as both safe and conducive to high productivity in nearly all instances. The projected costs for providing this level of ACP for air velocities of 0,5 and 1,5 m/s at wet-bulb temperatures of 27 and 29°C, as indicated in Figure 8.3.2, are based on the characteristics of the model mine considered. The significant cost increases associated with greater air velocities highlight the main conclusion of this analysis, namely that total air-flow quantity should be minimised and wet-bulb temperature reduced to achieve the required ACP in ultra-deep mines. Designing an environmental control system in accordance with these criteria would allow higher wet-bulb temperatures in areas that naturally have higher air velocities, e.g. intake airways, but the general approach indicated is to reduce wet-bulb temperatures through cooling, and not to increase air velocity.

The practicability of thermal design limits, including those for air velocity and wet-bulb temperature, and the physiologically/psychologically related requirements of the work force will be principal determinants of the design criteria ultimately adopted for ultra-deep mines. The investigation of worker-related factors should also receive attention.

These cost simulations demonstrated the significant impact of higher stope face air velocities on both cooling and ventilation costs. The results therefore form the basis for the recommendation that environmental control systems in ultra-deep mines should be designed to provide the required thermal environment (in accordance with the many and various criteria relevant to this critical decision) at a minimal total air mass flow rate. With possible cost variations as demonstrated by the current findings being as great as 60% (relative to the base case of 0,5 m/s), minimising the air mass flow rate should certainly be an integral part of an ultra-deep mine's ventilation strategy.

A logical extension of minimising air mass flow rate would be the inclusion of controlled air recirculation techniques into ultra-deep ventilation and cooling strategies. Recirculation can enable reductions in fan power requirements of up to 60% to be made, according to previous research (Willis *et al.*, 1997). Furthermore, recirculation could greatly reduce the impact of auto-compression on total heat load. However, caution should be exercised in reducing the overall quantity of air supplied, as a reduction may introduce practical difficulties in providing the required cooling, as well as impose constraints or requirements for additional control measures where certain types of equipment are used, e.g. diesel-powered machinery. The beneficial effects of recirculation on environmental control costs, together with the potential risks that its inappropriate implementation could impose, indicated the need for a separate investigation, which is dealt with in the following section.

It must again be emphasised that the cost analysis conducted during the present investigation was of a comparative and relatively simplistic nature, as a result of the need to use a single simulation model. Accordingly, these findings should be interpreted and applied with circumspection but, more importantly, also validated for specific mine designs as they are developed.

In a similar vein and to underscore previous comments, the heat stress limits ultimately adopted for ultra-deep mining should include criteria for work performance, in order to ensure that productivity levels contribute to the profitability of such mining projects.

9. Recirculation of mine air

9.1. Introduction

The provision of acceptable environmental conditions is an important aspect of all underground mining activities. As mines have (and will still) extended deeper and working areas have become more remote, the costs and practical problems of ensuring adequate supplies of ventilation air at the appropriate places have become a matter of serious concern. The potential benefits of recirculating air underground are great, but this practice has hardly been used in South Africa in the past, largely because of concerns of safety. Work carried out in the early 1980s indicated that recirculation of air can in fact be regarded as a safe, reliable and effective procedure, provided that certain precautions are taken (Burton, October 1984).

In South African gold mines in the past very little use has been made of controlled recirculation. A trial was conducted at East Rand Propriety Mines in 1948, when a portion of the upcast air was cooled and then mixed with the downcast air (Gorges, 1952). In 1973, Holding reported on a recirculation scheme in a single stope (Holding, 1973). An intensive investigation into the use of controlled recirculation was pursued, which resulted in a trial conducted at Loraine Gold Mines Ltd (Burton *et al.*, 1984; Fleetwood *et al.*, 1984). A great deal of success was achieved with this trial and other subsequent studies followed.

Although there are a considerable number of technical papers available concerned with the use of controlled recirculation in mines, they suffer from two main disadvantages. First, the majority of the papers are concerned with small-scale recirculation systems in British collieries and secondly, each paper tends to deal only with individual aspects of recirculation. Burton in his report covered all the relevant aspects of recirculation of air for deep gold mines in full detail (Burton, October 1984).

9.2. The role of controlled recirculation

9.2.1. Background

Control of the underground environment relies on the use of ventilation air to perform a wide variety of functions. An understanding of the role of controlled recirculation can therefore only be gained if these functions are clearly defined. There are four major reasons for which air is required in mines. First, an adequate supply of oxygen is needed to support the necessary complement of workers and also to supply any diesel locomotives or other diesel trackless equipment that may be operating. The amount of air required for this reason, although vital, is small compared with that demanded for any of the other reasons and does not require special consideration.

Secondly, sufficient air is required to dilute and remove noxious and inflammable gases so that gas layering is minimised and acceptable concentration limits are not exceeded. An adequate supply of air is also needed to remove blasting fumes timeously.

Thirdly, air is essential for the dilution and removal of respirable dust since the measures used to suppress dust at its source are not always completely successful. A large number of dust filters are used underground, particularly at tipping and transfer points, and air is also required to convey the dust to these filters.

Finally, although chilled water can be used to absorb some of the heat load in working areas, most of this heat load will be absorbed by air. In shallow mines the effects of auto-compression and heat transfer from the rock are small, and downcast ventilation air, without any refrigeration, is capable of absorbing the underground heat load, while maintaining reject temperatures at an acceptable level.

As depths increase, this capability is progressively reduced. In summer at a depth of about 2°500 m below surface, downcast ventilation air reaches a wet bulb temperature of 28°C. Clearly, if the desired reject temperature is also 28°C, the downcast air has lost all its capacity for cooling. Therefore, below this depth, all the heat produced underground must be removed by refrigeration. However, although the usefulness of downcast ventilation air decreases with increasing depth, air is still required for two important cooling purposes: as a medium for distributing the necessary amount of refrigeration, and also as a means of removing heat from any underground refrigeration plant.

9.3. The use of controlled recirculation

Controlled recirculation of ventilation air can be defined as the reintroduction of a portion of the return air from an area into the intake of that same area. The term 'area' is used in a general sense and can mean a development end, a single stope or a whole section of a mine. For illustration purposes a simplified 'controlled recirculation system' is shown in Figure 9.3.

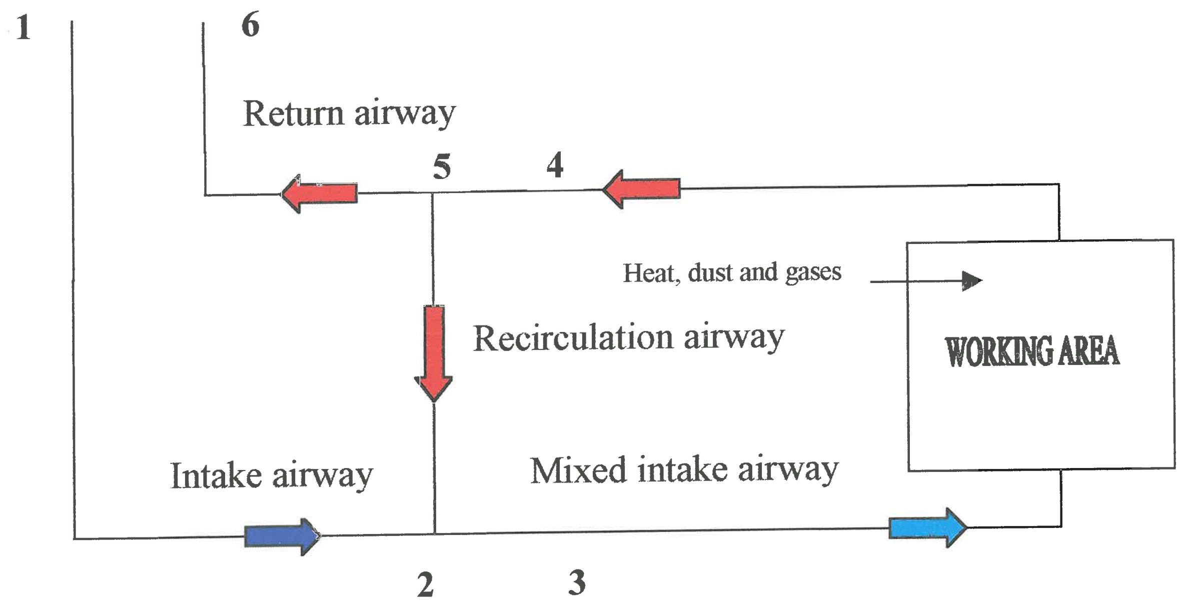

Figure 9.3 ***Simplified section of a controlled recirculation system for a mine***

In the figure 'fresh' intake air flows along the path from point 1, passes through the working area and is finally rejected at point 6. Meanwhile recirculation takes place along the route 2-3-4-5-2. It can be observed that the rate at which air is recirculated is dependent on the quantity of the intake air; essentially, recirculation is merely the enforced circulation of the fixed volume of air contained in the area 2-3-4-5-2, and this can be done with any quantity of intake air.

It will be apparent that the effect of superimposing recirculation onto the intake air flow is to increase the flow rate of air at point 3. Clearly, air velocities will also increase throughout the working area and this increase has two beneficial effects: increased values for cooling power without any changes in the actual air temperatures (Stewart, 1982) and increased air velocities which can also minimise gas layering (Bakke *et al.*, 1964).

However, the most important advantage of an increased air flow is that it provides a means of more conveniently distributing refrigeration within an area. When air flow is restricted, air temperatures will increase rapidly for a given heat load and repeated cooling using small air coolers becomes necessary. With an increased air flow created by recirculation, the rate of temperature rise decreases. The cooling needed for a specific area can then be achieved by means of a single bulk air cooler. It is important to recognise that controlled recirculation can only be used as a means of distribution and not as a source of refrigeration. The amount of refrigeration required in an area is virtually independent of any controlled recirculation that may take place.

Controlled recirculation on its own does not affect the return-air dust concentration. This concentration is largely dependent on the intake air quantity and on the amount of dust that is produced and becomes airborne within the working area. However, a situation could arise in which the intake quantity is insufficient to maintain the return air dust concentration at an acceptable level. Controlled recirculation, with bulk air dust filtration, could then be used as an alternative to increasing the intake air quantity.

So far, the beneficial effects of controlled recirculation, *per se*, and it's potential role in distributing refrigeration and controlling dust levels have been described. It is now necessary to identify those functions that controlled recirculation cannot perform.

Clearly, controlled recirculation cannot provide any oxygen that is required. The intake air must supply the oxygen. It is also important to note that the return air concentrations of any noxious or inflammable gases that are produced in a working area (excluding those produced by the blast) are not affected by controlled recirculation. Such concentrations are largely dependent on the intake air quantity and the amount of gas produced within the area. Furthermore, although blasting fumes are a special case for consideration, controlled recirculation has little or no effect on the rate of removal of such fumes; again, the overriding influence is that of the intake air quantity. Some practical considerations regarding the continuation of recirculation during and after a blast will be discussed later.

Finally, since controlled recirculation has no effect on the quantity of return air (at point 6, Figure 9.3), it can be of no assistance in rejecting heat from any refrigeration plant. This could have an important influence on the future use of recirculation. This aspect will also be discussed in detail later.

9.4. Effects of controlled recirculation on the environment

9.4.1. Background

There are a number of mathematical models that can predict the effects of controlled recirculation on gas and dust concentrations and on the rate of removal of blasting fumes. A simple model was also developed by Burton which showed the effect of combining controlled recirculation and bulk air cooling on the temperature levels within an area. In the models, reference is made to three air quantities. By referring to Figure 9.3 they are defined as follows: 'Fresh intake air' flows from points 1 to 2, 'recirculated air'" flows from points 5 to 2, and 'mixed intake air' flows from point 3, through the working area to point 4.

9.4.2. Recirculation model for gaseous contaminants

Burton made a conclusive statement in saying that the concentration of gaseous contaminants in the return air is not affected by recirculation. This concentration is determined by the intake air gas concentration, the intake air quantity, and the amount of gas produced in the working area. Controlled recirculation, at whatever rate, does not change the concentration at all.

However, it must be noted that the mixed intake concentration is dependent on recirculation and that the concentration increases as the recirculated fraction increases. For a given quantity of intake air, the recirculated fraction can vary only between zero and some value close to unity. If there is no recirculation, the mixed intake concentration is merely the intake concentration. As the quantity of recirculated air is increased, the recirculated fraction approaches unity and the mixed intake concentration will approach, but not exceed, the return air concentration.

In summary, the maximum or return air gas concentration is not affected by recirculation and all other concentrations in an area will remain below this level. Similar models have been developed previously, and the predictions have been confirmed on many occasions, particularly in regard to methane concentrations in United Kingdom collieries (Bakke *et al.*, 1964; Leach, 1966;Leach and Slack, 1969). In 1984 a major field trial of a controlled recirculation system was conducted at Loraine Gold Mines Ltd. (Burton *et al.*, 1984). In Figure 9.4.2 below, the predicted and measured values of carbon dioxide concentrations in the return and mixed intake air are presented. It can be seen that the predicted values are an accurate reflection of the measured values. Although they are not included here, similar results were obtained for other gases such as carbon monoxide and the nitrous fumes produced by diesel locomotives. A minor deviation from the values predicted may be obtained in the case of nitrous fumes, since nitric oxide is slowly oxidised to nitrogen dioxide.

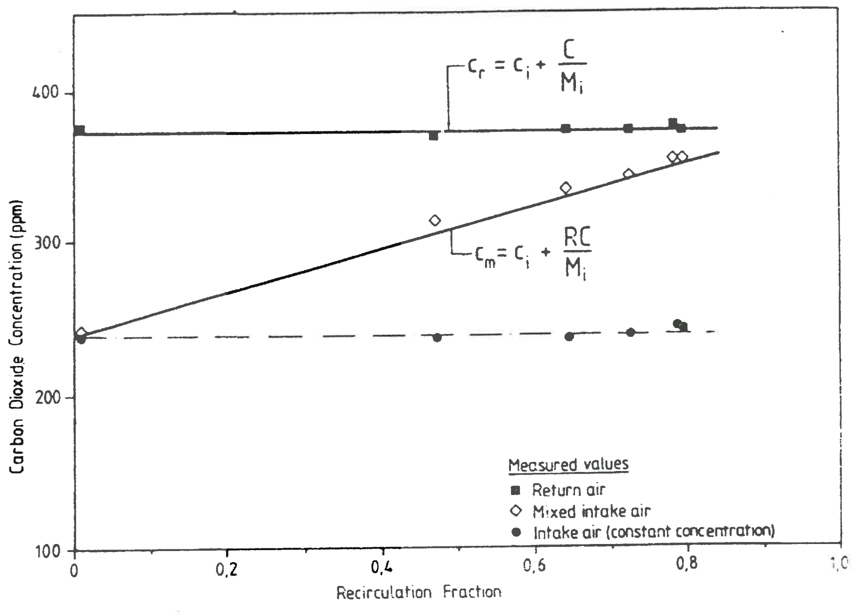

***Figure 9.4.2** Relationship between carbon dioxide concentration and recirculated fraction*

The equations as were used by Burton can also be used to predict the oxygen concentration in the return and mixed intake air. However, the value for the gas produced in the working area, C, would in this case be negative since oxygen is being consumed rather than produced. It can be seen that the minimum or return air oxygen concentration is not reduced by controlled recirculation.

Some important statements must be made regarding the effect of controlled recirculation on radon gas concentrations, and particularly on working level contamination. The equations presented in the paper by Burton can also be used to predict the return and mixed intake radon gas concentrations fairly accurately. In other words, radon gas behaves in much the same manner as the stable gases previously mentioned. A more comprehensive model was developed by Rolle (1982), which in fact showed that, due to the progressive decay of radon gas into its three daughters, return air radon gas concentrations are, in general, slightly lower with recirculation than without.

However, because of this decay, and also because controlled recirculation leads to increased residence times, working level contamination is increased by recirculation. Rolle showed that, for large quantities of recirculated air relative to intake air (R=0.8), the maximum or return air working level can be about 50% higher than the value existing when only the same quantity of intake air is supplied to an area. It should be emphasised, however, that at expected recirculated fractions of about 0,5 the increase is limited to about 25 to 30%. Clearly in uranium mines and those gold mines in which radon is present, consideration must be given to such effects. The quantity of intake air must be chosen to ensure that acceptable working levels are achieved. An increase could be largely prevented if dust filtration were to be carried out, since radon daughters become attached to dust particles.

