HEATING AND VENTILATION STUDY OF INCO'S CREIGHTON MINE

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Abstract

Heat and Ventilation Study of Inco's Creighton Mine

As near surface deposits are depleted, it becomes increasingly apparent that we will have to mine deeper in order to meet future world demand for metals.

Along with deeper mining comes its associated challenges: increased stresses and seismicity, increased heat load and increased inefficiencies due to hoisting constraints and travel time. All of these challenges, and more, contribute to safety concerns, higher initial capital costs and higher operating costs, which combined, can make deep reserves uneconomic.

In order to meet these challenges, we must closely examine present deep mining infrastructures and operating practices with a view to learn and enhance upon "Best Practices".

This thesis will examine the challenges of providing ventilation within deep, hot mines. Specifically, we will examine Inco Limited's Creighton Mine. A current expansion at Creighton will see mining progress to the 7660 level within the next few years. A key issue, which arises, is the question as to whether Creighton will need a refrigeration system or can it continue to rely on its natural heat exchange capacity.

Résumé

Etude sur la chaleur et l'air ambiant à la mine Creighton d'Inco.

Afin de répondre à la demande mondiale des métaux, Inco réalise l'importance d'accentuer l'exploitation minière plus en profondeur au fur et à mesure que les réserves en surface diminuent.

L'extraction du minerai en profondeur présente des défis techniques: augmentation des stress dus au minage et à l'activité sismique, augmentation de la température ambiante et une perte d'efficacité causée par des distances de transport du minerai et de matériaux plus grandes. Ces difficultés engendrent également des coûts d'opération et un investissement initial en capital supplémentaires et une réévaluation de la sécurité au travail. Tous ces facteurs réduisent la rentabilité des projets miniers en profondeur.

Pour répondre à ces nouvelles exigences, nous proposons d'examiner l'infrastructure et les méthodes d'exploitation minière actuelles en milieu profond afin de mieux comprendre et d'améliorer les meilleures méthodes existantes.

Cette recherche présente les défis reliés à la livraison d'une quantité et qualité adéquate d'air dans les mines profondes et chaudes. Nous étudierons plus particulièrement la mine Creighton d'Inco Limitée. Dans les prochaines années, Inco projette d'accroître le développement de la mine Creighton jusqu'au niveau 7660. Eventuellement, la mine Creighton devra évaluer si les besoins de ventilation de la mine sont mieux servis avec un système de réfrigération ou le procédé actuel d'échange d'air naturel.

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Chapter 1

1.1 Introduction

In order to meet world demand for metals, there is little doubt that we will have to deepen existing mines and explore deposits that were once considered too deep and therefore uneconomical. Examples are Inco's Creighton Mine, which is undergoing a current deepening expansion to the 7660 level. Also, Falconbridge is presently conducting a Feasibility Study of the Onaping deposit, which contains some 20 million tonnes of ore, located between 2000 and 3000 meters below surface.

These are only examples, much of the world's known sulphide base metal reserves lie at depths below 2 km. At these depths, there are associated challenges such as seismicity and rockbursts. Operating costs increase with mining depth as a result of the additional ground support required and regulating air quality becomes more challenging with hotter temperatures as a result of geothermal gradients.

In this thesis, most of the elements that need be considered when designing a ventilation system for a deep hot mine will be examined. Special attention will be given to one of the deepest mines in Canada, which is the Inco Creighton Mine located at Sudbury Ontario.

This Chapter describes Creighton's history, geology, seismicity and present ventilation system. It also provides the terms of reference for designing future ventilation expansion.

Chapter Two outlines general ventilation design and safety implications of its infrastructure, discusses key aspects of the ventilation design, namely; heat load generation, diesel exhaust, blasting fumes, dust contaminants and air velocity.

Furthermore, Chapter Two, outlines in detail, the strength and weaknesses of heat load prediction and airflow requirement prognosis by empirical and mathematical methods.

Chapter Three reviews the innovative methods of heating and cooling air. Examples of past and current practices are discussed.

Chapter Four presents the effect of climatic variations on human body and outlines methods of heat stress quantification and their limitations.

Chapter Five lists principal contributors of heat sources in a mine. The examples of worldwide geothermal gradient data are given.

Chapter Six examines in quantitative manner the phenomena of essentially free form of air conditioning and provides prognosis of estimated cooling and heating capacity.

Chapter Seven describes the various refrigeration techniques available, the theory of mechanical refrigeration and design parameters.

Chapter Eight outlines and compares prognosis of heat load and airflow requirements for various mining depths as a function of varying heat load factor (1993 – 1999 time frame). Provides estimates of mechanical refrigeration requirements and addresses current challenges.

Chapter Nine documents research findings and provides recommendation for future work.

1.2 Creighton Mine

Inco's Creighton Mine, which is one of the deepest and largest mines in the Western Hemisphere, has been continually developing since mining first began there in 1901.

Creighton Mine is located in the Regional Municipality of Sudbury, approximately 18 km west of the city proper. It is owned and operated by Inco Limited, and presently mines 3175 tonnes per day of nickel and copper ore.

The mine comprises two distinct areas named after the shafts that service these areas. The upper portion of the orebody is serviced by No. 3 Shaft; thus, this area is referred to as No. 3 Mine. Due to the low grade of this area, No. 3 Mine predominantly employs bulk mining methods. It is presently in stand by mode.

The deeper section of the orebody, which is of higher grade, is serviced by No. 9 Shaft. This area is called No. 9 Mine and was converted from a selective cut and fill operation to a bulk Vertical Retreat Mining (VRM) operation in the mid-eighties.

1.2.1 No. 3 Mine

Production at Creighton began in 1901 by means of open pit mining. Two years later in 1903, No. 1 shaft was excavated in the footwall, after which No. 2 shaft was sunk in 1905.

Pit production at this point had reached 700 tonnes per day. By 1906, underground mining had begun using the overhand bench method. By 1913, No.

1 shaft was engulfed by the open pit, which created an excavation 200 meters long, 120 meters wide and 56 meters deep. At this point in time, No. 2 shaft was down 106 meters.

No. 3 shaft was collared in 1915 at 55 degrees and by 1924 was down to 1900 level. By 1929, fill method mining was introduced in the form of square set with some cut and fill.

Due to the depletion of high-grade ore, block caving was introduced to No. 3 Mine. In order to hoist the large volumes of ore produced by this method, No. 7 shaft was sunk to the 1900 level. This shaft handled up to 12,700 tonnes per day.

In 1972, No. 3 mine operations were discontinued as a result of secondary blasting problems coupled with poor grade control. In 1974 it was reopened and scheduled for 7300 tonnes per day. The mining method employed to meet this schedule was large diameter blast holes using in-the-hole drills. This method yielded better results than block caving in terms of fragmentation and grade control. No. 3 Mine was shut down again in 1991 and remains so as of today.

1.2.2 No. 9 Mine

Serviced by No. 5 shaft and No. 9 shaft, Creighton No. 9 Mine consisted of all mining below the 2300 level. No. 5 shaft was sunk to the 3800 level between 1934 and 1936. In 1940, No. 6 Winze was sunk from the 4800 level to the 5400 level, and in 1956, No. 8 Winze from 4800 level to the 6600 level. Operations at No. 5 Mine were curtailed in 1985 as a result of a major rockburst.

Between 1965 and 1969, No. 9 Shaft was sunk from surface to the 7137 foot level. The main method presently employed is Vertical Retreat Mining (V.R.M.). A conversion to V.R.M. from mechanized cut and fill (MC&F) in the mid-eighties was necessary due to safety concerns and economics. As MC&F crowns below the 6600 level came within 22 meters of the upper level, severe rockbursts would occur in the stopes.

A ramp is presently being driven from the 7200 level to the 7660 level for the next phase of mining. Most lessons learned with regard to mining using V.R.M. were learned between the 7000 and 7200 level.

As can be seen from the historical highlights, mining at the Creighton Complex has progressed deeper and further away from the original workings. Along with increases in depth came related problems such as ground control, rockbursts, heat and logistics.