9.4.3. Recirculation model for blasting fumes

In the paper presented by Burton, the conclusion was made that the rate of removal of blasting fumes depends largely on the quantity of intake air, and that the recirculation has either no effect or, because of better mixing, a slightly beneficial effect. In order to test the model, a series of measurements were made at the recirculation system installed at Loraine Gold Mines Ltd. An analysis was carried out on the return air, blast fume decay curves for both nitrous fumes and carbon monoxide for various quantities of intake and recirculated air. It was found that the decay curves were exponential in nature and a value for the time constant τ was determined in each case. In Figure 9.4.3 the relationship between the time constant and the intake air quantity is shown (note that the decay is inversely proportional to the time constant). It can be seen that the rate of decay and hence the removal of blasting fumes is closely correlated with the intake air quantity.

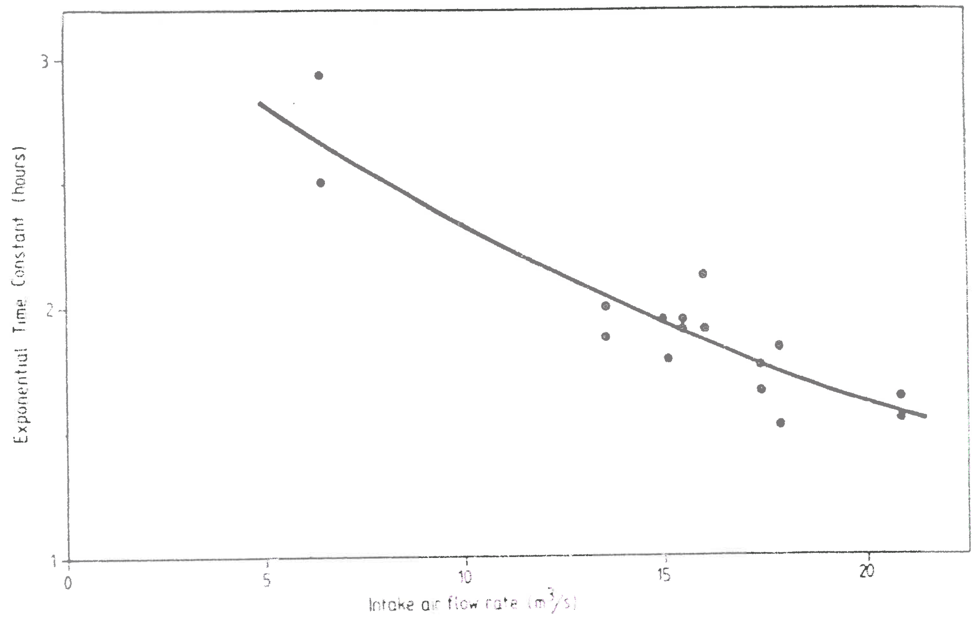

Figure 9.4.3 ***Relationship between the exponential time constant τ and the intake air quantity***

Each point in Figure 9.4.3 has a different value of recirculated air flow associated with it. The value varied from 49 m^3/s, with a corresponding variation in the recirculated fraction, R, of zero to 0,87. The figure therefore includes any effects that recirculation may also have on the blast contaminant decay. Clearly, it would have been desirable to isolate the two effects completely by independently varying the quantities of intake and recirculated air, but this was found to be difficult in practice. However, analysis shows that the quantity of recirculated air has very little effect on the value of the time constant and hence on the rate of decay. Burton also noted that considerably more data would be required to confirm the model.

Although the model has been shown to be a good representation of blast fume decay, the assumption made earlier that blasting results in a 'uniform' concentration of noxious contaminant may not be justified. In practice a dense 'plug' of fumes is usually formed and this travels along the return airway immediately following the blast. By stopping recirculation for a short period, this plug could be removed from the area completely, and the overall time for reducing noxious concentrations to acceptable levels may be decreased. This will be discussed in detail later.

9.4.4. Recirculation model for dust

The model developed for gaseous contaminants can also be applied, in general, to predict the effects of controlled recirculation on respirable (and total) dust levels. Briefly, if there is no dust filtration, the intake air quantity, the intake air dust concentration and the amount of dust produced that becomes airborne in the working area dictate the return air dust concentration. Controlled recirculation will not affect this concentration. Burton also suggested that as an alternative, air recirculation be used in combination with dust filtration.

Burton used a model to simulate various dust concentrations and concluded that the return air dust concentration would not change with recirculation and that the mixed intake air concentration would increase with recirculation. He also highlighted the aspect of dust collection efficiency and the results thereof are shown in Figure 9.4.4.

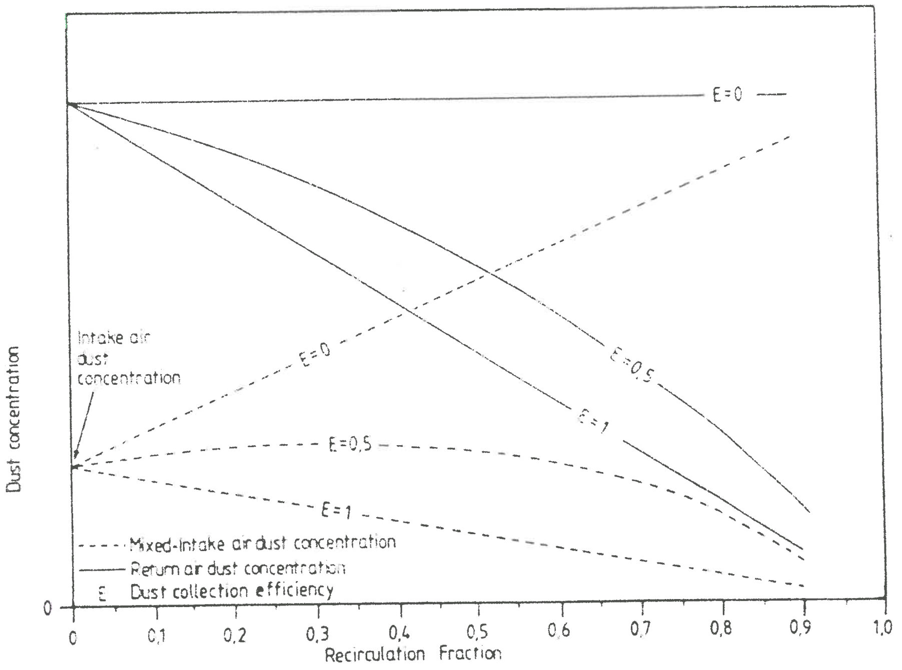

Figure 9.4.4 ***Effects of recirculation and dust filtration on the mixed intake and return air dust concentrations***

Controlled recirculation and dust filtration has been used in 'advance headings' in British collieries (Robinson, 1972; Pickering and Aldred, 1977). However, the filtration of respirable dust from large volumes of air, as envisaged in recirculation systems in gold mines, has still to be investigated (Burton, 1984).

9.4.5. Recirculation model for air cooling

The main motivation for installing recirculation systems in deep gold mines will be to provide acceptable temperatures in working areas as well as to reduce the load on the surface fans. Controlled recirculation, as mentioned before, should be regarded only as a means of more conveniently distributing the required amount of refrigeration in an area, since recirculation *per se* will not lead to lower temperatures. However, the higher air quantities that recirculation achieves can allow the use of a single bulk air cooler to maintain acceptable temperatures throughout the area.

When cooling is introduced, both the mixed intake and the return air enthalpies, and hence the temperatures, can be much reduced. At the Loraine field trial, the mixed intake wet-bulb temperature was reduced from 27,9°C to 22,3°C, and the return air wet-bulb temperature was reduced from 31,5°C to 28,4°C (Fleetwood *et al.*, 1984).

For the purpose of this investigation the various recirculation percentages, and the cooling associated with them, were calculated with the aid of the ENVIRON heat simulation program.

9.5. Summary of simulation models

Burton found that the various models that do predict the effects of controlled recirculation are in fact very reliable. The models have shown that, in general, the intake air and the recirculated air perform separate functions, and the respective air quantities that are used in a specific area must be chosen so that these functions can be achieved.

9.6. Factors affecting the introduction of controlled recirculation

9.6.1. Background

Controlled recirculation can be used as a substitute for downcast ventilation air, but there are certain issues that have to be kept in mind. One of the key aspects to consider is when recirculation of air should be prohibited and in what cases it will be beneficial to use it.

9.6.2. Reasons for using controlled recirculation

There are many factors that influence the ventilation and cooling arrangements required by a mine. An attempt is made in this study to provide a general recommendation on the introduction of controlled recirculation. There are, however, certain important factors that can indicate where controlled recirculation is likely to offer certain benefits, when compared with current ventilation practices. Burton investigated these issues and found that the minimum quantity of intake air that must be provided is that which is sufficient for:

- the provision of oxygen
- the dilution and the removal of gases during on-shift periods
- the timeous removal of blasting fumes.

This minimum value could be increased so that the intake air quantity is also sufficient for the dilution and removal of airborne dust. For shallow mines this figure can be 2,5 kg/s/kt of rock mined per month (Appelman and Schröder, 1984) and for deeper mines the figure can go as high as 6 kg/s/kt of rock mined per month. Extra air is sometimes needed to combat heat-related problems and it is this extra air that can be replaced by using controlled recirculation. The approach that Burton suggested was to limit the downcast air to not more than 2,5kg/s/kt per month, and to introduce controlled recirculation when extra quantities of air are required for distributing refrigeration.

In existing deep mines, there are already considerable practical difficulties in ensuring an adequate supply of ventilation air from surface. As depths and distances from the downcast shafts have been increased and will in future increase, so higher fan pressures are required. This results in large pressure differences between the intake and return airways, and large amounts of air are lost due to leakage. Such losses are difficult to control, but by limiting the volume of downcast air, these losses could be reduced. By using controlled recirculation, the larger mixed intake air quantities are only produced close to where the air is actually required, thus reducing the opportunities for leakage to occur. As mines are now deeper and distances from the downcast shaft have increased, the practical benefits associated with using controlled recirculation become more evident.

Although the practical benefits are considerable, the most important benefits are economic. Considerable savings in fan power could be realised for ultra-deep mines if the specific downcast air quantity could be limited to the suggested value of about 2,5 kg/s/kt per month. Limiting the downcast air quantity could also, in some circumstances, reduce the amount of refrigeration required to combat the effects of auto-compression. As mines become deeper, the combined effects of auto-compression and higher heat transfer from the rock may mean that the intake air has to be cooled.

A further consequence of not increasing the downcast air quantity is that this could delay the need for larger or multiple shafts and airways at greater depths. This is particularly relevant when considering the number of return airways and upcast ventilation shafts that are required. Although the benefits of introducing controlled recirculation in current mines can be considerable, there may be many constraints caused by established infrastructure. It is therefore important that when new ultra-deep mines are planned, the various advantages of controlled recirculation are realised by building recirculation into the design. An important aspect to remember is that controlled recirculation can be of no assistance in rejecting heat from any underground refrigeration plant. When the introduction of controlled recirculation is considered, careful thought must be given before introducing any new underground refrigeration plant into an area where recirculation is being proposed, particularly if the aim is to limit the intake air quantity.

From the work done in this study so far, it is obvious that the cost of providing air from surface becomes very expensive and that the recirculation of air almost becomes compulsory. Some of the objectives of this investigation are to determine to what extent and to what specifications recirculation should be used in ultra-deep mines..

9.6.3. Potential hazards of controlled recirculation

Underground fires are a major hazard in mines. This is particularly so in gold mines where explosives are employed and extensive use is made of wood for support. Controlled recirculation has certain implications regarding both the immediate and long-term effects of fires.

Depending on the position of the start of the fire, recirculation can have either no effect or may reintroduce the smoke generated by a fire into the stope. In the latter case, the recirculation fans will have to be stopped. In this way, an area can be immediately restored to a normal once-through ventilation system and standard fire procedures can be enforced. The controls for stopping the recirculation fans can be linked to the detecting devices and in that way they are made foolproof. These systems will, however, have to be tested on a regular basis.

It is therefore important that the available routes and associated risks be carefully considered for the various circumstances encountered. However, it is almost certain that recirculation of air will have to be stopped once a fire breaks out. Another important potential hazard associated with controlled recirculation is that of ensuring a continuous supply of intake air. If the intake air stops for any reason, there would be a gradual build-up of gases and dust within an area (Burton, 1984).

9.6.4. Prohibition of recirculation

It must be stated that there are certain instances where recirculation, whether controlled or uncontrolled, is specifically prohibited. In terms of the Mines and Works Act and Regulations, 1956, as amended, certain mines or part of mines are classified as fiery. In such circumstances recirculation is prohibited, unless an exemption from the Regulation is obtained.

It is interesting to note that although all British collieries would probably fall into this category, a total of 61 recirculation systems were operating in January 1983 (Allan, 1983). Indeed, it has also been suggested that recirculation of air could reduce methane layering in South African coal mines (Thorp, 1982; Greig 1982; Burton, 1984).

9.7. Design of a controlled recirculation system

9.7.1. Introduction

The variable circumstances encountered in mines preclude any rigid recommendation for the design of a recirculation system. However, a number of features will be common to all recirculation systems, namely the determination of the required quantities of intake and recirculated air, the arrangements for the establishment of these air flows, cooling arrangements and arrangements that will ensure that the recirculation system can be operated safely.

9.7.2. Determination of air quantities

Burton suggested that it would be inappropriate to give a recommendation on the quantity of air required. The actual value required in any area will depend on the specific circumstances encountered. There are recommended and statutory limits for certain contaminants and these must be observed. However, the final choice of the quantity of intake air for ensuring safe, acceptable conditions will rest heavily on the experience of the responsible environmental engineer. In making this choice, it can be noted that the quantity of recirculated air will not affect the maximum or return air contaminant levels. The major exception is working level contamination, which was discussed above.

9.7.3. Ventilation arrangements

For recirculation, the location of fans will to a large extent depend on specific circumstances. Such circumstances will include the availability of electrical power and the suitability of various locations with regard to tramming activities. The fans that are chosen must be capable of being adjusted to give higher air flows.

In the planning of a recirculation strategy for ultra-deep mines, one must keep in mind that the intake flow will be reduced because of an increased pressure drop in the recirculation circuit. Figure 9.7.3 below gives a simplified 3D recirculation layout of the possible ventilation arrangements as used at Loraine Gold Mines Ltd in 1984. An important issue that must be kept in mind is the total heat load of the fans, which can also have an effect on the amount of cooling required. Within the area itself, regulators or even extra fans may be necessary to direct the required quantities of mixed intake air to the individual working areas.

Burton in his paper discussed two major reasons why it may possibly be desirable to stop recirculation at the time of each blast. First, the plug of fumes that normally occurs at each blast may be passed directly to the return airway. Furthermore, the dust and corrosive fumes caused by the blast would not pass through the recirculation fans or cooling sprays. Secondly, if recirculation is continued at the blast, the entire mixed intake airway must be included in the re-entry zone (for the safety of essential personnel).

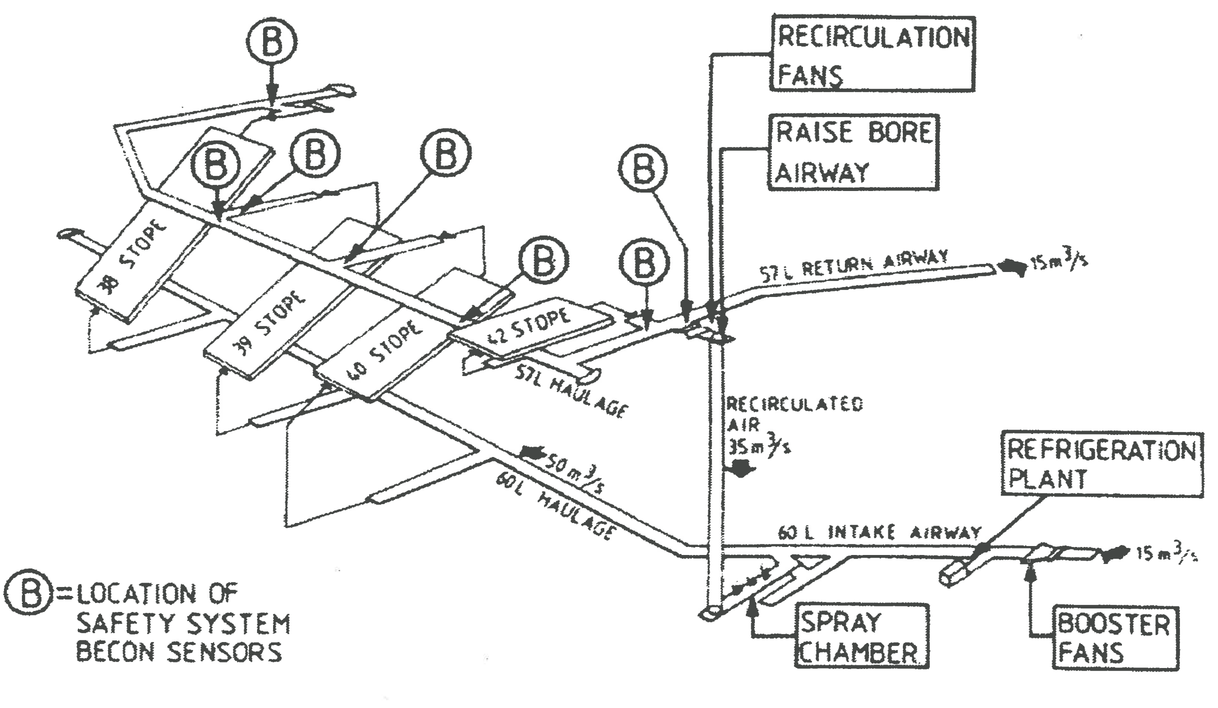

Figure 9.7.3 ***Simplified 3D drawing of a recirculation of air layout***

9.7.4. Cooling arrangements

One of the main objectives in providing a recirculation system is to allow more bulk air spray cooling to be employed, rather than the troublesome cooling coil networks currently used. If the recirculation of air stops for blasts, it would be better to locate the coolers in the mixed intake air. Burton stated that this would offer two advantages, namely the intake air would continue to be cooled and that this cooling load may be sufficient to keep the refrigeration plant running.