1.3 Stresses and Seismicity at Creighton

1.3.1 Regional Geology

The Sudbury Basin is located 70 km northeast of Lake Huron in Ontario and is situated at a contact between the early Proterozoic Huronian Formations of the Cambrian Shield's Southern Province and the Archean plutonic rocks of the Superior province (Cochrane, 1991). It extends 70 km in an east-northeast, west-southwest direction and is about 55 km wide. Figure 1.1 is a plan of the Sudbury Basin and including some mine locations

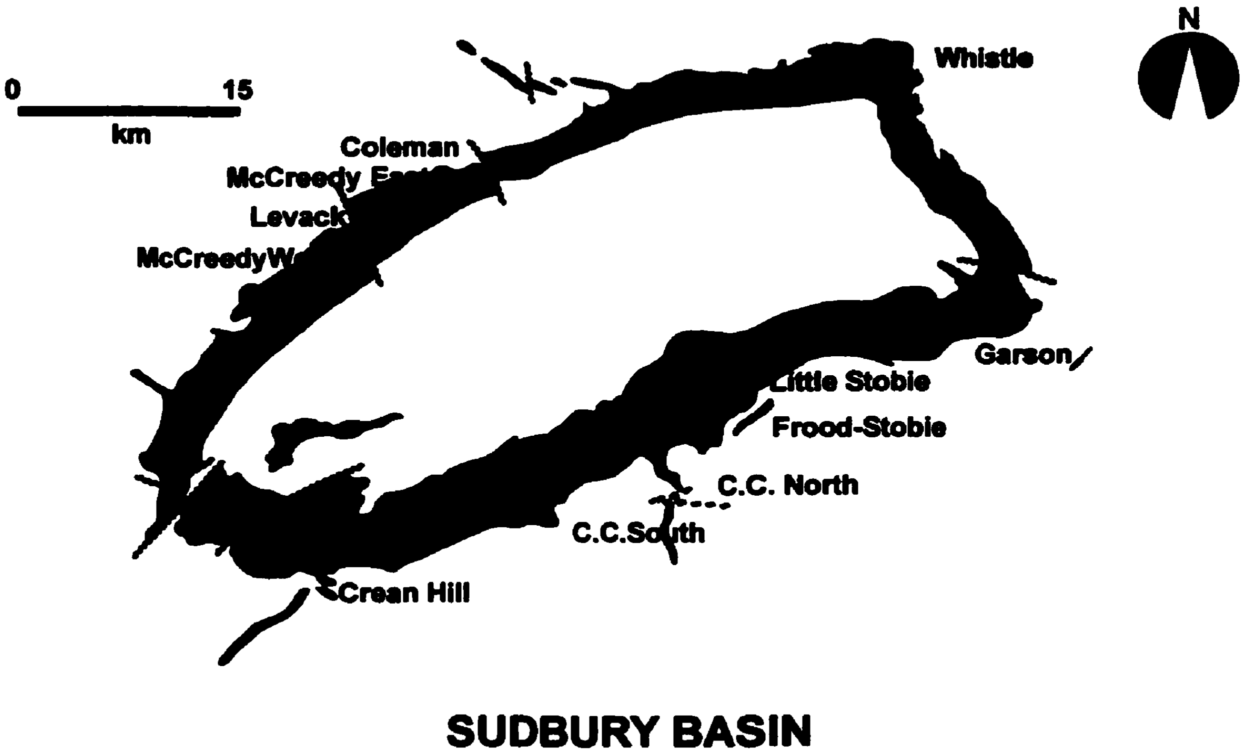

Figure 1.1: Plan of the Sudbury Basin

The basin consists of two major units; these are the Nickel Irruptive and the older White Water Group. The Nickel Irruptive consists of a lower outer layer of Norite and an upper inner layer of micropegmatite (Herget, Pahl, Oliver, 1975).

At the base of the lower layer is a zone of inclusions and sulphide-rich intrusions known as the sub-layer. This sub-layer is the source of nickel, copper, iron and platinum group metals.

The most widely accepted genesis of the basin is that of meteorite impact (Dietz, 1964).