An important aspect to consider is the heat-rejection facilities that may be required. If the chilled water required by the bulk air cooler is provided by a refrigeration plant located elsewhere in the mine or on surface, then no special provision need be made. It is therefore quite important to keep this aspect in mind in the initial design for ultra-deep mines. If a refrigeration plant is planned in the recirculation area, the proposed quantity of intake air and hence return air may not be sufficient if the introduction of recirculation implies a reduction in return air quantity.

In the event of there being no other return airways nearby which could be used for heat rejection, the condenser water could be piped away for recooling, but this might be impractical. It is therefore important to keep this aspect in mind in planning a recirculation system for ultra-deep mines. The quantity of return air must remain sufficient for heat rejection.

9.7.5. Safety arrangements

If a fire were to start within the area served by the recirculation system, hazardous fumes would be introduced into the intake. There are automatic safety systems and they have been used successfully in the past. The objectives of such a system would be:

- To detect reliably and quickly the occurrence and, preferably, the location of a fire
- To stop the recirculation fans immediately and restore the area to a once-through ventilation system
- To raise an alarm with the responsible personnel so that normal fire procedures can be followed.

These objectives must be achieved, either by using trained personnel or by installing a fire detection system or a combination of both. It must be emphasized that controlled recirculation should not be contemplated without the use of a suitable system (Middleton *et al.*, 1985).

Another important hazard that can occur is the gradual build-up of contaminants if the intake air is interrupted. If this happens, the contaminants would build up in virtually the same way as if there had been no controlled recirculation in the first place. Controlled recirculation would merely 'spread' the contaminants within the area and would not add to them.

In a normal once-through ventilation system, all the air supplied to an area is, of course, the intake air and workers would soon be aware of it if the fans stopped. However, if the recirculation fans were to continue to run, workers might not be aware that the intake air was no longer being supplied. Although some effects would not constitute an emergency, mines normally consider interlocking the recirculation fans with the intake booster fans so that recirculation cannot take place on its own (Burton, 1984).

9.8. Operation of controlled recirculation systems

Burton suggested that the use of controlled recirculation may require some changes to standard procedures in existing mines. In the design for a new mine, the various parameters related to the recirculation of air can be included beforehand so as to eliminate unnecessary problems. The quantity of intake air, however, should be sufficient for controlling return air contaminant levels and it should be ensured that controlled recirculation does not, in general, affect such levels. Once a recirculation system is established, a careful check must be made as to whether the intake quantity is actually sufficient. This will require monitoring in the return air during on-shift periods. In addition, measurements must be made to check that blasting fumes are removed at a sufficient rate from the area. The quantity of intake air may need to be increased if contaminant levels exceed acceptable values (Burton, 1984).

9.9. Aspects relevant to controlled recirculation

Controlled recirculation of air can and will play an important role in the provision of an acceptable underground environment for ultra-deep-level mining. More specifically, the following can be stated:

- Controlled recirculation provides a means of distributing the required amount of refrigeration in underground workings. This is achieved by increasing the air flow in an area without any increase in downcast air quantity.

- Controlled recirculation does not, in general, lead to an increase in return air contaminant levels, nor does it lead to a gradual build-up of contaminants over time.

- In all recirculation systems a quantity of 'fresh' intake air must be supplied to maintain acceptable gas levels and to remove blasting fumes timeously. This air must also be sufficient to control dust levels if no dust filtration is practised.

- Controlled recirculation will offer considerable practical and financial benefits in ultra-deep gold mines and the extent of the financial benefits will be investigated in following sections. In planning new mines, many of the recirculation constraints experienced by existing mines can be eliminated so as to optimise the benefits possible.

- The most important hazard associated with the use of controlled recirculation is fire. A reliable means of identifying, as soon as possible, the occurrence and location of a fire and of stopping recirculation is an essential requirement of all recirculation systems (Burton, 1984).

9.10. Simulation models and ENVIRON modelling parameters

9.10.1. Introduction

The representativeness and prediction capability of environmental software such as ENVIRON depends heavily on the accuracy of the mining layout and other operational considerations. For the base-case simulation model, the following applied:

The mine was assumed to be in the Carletonville area and all weather data, rock and geothermal properties used were taken from the MVS Data Book. The depth of the block of ground to be mined was between 3 800 m and 5 000 m. The peak stoping production was limited to 45 000 m^2 per month. This value corresponded to a nominal tonnage of 192 954 tons per month at a stoping width of 1,5 m. The development advance rate was taken at 60 m per month, which gave an approximate additional tonnage of 45 804 tons per month. The total tonnage mined for the simulation model was 238 758 tons. The mining method that was chosen for this simulation was the longwall, follow-behind system.

In essence this mining method means that the development is always behind the mined-out stope faces so as to ensure that the development always takes place in a destressed area. This mining method was used in setting up the simulation model and the various terms used, such as lower follow-behind (LFB) and upper follow-behind (UFB) relate to the position of the development in relation to the actual stoping. The main intake air per raise line, as indicated in Figures 9.10.1a and 9.10.1b, is on the lower level through an airway leading to the stope, and the return air passes through a ventilation boxhole down to the upper follow-behind airway, back to the main return airway.

For the base-case simulation model, a fixed mass flow was used to satisfy the design criteria chosen for the model. For the base case, no recirculation of air was considered and the air mass flow rate down the shaft was 934 kg/s. This meant a planned air requirement of approximately 3,9 kg/s/kt mined. The base case was then used as a reference in setting up the global and recirculation simulation models. For both the global and localised recirculation models, controlled recirculation percentages of 20, 30, 40, 50 and 60% were assumed and these different percentages of re-circulated air were built into the base-case simulation model by using fixed mass flows for that specific area. Figures 9.10.1a and Figure 9.10.1b are simplified drawings to explain the difference between global and local recirculation as it was incorporated into the base-case model.

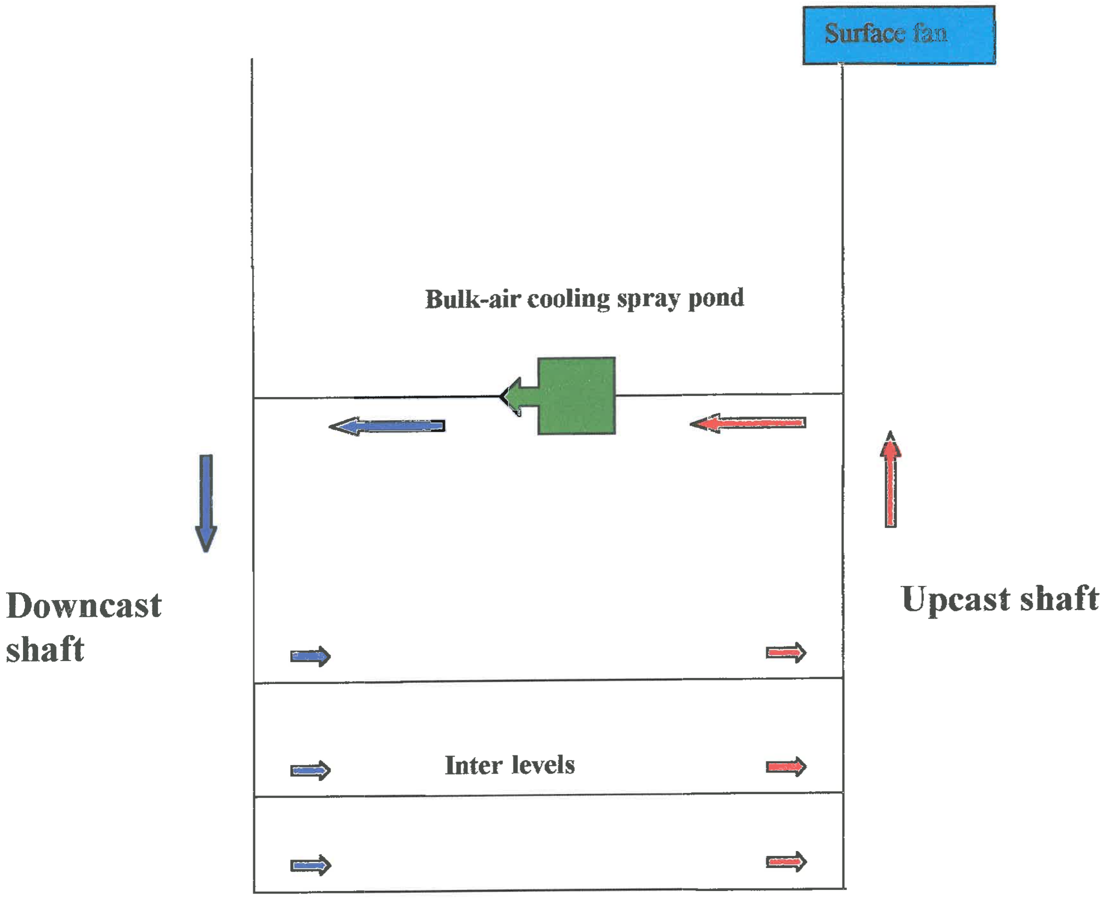

Figure 9.10.1.a ***Simplified drawing of a global recirculation layout***

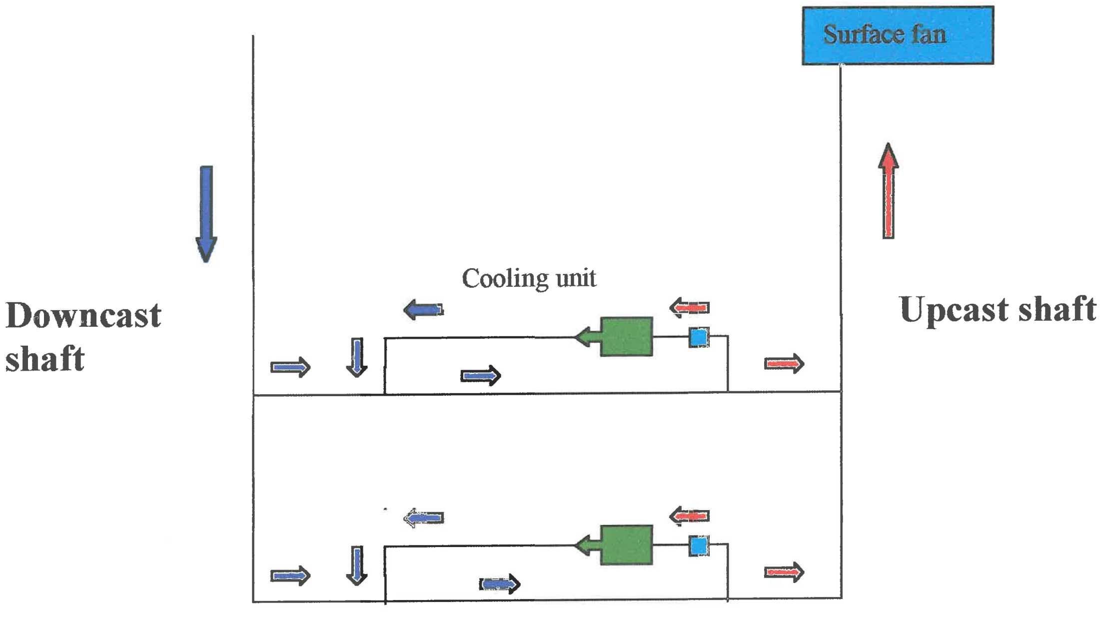

Figure 9.10.1.b ***Simplified drawing of a localised recirculation layout***

For the simulation of the global recirculation model, the amount of air to be recirculated was drawn off on level 3 500 m as a percentage of the main upcast return air mass flow and this meant a reduction in the total amount of downcast air from surface. The amount of air that was drawn off was then reintroduced to the main downcast shaft where it was mixed with the now reduced amount of fresh downcast air and was sent through the bulk air cooler, which was also situated on level 3 500 m. For the base-case scenario, the full 934 kg/s coming from surface was sent through the bulk air cooler. In the case of the localised recirculation model, only the reduced amount of downcast air was sent through the bulk air cooling system on 3 500 m. The amount of air recirculated on each level was cooled by coolers introduced for each of these recirculated circuits.

With controlled localised recirculation of return air, an amount of air equal to a set percentage of the return air was drawn off at a specific point on every level where the return air coming from a raise and incline connection rejoined. This was indicated by the longwall follow-behind mining method. This air was then cooled and reintroduced into the main intake air feeding that specific raise line and incline. This meant a reduction in the total air required per level and hence a reduction in the total air required from surface. This recirculation of return air was repeated for all levels at different recirculation percentages in order to bring about an overall reduction in the total air needed from surface.

9.10.2. Other ENVIRON modelling parameters

9.10.2.1 General parameters

The design reject wet-bulb temperature from all stopes was limited to 27°C ± 1°C. This temperature range rendered a value for specific cooling power of at least 300 W/m^2 at a velocity of 0,5 m/s as was proposed in the initial stages of this investigation. Air quantities entering the stopes were planned to ensure that face velocities remained between 0,5 and 1,0 m/s. The minimum face velocity as required by South African regulations is 0,25m/s, which would mean an over-design and well within limits. The same velocity was used for all the simulation models and it was therefore a constant factor throughout the investigation. The face zone width was assumed at 5 m (as a normal standard used for ventilation control). The average face advance was taken as 12 m/month.

Maximum design wet-bulb temperatures in all intake airways, development ends and travelling ways were limited to 29°C±1°C. Ventilation for development ends was based on a minimum requirement of 0,3 m^3/s per m^2 of face (double the amount required by South African regulations) and the minimum air quantity for intake airways was 10 m^3/s. In all instances a leakage percentage of 30% was assumed.

The main vertical downcast shaft was limited to a depth of 3 500 m. For purposes of evaluating main fan power, the shaft diameter was calculated on the assumption that the nominal air velocity would be about 11 m/s. For the purpose of comparison, the shaft diameter was not changed for the various simulation models, but it could become a major cost-saving issue needing investigation once the optimum recirculation percentage has been determined.

A sub-vertical downcast shaft was modelled from a depth of 3 500 m to 5 000 m and the diameter was based on an air velocity of about 11 m/s. Upcast shafts were sized to ensure that air velocities remained at about 20 m/s and avoided the critical velocity zone.

The temperature of the service water arriving at stopes was assumed to be 12°C for all workings up to 800 m from shaft; thereafter it was modelled to be proportionally warmer due to the additional pipe losses. The addition of linear heat in the intake airways in which chilled water pipes were installed was set to a negative value of 150 W/m (net) and for the remaining intake airways a net value of 50 W/m was used. Intake airways were not insulated and the haulages were considered to be 'wet' but without drains. It was assumed that the drain water was collected and returned via pipes from the working areas back to the return-water dams. The chilled service water consumption used was 1 ton of water per ton of rock mined. The overall fissure water inflow was 0,4 tons per ton of rock mined.

The wet-and dry-bulb temperatures of the ambient air entering the shaft system on surface were set to 17°C and 27°C respectively. The primary air cooler was situated at the top of the sub-vertical shaft at 3 500 m and cooled the mixed downcast and recirculating air to 18°C wet-bulb. Secondary air coolers were located in intake airways and travelling ways. Chilled service water was provided in all cases and in-stope air cooling when needed. The temperature of the air leaving the coolers was set at 20°C wet-bulb.

For layouts using backfill, the temperature of the backfill arriving at the stopes was assumed to be 27°C and thus, in terms of the simulation modelling, created neither a heating nor a cooling load in the stopes. In layouts where backfill was used, placement corresponded to an effective area of 70%.

The Atkinson method was used in determining pressure losses due to friction. The following values were based on past experiences and reference material for the various network components:

Downcast shafts	0,025 Ns^2/m^4
Upcast shafts	0,007 Ns^2/m^4
Intake airways	0,010 Ns^2/m^4
Upcast shafts	0,004 Ns^2/m^4

In-stope ventilation, air pressure loss was assumed to be 20 Pa per 10kg/s for a single-sided stope measured over six panels. This figure was based on past experience and on analyses of measured data collected during stope heat flow studies in similar settings.

The linear heat loads modelled in tunnels and cross-cuts were set to 50W/m. This figure was considered to be reliable and is typical of linear heat loads encountered in a deep-level mine. It was also assumed that electric locos would be predominantly used, thereby minimising the heat load. Additional heat generated by other types of locos, such as diesel, was catered for in the assumed overall average linear heat figure.

The total heat load depends to an appreciable extent on the age of the excavations. Based on anticipated mining rates, the average age used in this simulation model was four years. This figure is typical for the mining area, and thus for the rock surfaces, at various times over its life. No spot heat loads were added in these models.

9.10.2.2 Diagrammatic layout for base-case and air recirculation models

In the base-case model each level and its various subsections were subdivided into sections and were marked accordingly from A to S. This numbering was also later used in identifying the various recirculation sections in the comparison with the localised recirculation section. The important issue was therefore to find out to what extent the cooling and fan requirements would change with various changes in the amount of air being recirculated globally. The results would then be used in a costing model to determine the optimum recirculation of air (percentage). Figure 9.10.2.2 below shows a diagram of the simplified numbering of sections and the average depth for each level, as used in all the simulation models.

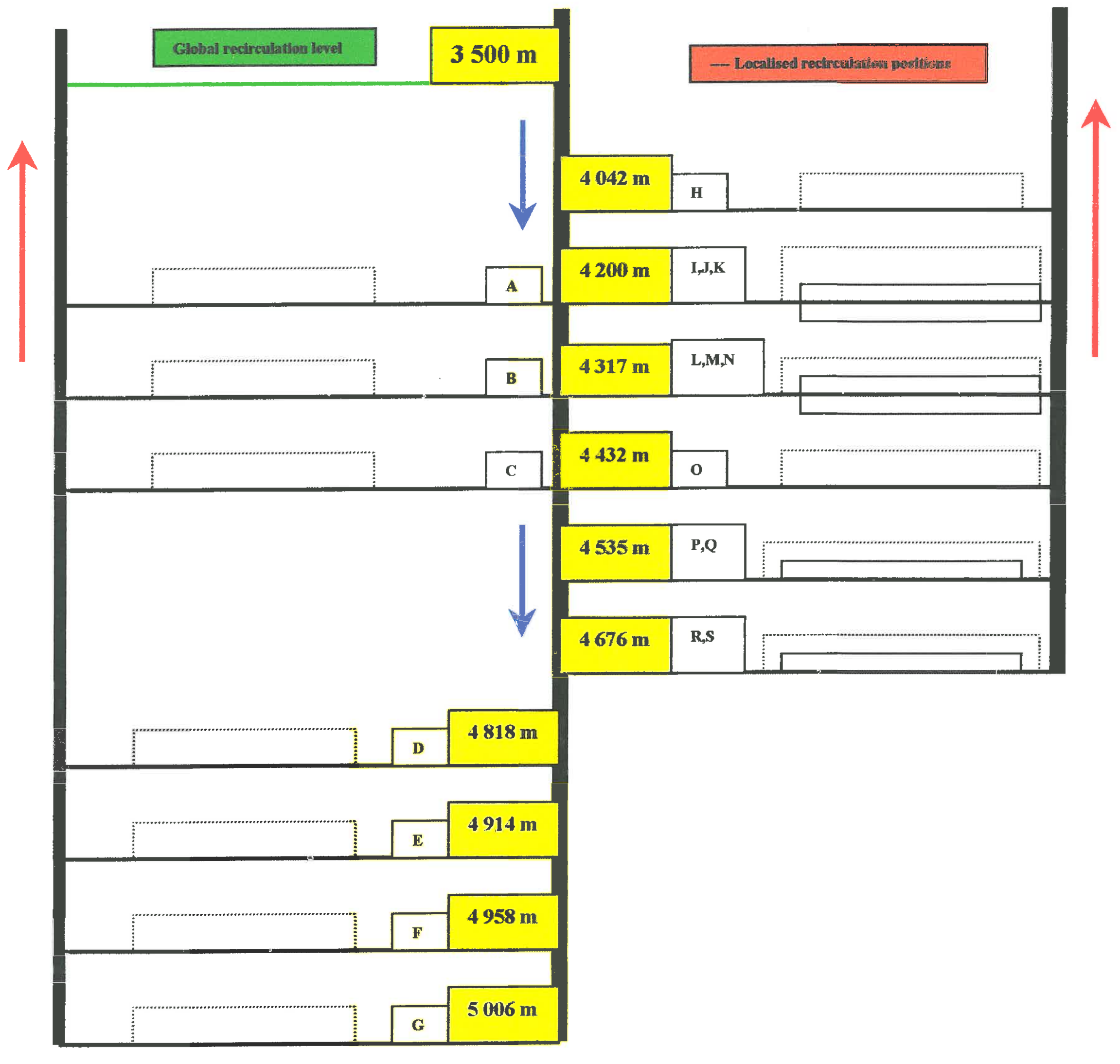

Figure 9.10.2.2 *Simplified numbering of sections and average depth*

9.10.2.3 Global features of base-case simulation model

In order to establish the base-case simulation model, a set of basic parameters as well as the assumptions mentioned before were used. Some of the other features of the base-case simulation model are listed in Table 9.10.2.3 below.

Table 9.10.2.3
Basic features of simulation model

Number of regulators	55
Number of fixed branches	1 for base case, 2 for the global recirculation model and 20 for the localised recirculation model
Number of fans (additional)	nil (fixed flow used as reference)
Number of hoists	1
Number of air coolers employed	74
Number of branches	456
Number of nodes	399
Number of working levels	10 (excluding bulk air cooling level)
Summary of tunnel results:	
Number of tunnels	193 (total length 98 021 m)
Summary of air cooler results:	
Number of air coolers employed	74
Number in operation	70
Summary of development results:	
Number of development ends	45
Summary of shaft results:	
Number of shafts	21 (including inclined shafts, and shaft split up into sections taken as additional shafts)
Summary of shaft station area results:	
Number of shaft station areas	11
Length of tunnels	11 km
Summary of stope results:	
Total number of stopes	54 (single and double-sided stopes)

10. Comparison of simulation model results

10.1. Introduction

The results achieved from the base-case simulation model (0% recirculated) were compared those obtained from the global and localised recirculation of return air models. These results were for various percentages of air recirculated, as specified before. In the comparison of all the results obtained, all aspects related to the costs for cooling and fan requirements were considered and evaluated. All the assumptions and calculations used for all comparisons will be discussed in detail.

10.2. Base-case simulation and global recirculation results

From the various global recirculation runs that were done on ENVIRON, a summary of each percentage global recirculation simulation was drawn up and compared with the results obtained from the base-case simulation. The results of the comparison of the physical parameters for the base-case and global recirculation models are summarised in Table 10.2 below.

Table 10.2
Summary of physical parameters for base-case and global recirculation models

	Base case	20%	30%	40%	50%	60%
Summary of global parameters:						
Number of shafts	21	22	22	22	22	22
Number of regulators	55	55	55	55	55	55
Number of fixed branches	1	2	2	2	2	2
Number of fans	0	0	0	0	0	0
Number of hoists	1	1	1	1	1	1
Number of air coolers	74	74	74	74	74	74
Number of dev. ends	45	45	45	45	45	45
Number of shaft stations	11	11	11	11	11	11
Number of stopes	54	54	54	54	54	54
Number of spot heat sources	0	0	0	0	0	0
Number of tunnels	193	193	193	193	193	193
Number of branches	456	457	457	457	457	457
Number of nodes	399	399	399	399	399	399
Summary of tunnel results:						
Number of tunnels	193	193	193	193	193	193
Total length (m)	98 021	98 021	98 021	98 021	98 021	98 021

Table 10.2 continued
Summary of physical parameters for base-case and global recirculation models

Summary of air cooler results:						
Total number of air coolers	74	74	74	74	74	74
Number in operation	70	70	70	70	70	70
Summary of development end results:						
Total number of development ends	45	45	45	45	45	45
Total production per month(tons/month)	45 804	45 804	45 804	45 804	45 804	45 804
Total service water used (tons/month)	32 063	32 063	32 063	32 063	32 063	32 063
Total fissure water (tons/month)	18	18	18	18	18	18
Summary of hoist results:						
Total number of hoists	1	1	1	1	1	1
Summary of shaft results:						
Total number of shafts	21	22	22	22	22	22
Total length (m)	10 012	10 012	10 012	10 012	10 012	10 012
Summary of shaft station area results:						
Total number of shaft station areas	11	11	11	11	11	11
Total length of tunnels (m)	11 000	11 000	11 000	11 000	11 000	11 000
Summary of stope results:						
Total number of stopes	54	54	54	54	54	54
Total production per month (kt/month)	193	193	193	193	193	193
Total service water used (kt/month) 193	193	193	193	193	193	
Total fissure water (t/month)	77 181	77 181	77 181	77 181	77 181	77 181
Summary of service water:						
Service water used in stopes (kt/month)	193	193	193	193	193	193
Service water dev. ends (kt/month)	32	32	32	32	32	32
Total service water used (kt/month)	225	225	225	225	225	225
Summary of production:						
Production stopes (kt/month)	193	193	193	193	193	193
Production dev. ends (kt/month)	45,8	45,8	45,8	45,8	45,8	45,8
Total production (kt/month)	239	239	239	239	239	239

From the table above it is obvious that the base-case model was kept almost the same for the global recirculation of air models, except for the fact that a recirculation circuit was introduced on level 3 500 m. This was done by introducing a fixed mass flow branch between the upcast and downcast shafts, as mentioned before. In terms of mining-related figures, all basic figures stayed the same so that a real comparison of results could be made eventually. The number of coolers in operation stayed the same, because all the air recirculated on the bulk air cooling level at 3 500 m underground went through the bulk air cooler as in the base case. The configuration of coolers and air-flow quantities for the lower levels therefore stayed the same.

10.2.1. Cooling and pumping requirements

In order to establish the cooling and pumping requirements, it was important to summarise the results achieved per level. By using the abovementioned method of subdividing the various levels, it was possible to summarise the cooling and pumping requirements for each level when global recirculation was considered. Table 10.2.1a gives a comparative summary of the results achieved for the cooling requirements for the base case and the percentages of air being recirculated globally.

Table 10.2.1a
Summary of cooling requirements for the base-case model and the models with various percentages for global recirculation of return air

Item description	Average depth	Percentage of air re-circulated globally					
		0%	20%	30%	40%	50%	60%
Bulk air cooling requirement (kW)	3 500	33 868	34 222	34 245	34 035	33 793	33 482
Cooling requirement per level (kW)	4 042	1 982	1 991	1 983	1 984	1 984	1 984
	4 200	6 482	6 526	6 511	6 519	6 526	6 531
	4 317	7 126	77 140	7 133	7 143	7 152	7 158
	4 432	5 438	5 404	5 147	5 387	5 384	5 381
	4 535	6 141	6 144	6 139	6 136	6 134	6 132
	4 676	3 650	3 698	3 696	3 701	3 703	3 705
	4 818	4 113	4 161	4 158	4 156	4 155	4 154
	4 914	4 638	4 689	4 681	4 679	4 677	4 675
	4 958	4 482	4 532	4 526	4 523	4 520	4 519
	5 006	3 861	3 920	3 912	3 909	3 908	3 906
Total cooling required (kW)		**81 781**	**82 428**	**82 130**	**82 171**	**81 935**	**81 628**

The table above is a summary of the results obtained from the detailed analysis for each level and each recirculation of return air (percentage).

In essence, the amount of bulk air cooling and cooling on the different levels basically stayed the same, as was expected. This is because of the fact that the amount of air flow down the sub-vertical shaft was kept constant. The only difference was that the hot return air from the upcast shaft was reintroduced into the main downcast air and cooled. This had the effect of slightly decreasing the amount of cooling required for each of the global recirculation of air models considered. This can be explained by the fact that the auto-decompression of air coming up from 5 006 m to 3 500 m below surface played a more significant role than the increase in temperature of the air coming from surface originally. Table 10.2.1b below gives a summary of the physical results for the comparison of the base-case simulation model and the various global recirculation percentages of return air.

Table 10.2.1b
Summary of total cooling requirements for global recirculation

	Base case	20%	30%	40%	50%	60%
Tunnels:						
Total heat load (kW)	41 467	41 309	41 166	41 141	41 088	41 043
Total heat from linear sources (kW)	-9 518	-9 518	-9 518	-9 518	-9 518	-9 518
Total heat load per metre (W/m)	423	421	420	420	419	419
Summary of air cooler results:						
Total cooling duty (reg. air coolers)(kW)	78 543	79 188	78 883	78 926	78 689	78 381
Summary of development end results:						
Total artificial heat load (kW)	707	707	707	707	707	707
Total heat load from service water (kW)	-596	-598	-604	-600	-601	-601
Total air heat load (kW)	5 100	5 074	5 036	5 053	5 046	5 040
Total heat load (kW)	5 695	5 672	5 640	5 653	5 646	5 641
Heat load/ton/month (W/ton/month)	124	124	123	123	123	123
Summary of hoist results:						
Total heat load (kW)	1 000	1 000	1 000	1 000	1 000	1 000
Summary of shaft results:						
Total heat from linear sources(kW)	-720	-720	-720	-720	-720	-720
Total heat load (kW)	1 761	1 769	1 768	1 777	1 778	1 777
Heat flow per metre (W/m)	176	177	177	177	178	178
Summary of shaft station area results:						
Total machine heat load (kW)	600	600	600	600	600	600
Total heat load (kW)	6 125	6 111	6 102	6 095	6 090	6 085

Table 10.2.1b continued
Summary of total cooling requirements and heat loads for global re-circulation

Summary of stope results:						
Total number of stopes	54	54	54	54	54	54
Total artificial heat (kW)	1 861	1 861	1 861	1 861	1 861	1 861
Total heat from service water (kW)	-2 642	-2 642	-2 644	-2 645	-2 645	-2 646
Total heat from fissure water (kW)	5 531	5 531	5 530	5 529	5 529	5 529
Total air heat load (kW)	22 026	22 006	21 987	21 980	21 974	21 969
Total heat load (kW)	24 667	24 648	24 631	24 624	24 619	24 615
Total heat load (W/t/month)	128	128	128	128	128	128
Summary of service water:						
Total heat load from service water (kW)	-3 238	-3 240	-3 248	-3 245	-3 246	-3 247
Summary of heat loads and cooling requirements:						
Total heat load (kW)	80 716	80 510	80 308	80 290	80 221	80 162
Heat load per ton (W/t/month)	338	337	336	336	336	336
Cooling: Service water (kW)	3 238	3 240	3 248	3 245	3 246	3 247
Cooling: Regular air coolers (kW)	78 543	79 188	78 883	78 926	78 689	78 381
Total cooling (kW)	81 781	82 428	82 130	82 171	81 935	81 628
Air-flow quantities:						
Downcast air main shaft (kg/s)	934	746	654	560	467	374
Total amount of air recirculated (kg/s)	0	188	280	268	374	560
Fan input power requirements:						
Surface main fans (kW)	18 482	8 140	5 865	4 107	2 771	1 789
Total recirculation fans underground (kW)	0	679	1 009	1 336	1 660	1 982
Total fan input power requirements (kW)	18 482	8 819	6 874	5 443	4 432	3 771

The table above also shows a comparison of the total amount of air being recirculated and how the amount of downcast air is reduced from 934 kg/s to 374 kg/s. The significant change in the fan input power requirements is also worth noting. The total reduction in input power required was due mainly to a substantial decrease in the pressure drop across the fixed flow branches from surface to underground. This gave further support to the opinion expressed earlier to the effect that the recirculation of air would become imperative in the design of the air-flow requirements for an ultra-deep mine. There was no great change in the amount of cooling required, as indicated, but a large difference in the input power required for fans. This immediately indicated that it would be more beneficial to make use of the recirculation of air.

10.2.2. Requirements for fans on surface and underground

In considering the requirements for the fans on surface and underground, the normal basic fan strategy of centrifugal fans was adopted. The same fans were assumed for all the simulation models and at no stage was the optimisation of any fans considered. For the base-case simulation model, 934 kg/s was used as the total amount of downcast air for comparative purposes. The amounts of air recirculated on level 3 500 m below surface were derived as a percentage of this total amount. As mentioned before, this amount of air was drawn off from the main upcast shaft and reintroduced into the main downcast shaft. After it had mixed with the total lower amount of downcast air, the full 934 kg/s was then bulk air-cooled. One would expect the required fan input power of the main fans on surface to decrease and the fan input power of the fans used in the global recirculation of return air to increase. Table 10.2.2 below gives a summary of the results obtained for the base-case and global simulation models.