1.3.2 Creighton Geology

Creighton Mine is located at the southeast corner of an embayment of the Nickel Irruptive into the footwall rocks. Generally, the lower Norite member of the main Irruptive is the hanging wall with the footwall rocks being composed of Creighton Granite.

Between the main mass Norite and the footwall rocks lies the discontinuous unit of the Irruptive. It is the usual host for the ore and consists of basic to ultrabasic inclusions in a matrix of Norite and Sulphides (Willock, 1980).

The embayment generally dips at approximately 60° NW but can vary from 90° to 30°. As well, the orebody is found to extend from surface to a depth of at least 3-km.

The Creighton deposit generally can be divided into three orebodies. The two main orebodies associated with the embayment are the 118 orebody and the 400 orebody; the third is associated with a quartz diorite dyke in the footwall and is called the 403 orebody. (Bawden and Coulson, 1993).

The main orebody (the 400), extends from surface level to 7400 level, and has been intersected by diamond drill holes at depths exceeding 3000 meters.

This major ore zone dips at 60° NW and is up to 75 m. in true width as shown in Figure 1.2. Below 5000 level, there exists several structures or shears. The 2001 shear in the hangingwall norites and the #6 shaft shear, which affects the footwall ore zones, are examples of such structures (MacDonald, 1985). Others include the 1290 Shear and the 402 ore pass shear.

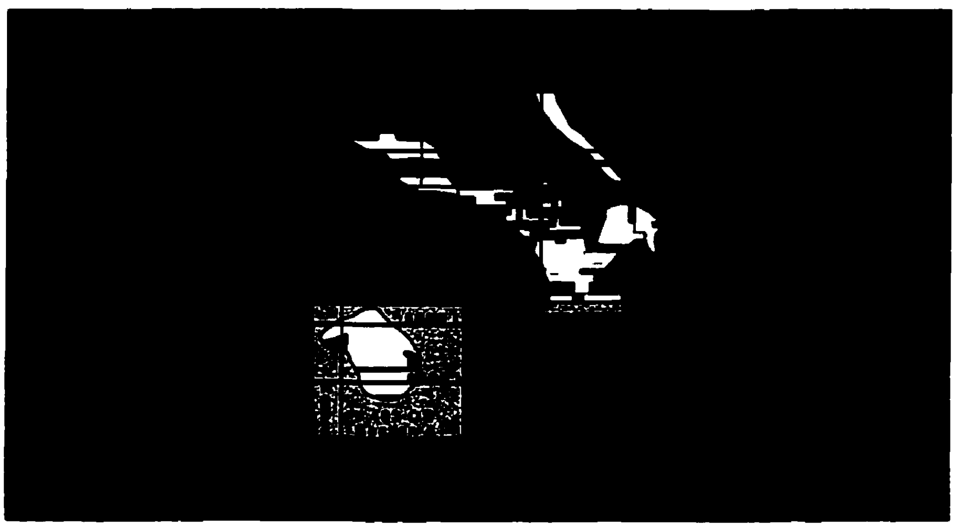

Figure 1.2: Section of Creighton Orebody

1.3.2.1 1290, 2001, RAR (Return Air), 402 Orepass Shears

The 1290 shear begins at approximately the 4000 level and continues down below the 7200m level. This shear is moderately seismically active and has associated with it a large stringer of massive sulphide ore, which extends out below 6800 level. Almost complete mining of this stringer has been possible largely due to a very disciplined and well planned approach of mining in a top down, inward outward sequence. However, seismic activity has caused much damage to any orepass system excavated within its vicinity.

Within the hangingwall are two large shears that run parallel to the orebody. The 2001 and 2002 shears.

The 2001 shear intersects and originates from the orebody on the 6000 level then dips at a shallower angle than the main embayment and diverges away from the orebody around 7200 level (Bawden and Coulson 1993).

Seismically the 2001 shear is the source of most major events (MacDonald 1990). The 2002 shear lies north approximately 240 m of the 2001.

The RAR shear is a steeply dipping structure that descends from 6600 level to below 7200 level. The rock mass adjacent the shear is continuously unstable (Wiles and MacDonald, 1990).