Table 10.2.2
Summary of total air-flow and fan requirements for global recirculation

Item description	Average depth	Percentage of air recirculated globally 0%	20%	30%	40%	50%	60%
Air mass flow - downcast shaft		934	746	654	560	467	374
Mass flow of air recirculated	3 500	0	188	280	268	374	560
Main fans input power (kW)	0	18 482	8 140	5 865	4 107	2 771	1 789
Recirculation fans input power (kW)	3 500	0	679	1 009	1 336	1 660	1 982
Total input power surface and U/G (kW)		18 482	8 819	6 874	5 443	4 432	3 771

*Fan efficiency assumed as 76%.

It can be seen from the results obtained that there was a substantial reduction in the input power requirements for the base case compared with the recirculation models. This was expected due to the large decrease in the total volume and overall pressure required by the main fans on surface. There was a slight increase in the input power requirements for the recirculation fans as the amount of air recirculated was increased, but it was not as dramatic as the decrease in the input requirements for the surface main fans. The input power for the fans on surface was reduced by almost half for only 20% recirculation of air. For the purpose of this study an efficiency figure of 76% was assumed for all fans. However, the efficiency figure used here is a practical figure used for fans.

10.3. Base-case simulation and localised re-circulation results

From the various recirculation runs that were done on ENVIRON, a summary of each percentage localised recirculation simulation was obtained and compared with the results obtained from the base-case simulation. The results of the base-case and localised recirculation models and various other physical, related parameters are summarised and compared in Table 10.3 below.

Table 10.3
Summary of physical parameters for the base-case and localised recirculation models

	Base case	20%	30%	40%	50%	60%
Summary of global parameters:						
Number of shafts	21	21	21	21	21	21
Number of regulators	55	55	55	55	55	55
Number of fixed branches	1	20	20	20	20	20
Number of hoists	1	1	1	1	1	1
Number of air coolers	74	74	74	74	74	74
Number of dev. ends	45	45	45	45	45	45
Number of shaft stations	11	11	11	11	11	11
Number of stopes	54	54	54	54	54	54
Number of spot heat sources	0	0	0	0	0	0
Number of tunnels	194	194	194	194	194	194
Number of branches	456	481	481	481	481	481
Number of nodes	399	404	404	404	404	404
Summary of tunnel results:						
Number of tunnels	194	194	194	194	194	194
Total length (m)	98 021	98 021	98 021	98 021	98 021	98 021
Summary of air cooler results:						
Total number of air coolers	74	80	80	80	80	80
Number in operation	70	76	76	76	76	76
Summary of development end results:						
Total number of development ends	45	45	45	45	45	45
Total production per month (t/month)	45 804	45 804	45 804	45 804	45 804	45 804
Total service water used (t/month)	32 063	32 063	32 063	32 063	32 063	32 063
Total fissure water (t/month)	18	18	18	18	18	18
Summary of hoist results:						
Total number of hoists	1	1	1	1	1	1

Table 10.3 continued
Summary of physical parameters for base-case and localised recirculation models

Summary of shaft results:						
Total number of shafts	21	21	21	21	21	21
Total length (m)	10 012	10 012	10 012	10 012	10 012	10 012
Summary of shaft station area results:						
Total number of shaft station areas	11	11	11	11	11	11
Total length of tunnels (m)	11 000	11 000	11 000	11 000	11 000	11 000
Summary of stope results:						
Total number of stopes	54	54	54	54	54	54
Total production per month (kt/month)	193	193	193	193	193	193
Total service water used (kt/month)	193	193	193	193	193	193
Total fissure water (t/month)	77 181	77 181	77 181	77 181	77 181	77 181
Summary of service water:						
Service water used in stopes (kt/month)	193	193	193	193	193	193
Service water dev. ends (kt/month)	32	32	32	32	32	32
Total service water used (kt/month)	225	225	225	225	225	225
Summary of production:						
Production stopes (kt/month)	193	193	193	193	193	193
Production dev. ends (kt/month)	45,8	45,8	45,8	45,8	45,8	45,8
Total production (kt/month)	239	239	239	239	239	239

The table above also shows that the base-case design was not altered a lot. The only real difference was that 19 additional fixed flow branches were included on the ten working levels. This was done in order to simulate the total recirculation of return air per level. The strategy followed was that a percentage of the air coming from the boxhole airway connected to a stope and incline line on a specific level was recirculated. This process was repeated for every level and its effect was that the total amount of air that entered a specific level was now lower by the amount being recirculated. This led to a large reduction in the air in the downcast shaft. All other parameters were kept the same as in the base case.

10.3.1. Cooling and pumping requirements

As was the case with the global recirculation models, it was also necessary to summarise the results achieved per level for cooling and pumping requirements. By again using the method of subdividing the various levels as mentioned before, it was possible to summarise the cooling requirements for each level when localised recirculation on each level was considered. It was also obvious that the total amount of air in the sub-vertical shaft would now be less by the sum of the total amount of air that was recirculated on each level. In addition, the total amount of cooling required per level would increase due to the fact that the air being recirculated on each level had to be cooled. It was also one of the objectives to keep the total amount of air in each of the raise lines and inclines as close as possible to the base-case figures mentioned before. One would expect the fan input power of the main fans on surface to be reduced and the fan input power of the fans for localised recirculation of return air on each level to increase. Table 10.3.1a gives a summary of all the relevant cooling requirements for the base case, as well as the cooling requirements for the localised recirculation models using various percentages of return air.

Table 10.3.1a
Summary of cooling requirements for the base-case model and the models with various percentages of localised re-circulation of return air

Item description	Average depth	Percentage of air recirculated locally					
		0%	20%	30%	40%	50%	60%
Bulk air cooling requirement (kW)	3 500	33 868	29 606	27 368	25 070	22 757	20 365
Cooling requirement per level (kW)	4 042	1 982	3 000	3 124	3 248	3 367	3 474
	4 200	6 482	8 138	8 554	8 962	9 340	9 634
	4 317	7 126	8 298	8 568	8 813	9 023	9 201
	4 432	5 438	5 730	5 771	5 805	5 831	5 848
	4 535	6 141	6 360	6 457	6 544	6 623	6 693
	4 676	3 650	4 092	4 193	4 286	4 371	4 449
	4 818	4 113	4 241	4 217	4 195	4 174	4 153
	4 914	4 638	4 759	4 754	4 745	4 734	4 720
	4 958	4 482	4 598	4 594	4 585	4 576	4 564
	5 006	3 861	4 008	3 999	3 990	3 979	3 966
Total cooling required (kW)		81 781	82 830	81 599	80 244	78 775	77 067

The above table is a summary of the results obtained from the detailed analysis for each level and each percentage recirculation of return air.

A summary of the total cooling requirement results for localised recirculation is shown in Table 10.3.1b below.

Table 10.3.1b

Summary of total cooling requirements and heat loads for localised recirculation

	Base case	20%	30%	40%	50%	60%
Tunnels:						
Total heat load (kW)	41 467	43 497	43 118	42 654	42 079	41 294
Total heat from linear sources (kW)	-9 518	-9 518	-9 518	-9 518	-9 518	-9 518
Total heat load per metre (W/m)	423	443	439	434	428	420
Summary of air cooler results:						
Total cooling duty (reg. air coolers)(kW)	78 543	79 494	78 262	76 906	75 436	73 724
Summary of development end results:						
Total artificial heat load (kW)	707	707	707	707	707	707
Total heat load from service water (kW)	-596	-618	-619	-620	-621	-624
Total air heat load (kW)	5 100	4 985	4 975	4 965	4 954	4 939
Total heat load (kW)	5 695	5 603	5 594	5 584	5 575	5 562
Heat load/ton/month (W/t/month)	124	122	122	122	122	121
Summary of hoist results:						
Total heat load (kW)	1 000	1 000	1 000	1 000	1 000	1 000
Summary of shaft results:						
Total heat from linear sources(kW)	-720	-720	-720	-720	-720	-720
Total heat load (kW)	1 761	1 734	1 700	1 662	1 620	1 566
Heat flow per metre (W/m)	176	173	170	166	162	156
Summary of shaft station area results:						
Total machine heat load (kW)	600	600	600	600	600	600
Total heat load (kW)	6 125	6 049	6 004	5 956	5 903	5 847
Summary of stope results:						
Total number of stopes	54	54	54	54	54	54
Total artificial heat (kW)	1 861	1 861	1 861	1 861	1 861	1 861
Total heat from service water (kW)	-2 642	-2 718	-2 718	-2 718	-2 718	-2 720
Total heat from fissure water (kW)	5 531	5 500	5 500	5 500	5 500	5 500
Total air heat load (kW)	22 026	21 755	21 754	21 763	21 769	21 764
Total heat load (kW)	24 667	24 474	24 473	24 481	24 487	24 484
Total heat load (W/t/month)	128	127	127	127	127	127

Table 10.3.1b continued
Summary of total cooling requirements for localised recirculation

Summary of service water:						
Total heat load from service water (kW)	-3 238	-3 336	-3 337	-3 338	-3 339	-3 343
Summary of heat loads and cooling requirements:						
Total heat load (kW)	80 716	82 356	81 888	81 338	80 664	79 753
Heat load per ton (W/t/month)	338	345	3343	341	338	334
Cooling: Service water (kW)	3 238	3 336	3 337	3 338	3 339	3 343
Cooling: Regular air coolers (kW)	78 543	79 494	78 262	76 906	75 436	73 724
Total cooling (kW)	81 781	82 830	81 599	80 244	78 775	77 067

From these results it was also obvious that the base-case figures did not differ a lot from the results obtained for the various percentages of localised re-circulation of air. The most notable change was in the total amount of cooling required which decreased as the percentage recirculation of air on each level was increased. This could be explained by the fact that the bulk air cooling that was required on level 3 500 m below was much lower because the total amount of air that needed to be cooled was reduced according to the amount of air recirculated on the various levels. The detailed results showed that there was an increase in the amount of cooling needed per level. The temperature for air coming out of the coolers was set at 20°C saturated for all the coolers and no optimisation of cooler duty on the various levels was done. Since the nature of the study was comparative, this would not have changed the significance of the results obtained. In terms of the total cooling required, this will have to be considered in optimising the total cooling cost.

10.3.2. Requirements for fans on surface and underground

In considering the requirements for the fans on surface and underground for the localised re-circulation option, the normal basic fan strategy of centrifugal fans was adopted. As mentioned before, the same fans were assumed for all the simulation models and at no stage was the optimisation of any fans considered. In the case of the base-case simulation model, 934 kg/s was used as the total amount of downcast air for comparative purposes. No air was recirculated on level 3 500 m below surface anymore. The total amount of air sent through the bulk air cooler on level 3 500 m below surface decreased by the amount of the total sum of air recirculated per level. The reduced amount from surface was therefore the only air passing through the bulk air cooler. Additional coolers installed on each level cooled the recirculated air before it was reintroduced into the main stream on a level. One would expect the fan input power of the main fans on surface to decrease and fan input power of the fans recirculating the air on the various levels where recirculation of air took place to increase. Table 10.3.2 below gives a summary of the results obtained for the base-case and localised recirculation simulation models.

Table 10.3.2

Summary of total air-flow and fan requirements for localised recirculation of air

Item description	**Average depth**	**Percentage of air recirculated locally** 0%	20%	30%	40%	50%	60%
Air mass flow		934	800	733	666	600	533
Mass flow of air recirculated	3 500	0	134	201	268	334	401
Main fans input power (kW)	0	18 482	10 440	7 563	5 325	3 598	2 249
Recirculation fans input power (kW)	4 042		1,4	1,6	1,7	1,6	1,3
	4 200		9,8	13,6	1,7	1,2	20,9
	4 317		9,0	14,0	19,4	25,1	31,1
	4 432		2,6	3,9	5,2	6,4	7,6
	4 535		3,0	4,5	6,0	7,6	9,1
	4 676		2,3	3,6	4,8	6,1	7,5
	4 818		1,1	1,6	2,1	2,5	2,9
	4 914		1,1	1,7	2,3	2,9	3,6
	4 958		1,1	1,8	2,4	3,0	3,8
	5 006		1,0	1,5	2,0	2,6	3,2
Total recirculation input power (kW)		0	32,4	47,7	62,6	77,2	91,1
Total input power surface & U/G (kW)		18 482	10 472	7 611	5 388	3 675	2 340

*Fan efficiency assumed as 76%.

10.4. Costing for base-case simulation model

10.4.1. Physical cooling and pumping parameters

To be able to set up a costing model for the cooling and pumping requirements, a basic strategy for cooling and pumping was applied. From this basic strategy all the various input parameters were obtained and considered in the actual costing applicable to an ultra-deep mine. The amount of cooling required from level to level was determined for each recirculation of air model. The amount of cooling also differed from one simulation model to another. This had an effect on the water flow requirements on each level and hence on the pumping requirements for each simulation model.

The physical parameters relating to the cooling and pumping required for the base-case simulation, as mentioned before, are specified in Tables 10.4.1c, d, e, f, g and h below. These tables show all the relevant parameters associated with the cooling of air and water underground, as well as the arrangements for pumping the water. They also give an indication of which parameters were included and which excluded for the purpose of setting up a base-case cooling and pumping model. The parameters specified below were used in the actual costing exercise that was done for all the simulation models. The program used was an in-house cooling simulation model developed by CSIR Miningtek and used in the Deepmine research projects.

Table 10.4.1c
Physical parameters for cooling and pumping

STRATEGY INPUT						
Cooling generating systems:						
Surface refrigeration plants	Yes					
No. of parallel plants	1					
Availability	90%					
Chilled water temperature	3,0°C					
Intermediate temperature	10,0°C					
Total chilled water flow	750,0 kg/s					
Chiller configuration	Single units					
Condenser configuration	Series					
Cooling towers:	**Pre-cooling**		**Condenser**			
Cooling towers exist	Yes		Yes			
No. of parallel plants	1		1			
Availability (%)	80		80			
Approach temperature (°C)	3.0		5,0			
Dams:	**Chilled water**	**Clean return**		**Dirty return**		
Dams exist	Yes	Yes		Yes		
No. of dams	1	1		1		
Available (%)	90	90		90		
Storage time (min)	480	480		480		
Pumps:	**ChWD**	**CCT**	**PCT**	**CRW**	**DRW**	**WTP**
Pumps exist	Yes	Yes	Yes	Yes	Yes	No
No. of units	6	6	6	6	6	
Availability (%)	70	70	70	70	70	

Table 10.4.1c continued
Physical parameters for cooling and pumping

Surface ice plants	No	
U/G refrigeration plants (Cooling towers)	Yes	
U/G refrigeration plants (Return water)	No	
Energy-recovery systems:		
Pelton turbines	Yes	
Hydrolift devices	No	
Surface options:		
Pre-cooling towers	Yes	
Water treatment plants	No	
Chilled water dam	Yes	
Clean return water dam	Yes	
Dirty return water dam	Yes	
Level options:	**Turbine 1**	**Turbine2**
Depth (m)	1 750	3 500
Pelton turbines	Yes	Yes
Hydrolift devices	No	No
Throttle valve (OC)	No	No
Throttle valve (CC)	No	No
Refrig. plant (CT)	No	Yes
Refrig. plant (RW)	No	No
Chilled water dam	Yes	Yes
Ice mixing dam	No	No
Cool clean RW dam	Yes	Yes
Warm clean RW dam	Yes	Yes
Hot clean RW dam	No	No
Dirty RW dam	Yes	Yes
Settling dam	Yes	Yes
Water treatment plant	No	No

Several other inputs formed part of the simulation model for the pumping and cooling exercise. A very important aspect of the simulation in terms of cost-saving was the inclusion of a set of turbines on level 1 750 m below surface as well as on 3 500 m below surface. This was introduced as a basic strategy for all the simulations considered. The input figures relating to the turbines on 1 750 m and 3 500 m below surface are shown in Table 10.4.1d below.

Table 10.4.1d
Additional input figures for turbine on level 1 750 m underground

Pressure reduction:	**Pel. T**	**Hydro.**	**ThV(O)**	**ThV(C)**
Exist	Yes	No	No	No
No. of units	1			
Availability (%)	100			
Efficiency (%)	70			

Dams:	**ChW**	**Ice**	**CRWc**	**CRWw**	**CRWh**	**DRW**	**Sett**
Dams exist	Yes	No	Yes	Yes	No	Yes	Yes
No. of dams	1		1	1		1	1
Avail (%)	100		100	100		100	100
St. time(min)	480		480	0		0	0
L-to-W ratio	30		30	30		30	10

Pumps:	**HPCRW**	**LPCRW**	**HPDRW**	**LPDRW**	**UGCT**
Pumps exist	Yes	Yes	Yes	Yes	No
No. of units	6	6	6	6	
Availability (%)	100	100	100	100	
Efficiency (%)	70	70	70	70	

Pipes:	**HPChWo**	**HPChWc**	**LPIce**	**HPCRW**	**HPDRW**
Pipes exist	Yes	No	No	Yes	Yes
No. of units	2			1	1
Diameter (mm)	400			400	400

The input figures for the turbine on level 3 500 m below surface differed slightly from those used on 1 750 m and the differences are listed in Table 10.4.1e below. Figures for the dams, pipes and pumps were the same as those used on level 1 750 m. The amount of fissure water was taken as a constant figure for all levels and is therefore excluded from the figures for all simulation results.

Table 10.4.1e
Additional input figures for turbine on level 3 500 m underground

Cooling load					
Load (kW)	33 868				
To cooling load:					
Chilled water (from dam, l/s)	604,8				
Chilled water (from HP, l/s)	0,0				
Out of cooling load:					
Clean return water (cc, l/s)	0,0				
Clean return water (oc, l/s)	604,8				
Service return water (l/s)	0,0				
Water balance input:					
Water added to CC system (l/s)	983,3				
Ice added to mixing dam (kg/s)	0,0				
Estimated temp. RW to ice dam (°C)	30,0				
Pressure reduction:	**Pel. T**	**Hydro.**	**ThV(O)**	**ThV(C)**
Exist	Yes	No	No	No
No. of units	1			
Availability (%)	100			
Efficiency (%)	70			
Refrigeration plant (CT):				
No. of parallel plants	1			
Availability (%)	90			
Chilled water temperature (°C)	3,0			
Intermediate temperature (°C)	10,0			
Total chilled water flow	838,1 kg/s			
Chiller configuration	Single units			
Condenser configuration	Series			
Condenser cooling towers:				
No. of parallel plants	1			
Availability (%)	100			
WBT of return air (°C)	31,0			
Total return air (kg/s)	934,0			

The average depths for the levels at which the cooling and pumping equipment was situated, other than those already specified, were: 4 042, 4 200, 4 317, 4 432, 4 535, 4 676, 4 818, 4 914, 4 958, and 5 006 m. The pumping and cooling parameters used for all these levels are listed in Table 10.4.1f below.

Table 10.4.1.f
Input for pumping and cooling parameters for all other levels

Pelton turbines	No
Hydrolift devices	No
Throttle valve (OC)	Yes
Throttle valve (CC)	Yes
Refrig. plant (CT)	No
Refrig. plant (RW)	No
Chilled water dam	Yes
Ice mixing dam	No
Cool clean RW dam	Yes
Warm clean RW dam	Yes
Hot clean RW dam	No
Dirty RW dam	Yes
Settling dam	Yes
Water treatment plant	No

In the base-case simulation model, as well as for the other simulations, the cooling load and hence the water flow rates associated with the cooling varied from level to level. These are listed in Table 10.4.1g below

Table 10.4.1g
Changing cooling loads and water flow rates for each level

Cooling loads:				
			To cooling load:	**Out of cooling load:**
	Average depth	**Load**	**Chilled water from HP (l/s)**	**Clean water return (cc, l/s)**
Levels	4 042	1 982	33,4	33,4
	4 200	6 482	121,6	121,6
	4 317	7 126	138,3	138,3
	4 432	5 438	108,1	108,1
	4 535	6 141	125,1	125,1
	4 676	3 650	76,0	76,0
	4 818	4 113	88,3	88,3
	4 914	4 638	102,7	102,7
	4 958	4 482	101,4	101,4
	5 006	3 861	88,3	88,3

Input parameters relating to dams, pumps and piping formed an integral part of the simulation program and the figures and parameters used are listed in Table 10.4.1h below.

Table 10.4.1h

Constant input parameter figures for dams, pumps and pipes for all levels

Pressure reduction	**Pel. T**	**Hydro.**	**ThV(O)**	**ThV(C)**
Exist	Yes	No	No	No
No. of units	1			
Availability (%)	100			
Efficiency (%)	70			

Dams:	**ChW**	**Ice**	**CRWc**	**CRWw**	**CRWh**	**DRW**	**Sett**
Dams exist	Yes	No	Yes	Yes	No	Yes	Yes
No. of dams	1		1	1		1	1
Avail (%)	100		100	100		100	100
St. time(min)	480		480	0		0	0
L-to-W ratio	30		30	30		30	10

Pumps:	**HPCRW**	**LPCRW**	**HPDRW**	**LPDRW**	**UGCT**
Pumps exist	Yes	Yes	Yes	Yes	No
No. of units	6	6	6	6	
Availability (%)	100	100	100	100	
Efficiency (%)	70	70	70	70	

Pipes:	**HPChWo**	**HPChWc**	**LPIce**	**HPCRW**	**HPDRW**
Pipes exist	Yes	No	No	Yes	Yes
No. of units	2			1	1
Diameter (mm)	400			400	400

Water temperatures:	
Temperature of service return water from working areas	28,0°C
Temperature of clean return water from coolers	18,0 °C
Quality of U/G insulation:	New

10.4.2. Unit costs for pumping and cooling requirements

In determining the total cost associated with cooling and pumping, certain unit costs had to be considered and used. In this section all the basic costs associated with cooling and pumping have been incorporated into all the simulation models. A summary of all the various constant figures used in the cost calculations for the base-case and global recirculation models is shown in Table 10.4.2.

Table 10.4.2
Unit cost figures as used in the Deepmine 6.4.1 cooling cost model

Basic cooling unit costs				
Cooling generating systems:				
Capital cost (R/l/s)		330 000		
Maintenance cost (R/l/s)		10 600		
Dams (surface):		**Chilled water**	**Clean return**	**Dirty return**
Capital cost (R/m^3)		500	500	500
Maintenance cost (R/R/m^3)		10	10	10
Energy-recovery systems:				
Capital cost (R/l/s)		18 000		
Maintenance cost (R/l/s)		2 000		
Pressure reduction:				
Capital cost (R/l/s)		18 000		
Maintenance cost (R/l/s)		2 000		
Dams (underground):		**CRWc**		
Capital cost (R/m^3)		725		
Maintenance cost (R/m^3)		80		
Pumps (U/G):	**HPCRW**	**LPCRW**	**HPDRW**	**LPDRW**
Capital cost (R/kW)	5 000	700	5000	700
Maint. cost (R/kW)	300	80	300	80
Pipes (U/G):	**HPChWo**	**HPCRW**	**HPDRW**	
Capital cost (R/m)	3 380	2 920	2 920	
Maint. cost (R/m)	50	80	80	

The cost that was assumed for electricity was R1 120 per kW per annum. This figure was used in the calculations for cooling, pumping and fan-related running costs. All these costs were related back to a present value (PV) cost and an interest rate of 15% over a period of 10 years was used. This made it possible to compare the costs of the various simulation models for cooling, pumping and fan requirements.

10.4.3. Unit costs for fan requirements

In order to evaluate the total PV costs relating to fans, certain unit costs in terms of R/kW had to be assumed to be able to obtain the running, maintenance and capital costs related to fans for surface and underground. A total PV cost for fans on surface and underground was calculated and will be discussed in detail in the sub-sections that follow. Table 10.4.3 below shows the unit costs that were assumed for fans on surface and underground, for the base-case, global and localised recirculation models. The electricity cost for fans was assumed at R1 120 per kW per annum.

Table 10.4.3
Unit costs for fans on surface and underground

Item description	Maintenance cost (R/kW)	Capital cost (R/kW)
Main fans on surface	320	3 500
Recirculation fans underground	380	3 200

A sensitivity analysis of the effect of a change in the capital cost for fans was done and it was found that in terms of the global figure, a change in the capital amount would not have a significant influence on the total cost. The reduction in total cost related to fans was associated more with the actual running cost. The reduction in fan power required would therefore have a major impact on the results for the running costs. A reduction in the capital outlay would therefore also be applicable due to the reduced total power input requirement for fans on surface and underground, as mentioned before.

10.5. PV costs of global re-circulation of air

The total PV cost for the air being recirculated globally is made up of the sum of the total PV costs for the fan requirements and the total PV costs for the cooling and pumping required. It was therefore necessary to evaluate these two aspects in detail in order to be able to compare the various total PV costs related to each recirculation model. These aspects will be dealt with in detail in the sub-sections that follow.

10.5.1. PV costs for cooling and pumping

From all the relevant physical input parameters specified, as well as the cooling that was required for all the levels, a costing model was drawn up for the base-case, as well as for all the relevant simulation models for the global recirculation of return air. Table 10.5.1 below summarises the PV costs related to the cooling and pumping needed for various percentages of air recirculated globally.

Table 10.5.1
Total PV costs of cooling and pumping for global recirculation of return air

Costs in (kR)	**Percentage of air recirculated**					
	0%	**20%**	**30%**	**40%**	**50%**	**60%**
Running costs:						
Surface	9 096	9 096	9 096	9 096	9 096	9 096
Underground	77 640	80 866	82 310	85 289	89 615	92 882
Maintenance costs:						
Surface	9 313	9 313	9 313	9 313	9 313	9 313
Underground	50 068	50 450	50 102	50 207	50 178	50 138
Capital costs:						
Surface	299 000	299 000	299 000	299 000	299 000	299 000
Underground	796 935	802 846	798 300	799 870	799 020	797 867
Total costs:						
Surface	317 409	317 409	317 409	317 409	317 409	317 409
Underground	924 643	934 162	930 712	935 366	938 813	940 887
Total cooling costs - Surface and underground:	1 242 052	1 251 571	1 248 121	1 252 775	1256 222	1 258 296
Total PV cooling costs:	1 829 260	1 853 279	1 854 233	1 871 281	1891 997	1 907 039

The plant capacity on surface was kept the same for all the simulation models and therefore the amount of water that was sent down the shaft was kept at a constant figure of 750 kg/s. This meant that the only cooling figure required for a plant that would change would be for the plant on the level 3 500 m underground. This aspect is clearly shown in the table above. The results show that there is an increase in the PV costs for cooling and pumping as the amount of air that is recirculated increases. This could be expected due to the increase in cooling requirements, as was indicated before. The costs related to surface cooling stayed the same because a decision was made that the surface cooling requirement would stay the same for all the simulation models. Therefore, only the underground cooling requirements increased.

The table above is a summary of all the costs relating to the base-case and global recirculation models (various percentages). The details are given in Annexure A.

10.5.2. Total PV costs of fans

Annexure B gives a complete breakdown of the running, maintenance and capital costs associated with fans on surface and underground. The unit cost for maintenance and capital was an estimated figure, as was mentioned before. These figures were repeated in all the simulation models and made it possible to undertake a PV cost comparison.

Table 10.5.2 below gives a summary of all the various costs. The purpose of the exercise was to obtain an indication of the total PV cost associated with the global recirculation of return air and, if possible, to detect any trends in terms of a reduction or increase in total PV costs when the amount of recirculated air was changed.

Table 10.5.2
Total PV of fan costs for global re-circulation of return air

Costs in (kR)	Percentage of air recirculated 0%	20%	30%	40%	50%	60%
Running costs:						
Surface	20 707	9 120	6 571	4 602	3 105	2 004
Underground	0	760	1 131	1 497	1 860	2 221
Maintenance costs:						
Surface	5 914	2 605	1 877	1 314	887	572
Underground	0	258	383	508	631	753
Capital costs:						
Surface	64 686	28 491	20 526	14 375	9 700	6 262
Underground	0	2 171	3 229	4 276	5 312	6 344
Total costs:						
Surface	91 308	40 216	28 974	20 290	13 692	8 839
Underground	0	3 190	4 744	6 281	7 803	9 318
Total fan costs - Surface and underground	91 308	43 405	33 717	26 572	21 495	18 157
Total PV costs - Fans surface and underground	198 292	94 617	73 750	58 404	47 549	40 466

From the results obtained it was obvious that there was a large decrease in the PV costs for the fans when compared with the base-case scenario. This was due to the large decrease in pressure drop and air volume flow when global recirculation of air took place. This led to an overall decrease in the running, maintenance and capital costs for all the various recirculation models as the percentage of recirculated air was decreased.

10.5.3. Total PV costs for cooling and fans

From the results obtained above, a summary was compiled of all the costs related to cooling, pumping and air-flow requirements. The costs were subdivided into running, maintenance and capital costs relating to cooling and fans for surface and underground. PV costs for the models were then also derived. Table 10.5.3 below gives a breakdown of the total PV costs for different percentages of global recirculation of air.

Table 10.5.3
Total PV costs for global recirculation of return air

Costs in (kR)	Percentage of air recirculated					
	0%	20%	30%	40%	50%	60%
Running costs:						
Surface	29 803	18 216	15 667	13 698	12 201	11 100
Underground	77 640	81 626	83 441	86 786	91 475	95 103
Maintenance costs:						
Surface	15 227	11 918	11 190	10 627	10 200	9 885
Underground	50 068	50 708	50 485	50 715	50 809	50 891
Capital costs:						
Surface	363 686	327 491	319 526	313 375	308 700	305 262
Underground	796 935	805 017	801 529	804 146	804 332	804 211
Total costs:						
Surface	408 717	357 625	346 383	337 699	331 101	326 248
Underground	924 643	937 352	935 456	941 647	946 616	950 205
Total fan and cooling costs - Surface and underground:	1333 360	1294 976	1 281 838	1 279 347	1 277 717	1 276 453
Total PV costs - Fans and cooling:	2027 553	1947 896	1 927 984	1 929 685	1 939 546	1 947 505

The comparative results shown in the table above indicated a very interesting trend with regard to the total PV cost. The total PV cost for the base case was an amount of R 2,03 billion, which decreased gradually with an increase in recirculation up to 30%, after which it increased again. This finding is very important as it indicates that the optimum amount of recirculation would be around 30% global recirculation of return air.

The results given above are shown more clearly in graphical form, specifically with regard to the trend in terms of the total PV cost. Figure 10.5.3 below is a graphical representation of the total PV cost versus the amounts of air recirculated globally. The amounts recirculated varied as indicated before, from 0% to 60%.

The graph also clearly shows the downward trend in the total PV cost for the recirculation fans, as well as the increase in the total PV cost for cooling for the various recirculation percentages.

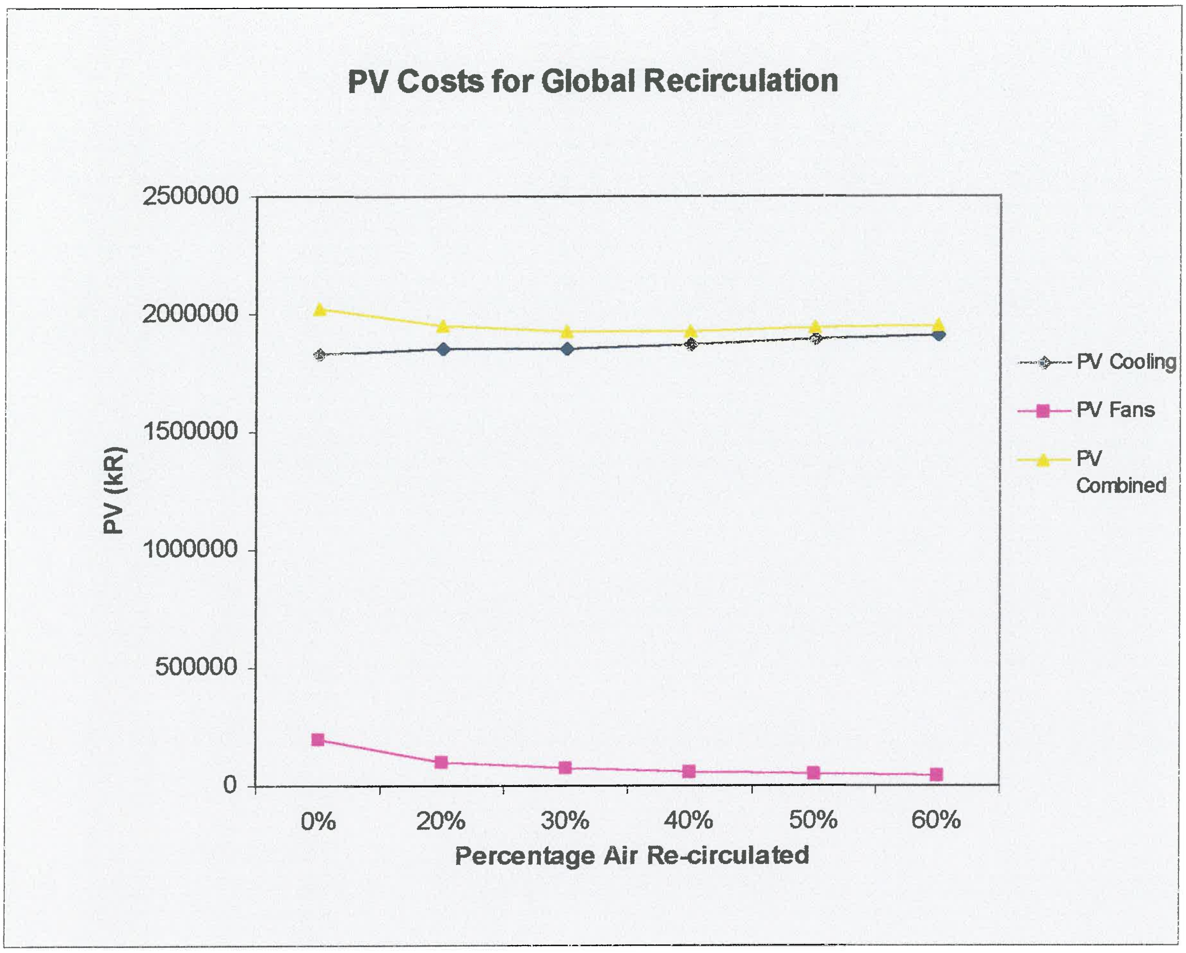

Figure 10.5.3 Total PV costs for the global recirculation of return air

10.5.4. Interpretation and evaluation of PV results for global recirculation

From the results obtained it is obvious that not many of the various parameters used in the base case changed in the global air recirculation simulation. This was to be expected as it was decided to keep the basic layout the same, but the factor expected to change would be the fan input power requirements on surface and underground. It was mentioned before that the cooling should stay the same because of the fact that the total amount of air down the sub-vertical shaft stayed the same as in the base case. The cooling required for each level was also expected not to change and this was confirmed by the results obtained. However, the results for the fan input power showed quite remarkable changes with regard to the power required for the base case compared with the combined power required for the global recirculation fans and the reduced input power required for the fans on surface. This was the result of a very large decrease in the pressure drop across the fans on surface when global recirculation was introduced. The total fan input power requirements for surface and underground were lower for all the recirculation percentages compared with the base case (almost 50 to 80% lower than the figure for the base case). From the graph (Figure 10.5.3) it is also obvious that the optimum percentage of air to be recirculated globally would be in the region of 30%. This is therefore an important aspect to keep in mind when the air and cooling requirements for an ultra-deep mine are being planned. From this result it is also obvious that recirculation of air is not only important but, in the context of ultra-deep mining planning, compulsory.

10.6. PV costs of localised recirculation of air

The total PV cost for the air being recirculated locally on each level is made up of the sum of the total PV costs for the fan requirements and the total PV costs for the cooling and pumping required. It was therefore necessary to evaluate these two aspects in detail to be able to compare the various total PV costs related to each recirculation model. These aspects will be dealt with in detail in the sub-sections that follow.

10.6.1. Total PV costs for cooling and pumping

From all the relevant physical input parameters specified, as well as the cooling that was required for all the levels, a costing model was drawn up for the base case and for all the relevant simulation models for the localised recirculation of return air. Table 10.6.1 below summarises the PV cost related to the cooling and pumping needed for various percentages of air re-circulated locally.

Table 10.6.1
Total PV costs of cooling and pumping for localised recirculation

Costs in (kR)	**Percentage of air recirculated**					
	0%	**20%**	**30%**	**40%**	**50%**	**60%**
Running costs:						
Surface	9 096	9 096	9 096	9 096	9 096	9 096
Underground	77 640	82 338	82 345	82 800	84 241	86 567
Maintenance costs:						
Surface	9 313	9 313	9 313	9 313	9 313	9 313
Underground	50 068	52 403	52 362	53 007	53 224	53 365
Capital costs:						
Surface	299 000	299 000	299 000	299 000	299 000	299 000
Underground	796 935	822 085	826 238	831 956	832 816	831 051
Total costs:						
Surface	317 409	317 409	317 409	317 409	317 409	317 409
Underground	924 643	956 826	960 945	967 763	970 281	970 983
Total cooling costs - Surface and underground	1 242 052	1 274 235	1 278 354	1 285 172	1 287 690	1 288 392
Total PV costs cooling	**1 829 260**	**1 889 707**	**1 893 689**	**1 904 928**	**1 914 109**	**1 924 725**

As mentioned before, the capacity of the plant on surface was kept the same for all the simulation models and therefore the amount of water that was sent down the shaft was kept at a constant figure of 750 kg/s. As in the case of global recirculation of air, this meant that the only required cooling figure for a plant that would change would be for the plant on level 3 500 m underground. The results obtained show that the PV costs for cooling and pumping increase as the amount of air that is recirculated increases. This was expected due to the increase in cooling requirements, as was indicated before. The costs related to surface cooling stayed the same because it was decided that the surface cooling requirement would stay the same for all the simulation models. The only increase in cooling requirements was therefore underground. The table above is a summary of all the costs relating to the base-case and localised recirculation models (various percentages) and the detailed analysis for the running, maintenance and capital costs for the cooling and pumping are shown in Annexure C. It therefore appears that the increased cooling required on the various levels had a negative impact on the total PV cost structure for increased air recirculation percentages. The decrease in the bulk air cooling required on level 3 500 m below surface therefore did not help in terms of the total cost-savings structure.

10.6.2. Total PV costs of fans

Annexure D gives a complete breakdown of the running, maintenance and capital costs associated with fans on surface and underground. The unit cost for maintenance and capital was an estimated figure, as was mentioned before. These figures were repeated in all the simulation models and made it possible to undertake a PV cost comparison. Table 10.6.2 below gives a summary of all the various costs. The purpose of the exercise was to obtain an indication of the total PV cost associated with localised recirculation of return air and, if possible, to detect any trends in terms of a reduction or increase in total PV costs for fans when the amount of recirculated air was changed.

Table 10.6.2.
Total PV costs of fans localised recirculation of return air

Costs in (kR)			**Percentage of air recirculated**			
	0%	**20%**	**30%**	**40%**	**50%**	**60%**
Running costs:						
Surface	20 707	11 697	8 474	5 966	4 031	2 520
Underground	0	36	53	70	86	102
Maintenance costs:						
Surface	5 914	3 341	2 420	1 704	1 151	720
Underground	0	12	18	24	29	35
Capital costs:						
Surface	64 686	36 540	26 471	18 638	12 591	7 873
Underground	0	104	153	200	247	292
Total costs:						
Surface	91 308	51 578	37 366	26 308	17 773	11 113
Underground	0	152	224	294	363	428
Total fan costs -	91 308	51 730	37 590	26 602	18 136	11 541
Surface and underground						
Total PV costs	198 292	112 359	81 658	57 804	39 426	25 111

From the results obtained it was obvious that there was a large decrease in the PV costs for the fans when compared with the base-case scenario. This was due to the large decrease in pressure drop and air volume flow when localised recirculation of air took place. This led to an overall decrease in the running, maintenance and capital costs for all the various recirculation models as the percentage of recirculated air was decreased.

10.6.3. Total PV costs for cooling and fans

From the results obtained above, a summary was compiled of all the costs related to cooling, pumping and air-flow requirements. The costs were subdivided into running, maintenance and capital costs relating to cooling and fans for surface and underground. PV costs for all the models were then also derived. Table 10.6.3 below gives a breakdown of the total PV costs for different percentages of localised recirculation of air.

<u>Table 10.6.3</u>
Total PV costs for cooling and fans for localised re-circulation of return air

Costs in (kR)	Percentage of air re-circulated					
	0%	20%	30%	40%	50%	60%
Running costs:						
Surface	18 192	18 192	18 192	18 192	18 192	18 192
Underground	155 280	164 676	164 690	165 600	168 482	173 134
Maintenance costs:						
Surface	18 626	18 626	18 626	18 626	18 626	18 626
Underground	100 136	104 806	104 724	106 014	106 448	106 730
Capital costs:						
Surface	598 000	598 000	598 000	598 000	598 000	598 000
Underground	1 593 870	1 644 170	1 652 476	1 663 912	1 665 632	1662 102
Total costs:						
Surface	634 818	634 818	634 818	634 818	634 818	634 818
Underground	1 849 286	1 913 652	1 921 890	1 935 526	1 940 562	1941 966
Total fan and cooling costs - Surface and underground	1 333 360	1 325 965	1 315 944	1 311 774	1 305 826	1299 933
Total PV costs - fans and cooling	2 027 553	2 002 066	1 975 348	1 962 732	1 953 535	1949 837

The comparative results shown in the table above also indicated a very interesting trend with regard to the total PV cost. The total PV cost for the base case was a total amount of R2,03 billion, as mentioned before, the main difference being that the there was no 'turning point' in terms of the total PV cost profile. The graph continued a downward trend but did not show total PV cost figures lower than those achieved with the global recirculation model. This finding is also very important as it indicates that when recirculation is being considered in the preliminary design of air and cooling requirements for an ultra-deep mine, global recirculation would, in most cases, be the better option to consider.

The results given above are shown more clearly shown in graphical form, specifically with regard to the trend in terms of the total PV cost. Figure 10.6.3 below is a graphical representation of the total PV cost versus the amount of air recirculated locally. The amounts recirculated varied as indicated before, from 0% to 60%.

The graph also clearly shows the downward trend in the total PV cost for the recirculation fans, as well as the increase in the total PV cost for cooling for the various recirculation percentages.

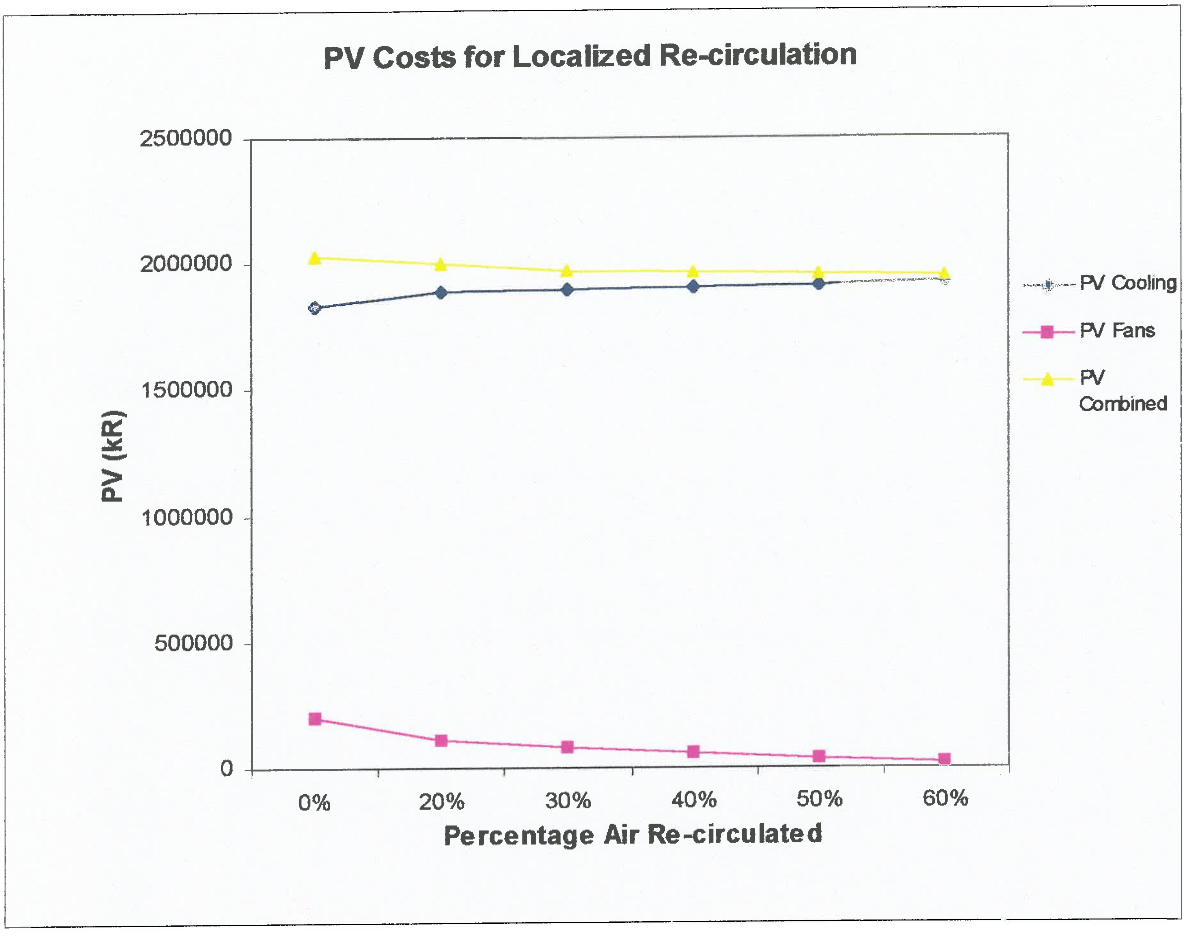

Figure 10.6.3 Total PV costs for the localised recirculation of return air

10.6.4. Interpretation and evaluation of results for localised re-circulation

From the graph it is obvious that the total PV cost shows a downward trend but, when compared with the results obtained for the global recirculation models, it also shows a higher PV cost for all the localised recirculation percentages considered. This aspect has to be kept in mind in planning the ventilation requirements for an ultra-deep mine when a longwall follow-behind mining layout is considered. It must also be noted that the saving in costs incurred with a lower bulk air requirement was cancelled out by the fact that there was a higher cooling requirement for all the localised recirculation levels.

10.7. Comparative PV costs of global and localised recirculation

The combined total PV costs for global and localised recirculation for various recirculation percentages is shown in Figure 10.7 below.

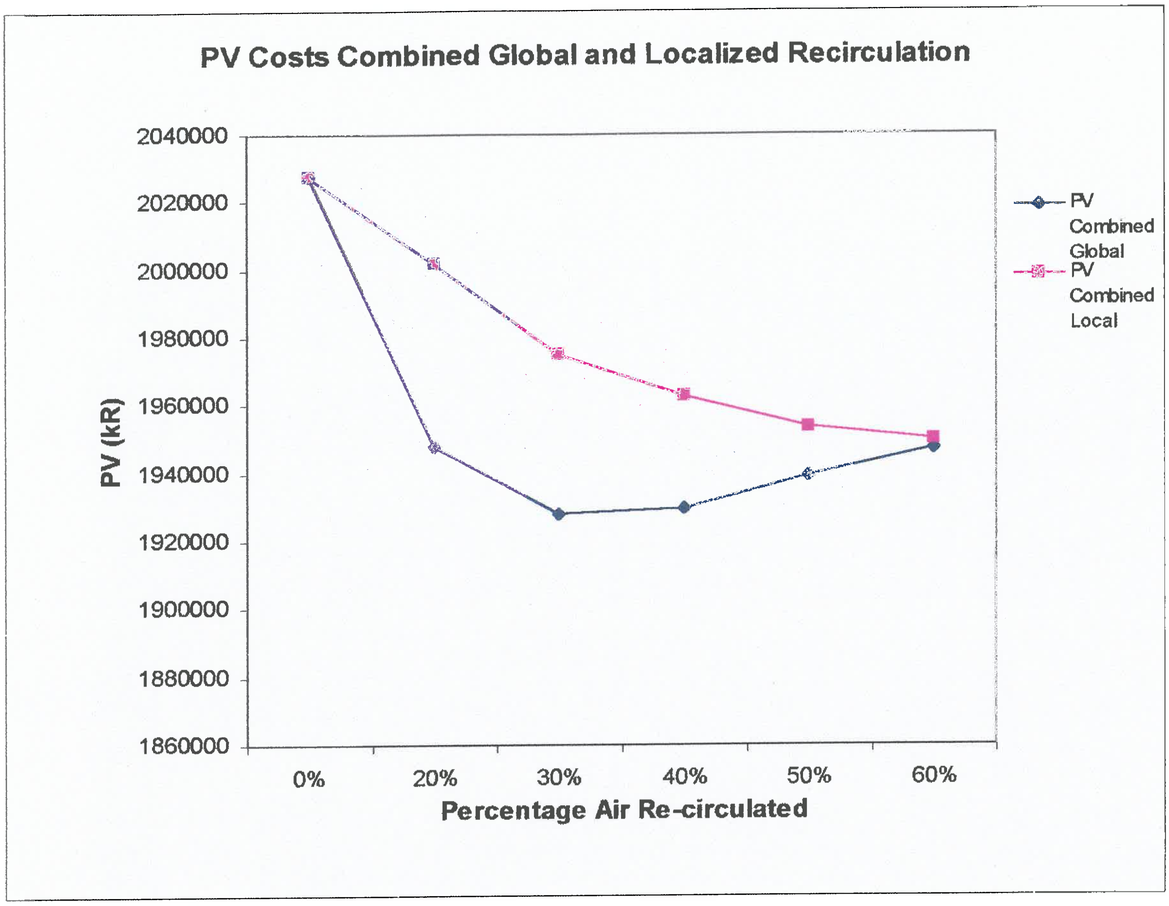

Figure 10.7 ***Combination of total PV costs for global and localised recirculation***

This graph is a clear indication that for any percentage of air recirculated, global recirculation would be the best option to consider in planning the ventilation requirements for an ultra-deep mine when a longwall follow-behind mining layout is planned. The optimum percentage of air globally recirculated to use in such planning is indicated as between 30 and 40%. In order to quantify the total PV cost finding in terms of dollar/ounce produced, the following assumptions and calculations have to be made:

Tons mined from stope per month	192 954	(from base case)
Tons produced per annum (million)	2.32	
Average grade assumed (g/t)	10	
Gold produced (kg)	23 155	
Gold produced (ounces)	815 038	
Rand/dollar exchange rate	6.,1	

Table 10.7.1 shows a summary of the total cost in dollar/ounce produced. For the 30% global recirculation option, a figure of $38.9/ounce of gold produced is indicated.

Table 10.7.1
Summary of total costs in terms of dollar/ounce of gold produced

	Percentages of air recirculated					
	0%	**20%**	**30%**	**40%**	**50%**	**60%**
Total PV costs Global recirculation	2 027 553	1 947 896	1 927 984	1 929 685	1 939 546	1 947 505
Total PV costs Localised recirculation	2 027 553	2 002 066	1 975 348	1 962 732	1 953 535	1 949 837
Costs in dollar/ounce Global recirculation	40.9	39.3	38.9	39.0	39.2	39.3
Costs in dollar/ounce Localised recirculation	40.9	40.4	39.9	39.6	39.4	39.4

The results obtained in the table above can be represented in graphical format and are shown in Figure 10.7.1 below.

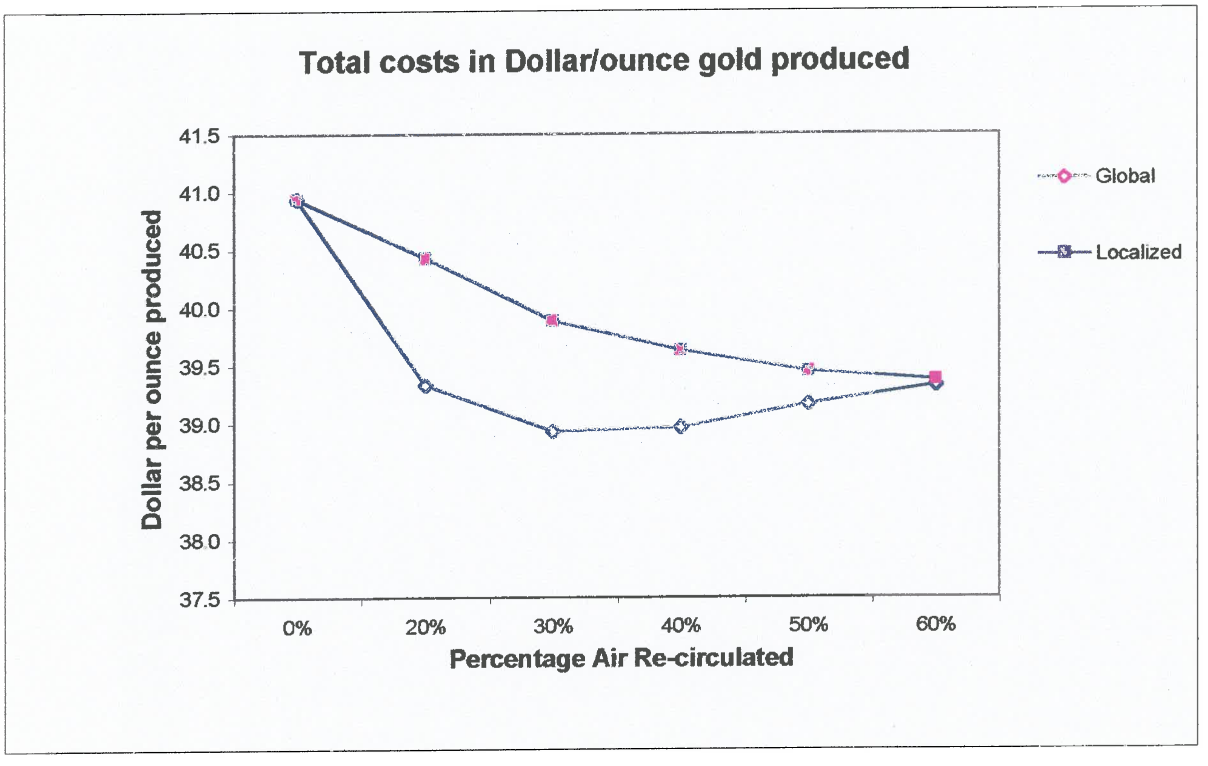

***Figure 10.7.1** Total costs in terms of dollar/ounce of gold produced*

10.8. Evaluation of results for recirculation of return air

It was shown in this study that the inclusion of controlled recirculation strategies in the planning of the ventilation requirements for an ultra-deep mine, when using a longwall follow-behind mining layout, would have beneficial economic effects. The proper application of recirculation strategies can result in significant reductions in the fan input power requirements. Recirculation greatly reduces the impact of auto-compression on total heat load. For the purposes of this study, the negative impact of 'too much' recirculated air was ignored as it was assumed that no matter what the recirculation percentage is, it would be viable in terms of ensuring a safe working environment.

Burton specified a minimum amount of 2,5 kg/s/kt of fresh intake air. In the worst-case scenario of 374 kg/s of air down the main vertical shaft, with 60% air being recirculated globally in this study, which is 216 kg/s/kt less than proposed by Burton, a possible problem becomes apparent. The total amount of air available for the dilution of gases and the minimum supply of fresh air in terms of underground mines need to be investigated.

In planning a recirculation system for an ultra-deep mine, it appears from this study that it would be best to adopt a global recirculation system when a longwall follow-behind mining layout is considered since a localised recirculation system will then be less cost-effective in terms of the total PV cost for cooling and fans. For the purpose of this exercise, a combination of local and global recirculation was not considered and this needs to be investigated further.

When a longwall follow-behind mining layout is used, the optimum percentage for global recirculation of air has been found to be approximately 30%, which means a possible saving of 5% in total PV costs. In the example used in this study, the total PV cost for the base case without any recirculation was R2.03 billion, which gives a total PV cost saving of R110 million if the life of the mine is taken at 10 years and an interest rate of 15% is used. It must be noted that in all the recirculation models used, the cooling requirements were not optimised because in almost all the stopes an ACP of more than the 300 W/m^2 that was proposed earlier was available. On the basis of the total PV cost for the proposed design, this could imply large savings in terms of cooling requirements. For the global recirculation of 30% used in this example, a ventilation cost of $38.9/ounce was indicated. It must be emphasised and kept in mind that a single method and layout were used and therefore this is a simplistic and conservative example of possible savings.

11. Conclusions and recommendations

The work in this document has been subdivided into various sub-sections and the conclusions will be dealt with according to the various objectives that were set.

11.1. Relevant environmental factors and dependence on depth

- The findings of the investigation indicated that the cost-effective provision of acceptable thermal conditions in the workplace will be the greatest environmental control challenge in ultra-deep mining, with the impact of most other environmental aspects not expected to differ materially from what prevails at current mining depths.

- The most notable exception, barometric pressure (and its potential to cause baro-trauma and increase the toxicity of airborne pollutants), is separate field of expertise and needs to be dealt with on its own in detail.

- Owing to their independence from depth, physical stresses such as noise, vibration, lighting/visibility, etc. are not expected to differ materially from what prevails at current depths and accordingly, present standards and exposure limits for their control can be regarded as sufficiently protective for workers in ultra-deep mining environments. In contrast, heat stress will tend to increase with depth.

11.2. Controlling the effects of heat stress on workers

- The findings on heat stress were based on locally and internationally applied standards, exposure limits and indices and, to the extent that the information is available, include the bases for thermal limits in hot, humid underground mines.

- The limited information available from international sources that is relevant to underground mining appears to indicate that more and better knowledge of human heat stress in a mining context has been developed locally. Accordingly, planning for cooling and ventilation requirements in ultra-deep mines should be based on the combined best features of relevant heat stress indices and locally developed knowledge of human tolerance limits for safe and productive work.

- The ability of the environment to remove metabolic heat from workers' bodies, and thus, to limit the negative impact on their health, safety and productivity, depends primarily on wet-bulb temperature and air velocity, with wet-bulb temperature having been proved to be the most useful single-measurement indicator of environmental heat stress. Although wet-bulb temperature alone is of limited value under conditions of high air velocity and radiant temperature, such conditions would be relatively uncommon in ultra-deep mining. Possible exceptions involving high radiant temperature, as a result either of newly exposed rock or the use of diesel-powered equipment, would indicate the need to consider this parameter in assessing such situations, either directly or by means of an appropriate heat stress index. Dry-bulb temperature has been shown to have a limited impact on the acceptability of workplace environments, with 37°C presently regarded as the upper limit for South African mines.

11.3. Heat stress indices and limits

❖ Any heat stress index or indices ultimately adopted for ultra-deep mining should satisfy the following criteria:

- Be applicable to and accurate within the range of conditions for which it will be used
- Take cognizance of all relevant parameters of heat stress
- Be applicable through simple measurements and calculations, i.e. practicable
- Apply valid weighting to all factors considered, in direct relation to their contribution to total physiological strain and impact on work performance
- Provide an appropriate and practical basis for determining and enforcing regulatory standards and exposure limits.

11.4. Design and planning

❖ A number of heat stress indices meet the above requirements to varying extents, but only a few satisfactorily address criteria for underground applications. Among those potentially suited to the design of cooling and ventilation systems, Air Cooling Power (ACP) appears to be the most useful as it is a rational index combining all determinants of environmental cooling capacity and relates directly to engineering design parameters. For planning purposes, an ACP level of 300 W/m^2 should be considered the minimum requirement.

11.4.1. Workplace monitoring

❖ Workplace monitoring should be performed on an ongoing basis and without undue reliance on specialised equipment or personnel, indicating the need for a highly practicable empirical index. Internationally, Wet-bulb Globe Temperature (WBGT) is the most widely used heat stress index, endorsed by ISO, the American Conference of Governmental Industrial Hygienists and the National Institute of Occupational Safety and Health (USA). Despite its high correlation with physiological responses to work in hot, humid environments, WBGT is less than practical as a means of routinely assessing environmental heat stress underground, mainly due to the number of parameters to be measured and the relatively high cost of purpose-designed instrumentation. Although WBGT estimates of heat stress become progressively less accurate under conditions of reduced humidity and where air velocity exceeds 1,5 m/s, this would not necessarily be a disadvantage in critical workplaces, where humidity levels are likely to be high and air velocities low. Contra-indications for the use of WBGT in ultra-deep mining relate to its limited practicability underground and the availability of more suitable alternatives.

❖ The wet-kata is an improvement over the wet-bulb temperature as a means of determining cooling power and assessing heat stress in a particular environment because it considers the combined effects of convection, radiation and evaporation.

❖ Wet-bulb temperature, as a single-measurement heat stress index, provides the best combination of practicability and accuracy, the latter indicated by its high correlation (0,8-

0,9) with physiological strain among exposed workers. Given the common use of wet-bulb temperature as an environmental design criterion and the fact that it is a principal determinant of Air Cooling Power (ACP), its use for monitoring and assessment should not result in serious discrepancies between ACP design levels and the conditions ultimately achieved in the workplace.

❖ Although it is not uncommon, both locally and internationally, for wet-bulb temperatures in underground workplaces to exceed 32°C, the acceptable limit for design purposes should be between 27°C and 28°C, with the upper limit for routine work set at 32,5°C. There is nothing in the way of subsequent research findings to support an expansion of these upper limits. On the contrary, their basis on physiological tolerance criteria, together with the fact that work performance and safety would have a critical impact on the success of ultra-deep mining, may indicate a need for lower wet-bulb temperature limits than those currently applied. Accordingly, decisions relating to the thermal limits ultimately adopted for ultra-deep mining should take into account the impact of heat stress on workers' performance and cognitive ability.

11.5. Issues identified by industry practitioners

❖ Input from environmental control practitioners indicates that the most critical interactions between environmental factors, most notably heat, and operational factors/contributors/constraints involve aspects that are closely related to the thermal environment. Examples of this are discussed in detail in this dissertation/report.

❖ This dissertation/report also incorporates the concerns and ideas of various practitioners in the industry and highlights various issues to consider in planning the environmental requirements for an ultra-deep mine.

11.6. Costing comparison for thermal standards and limits

❖ It has been the mining industry's experience thus far that small improvements in thermal conditions have been achieved through large increases in ventilation and, particularly, cooling costs.

❖ In contrast, the results of this study indicate that reducing the reject wet-bulb temperature would have less impact on the overall cost of ventilation and cooling than increasing the stope face air velocity. Over the range of reject wet-bulb temperatures considered (25 to 31°C), cooling and ventilation costs were shown to have a possible variation of up to 30% from the base case of 31°C.

❖ Reducing temperatures in ultra-deep mines would increase the absolute cost of environmental control, but the increase in relative or percentage terms would be less significant than for current mining depths. The work carried out in this study indicates that a given level of ACP in an ultra-deep mine can be achieved at far lower costs by reducing wet-bulb temperature than by increasing air velocity and air quantity. It then follows that, should health, safety and productivity-related requirements dictate a design reject wet-bulb temperature of, say, 25°C, it should be feasible (in terms of ventilation and cooling costs) to meet that criterion, provided a design temperature of 31°C was economically viable in the first place.

- Previous studies quantifying heat stress in terms of ACP have indicated that a level of 300 W/m^2 can be regarded as both safe and conducive to high productivity in nearly all instances.

- The significant cost increases associated with higher air velocities highlight the main conclusion of this analysis, namely that total air-flow quantity should be minimised and wet-bulb temperature reduced to achieve the required ACP in ultra-deep mines. Designing an environmental control system in accordance with these criteria would allow higher wet-bulb temperatures in areas that naturally have higher air velocities, e.g. intake airways, but the general approach indicated is to reduce wet-bulb temperature through cooling, and not to increase air velocity.

- The practicability of thermal design limits, including those for air velocity and wet-bulb temperature, and the physiologically/psychologically related requirements of the work force will be principal determinants of the design criteria ultimately adopted for ultra-deep mines. The investigation of worker-related factors should also receive attention.

- Cost simulations demonstrated the significant impact of higher stope face air velocities on both cooling and ventilation costs. The results therefore forms the basis for the recommendation that environmental control systems in ultra-deep mines should be designed to provide the required thermal environment (in accordance with the many and various criteria relevant to this critical decision) at a minimal total air mass flow rate. Considering the possible high cost variations, minimising the air mass flow rate should certainly be an integral part of an ultra-deep mine's ventilation strategy.

11.7. Recirculation of return air

- It was shown in this study that the inclusion of controlled recirculation strategies into the planning of the ventilation requirements for an ultra-deep mine would have beneficial costing effects.

- The proper application of recirculation strategies can result in significant reductions in the fan input power requirements.

- Recirculation greatly reduces the impact of auto-compression on total heat load. For the purposes of this study, the negative impact of 'too much' recirculated air was ignored as it was assumed that no matter what the recirculation percentage is, it would be viable in terms of ensuring a safe working environment.

- It appears from this study that in planning a recirculation system for an ultra-deep mine using a longwall follow-behind mining layout, a global recirculation system should be adopted.

- There is an indication that when a longwall follow-behind mining layout is used, the optimum recirculation percentage for global recirculation of air is approximately 30%.

- In all the recirculation models used, he cooling requirements were not optimised because in almost all the stopes an ACP of more than the $300W/m^2$ that was proposed earlier was available.

- A possible saving of 5% in total PV costs can be achieved through the adoption of a 30% global recirculation system for a longwall follow-behind mining layout. In the example used in this study, the total PV cost for the base case without any recirculation was R2.03 billion, which gives a total PV cost saving of R110 million. Further savings are possible if the cooling requirements for specific mining areas are optimised and also if a combination of local and global recirculation can be used. This aspect needs further investigation.

- The total cost in terms of dollar/ounce of gold produced when a global recirculation system was used for a longwall follow-behind mining layout was $38.9/ounce.

- It must again be emphasised that the cost analysis conducted during the present investigation was of a comparative and relatively simplistic nature as a result of the need to use a single simulation model. Accordingly, these findings should be interpreted and applied with circumspection and, more importantly, validated for specific mine designs as they are developed.

- In a similar vein and to underscore previous comments, the heat stress limits ultimately adopted for ultra-deep mining should include criteria for work performance in order to ensure that productivity levels contribute to the profitability of such mining projects.