A curvilinear shear that does not extend below the 6600 level, it has been associated with a large 3.2 M_L event on October 30, 1989 which displaced an estimated 600 tonnes of material. It has also precluded the reliable use of any orepass within its vicinity.

1.4 Historical Perspective

Mining at Creighton has progressed from the original open pit in 1901 down to its deepest current horizon 7400 level.

Throughout its 98-year history, many mining methods have been used as the mine deepened. (Open pit, sub-level cave, blasthole, underhand cut and fill, shrinkage, mechanized cut and fill, undercut and fill and vertical crater retreat).

Creighton's history of rockbursts dates back to the early 1930's with the first recorded burst occurring at a depth of 700m in 1935. It was recognized that bursting was a problem generally below 800m in depth. Below this horizon, bursting conditions began to present problems in shafts and raises. In lateral headings, bursting begins below 1220m (Oliver, 1985).

In July of 1984 a rockburst with a recorded magnitude of 4.0 occurred in the hangingwall of 5 Shaft (2300 level to 3800 level). This event caused the closure of 5 shaft thus concentrating present mining operations to the 400 orebody of 9 shaft (4600 level to 7400 level).

This large orebody, 250m in true width, is highly stressed. Early mining of the 400 orebody at depth (6400 level, 6600 level) was successful using the mechanized cut and fill method for primary mining and undercut and fill mining as a secondary method to retrieve pillars and crowns.

The mining success of 6400 to 6600 levels was somewhat overshadowed by injuries caused by local rockbursts within the MC&F units. This matter became much more of a concern as MC&F mining deepened to the 6800 and 7000 levels. With principal stresses being horizontal and 1.7 times the vertical stresses (Herget et al, 1975) stress concentrations at increasing depths exceeded rock strength.

Bursts in crown pillars within 22 m of the next horizon and bursts in rib pillars during stope silling operations on 6800 and 7000 levels were cause for concern. With the advent of VRM mining in the early 1980's, it was seen that pillarless, semi-remote (people removed from within ore zone) bulk mining could provide the safety required in mining at depth. In 1983 a VRM de-stress slot was extracted in the center of the 400 orebody from the last cut of the 6800 level MC&F unit up to 6600 level and VRM mining began in earnest at Creighton.

Today, Creighton is exclusively a VRM mine. Stope sequencing is used for stress redistribution of locations relative to service openings and personnel while de-stressing techniques are used to control local bursting in development headings.

As a result of these control factors, bursting in active headings and service areas are minimized. However, in recent years, some of the largest magnitude rockbursts in Ontario have occurred at Creighton. These bursts generally occur in the hangingwall and footwall and are connected to the existence of the shears that occur there (402, 4001, 4002) and large scale stress redistribution.

To this date, events have not caused considerable damage and injury (with the exception of a 1986, 3.5 event on 6600 level and a 1998, 3.4 event on 6800 level). They remain, however, an area of active research and study.

1.5 Ventilation System

1.5.1 Overview

Inco Limited has an excellent reputation for planning and control of the environment in its underground mines. This standing has been earned over the years by the work of its ventilation engineers. In particular, the work of the Inco Chief Mines Ventilation Engineers, J.G. Rutherford until his retirement in 1978, and succeeded by Dr. J.S. Stachulak, has maintained Inco free from most of the shortcomings of safety and health that have afflicted many other mining companies. Additionally, at times when major extensions or changes to underground systems have been contemplated, Inco has considered it appropriate to seek fresh viewpoints from outside consultants. The combination of internal expertise and occasional external review has provided for successful long-term planning of environmental conditions in Inco's underground mines. This has been a major factor for Inco to remain a well-known and respected company within international circles of mine ventilation expertise (McPherson, 1996).

Creighton is particularly recognized in the world of mine environmental engineering because of its primary heat exchange method, utilizing the heat storage capacity of fragmented rock to provide low-cost air-conditioning throughout the year. Mining of the Creighton orebody commenced with an open pit in 1901. When this progressed subsequently into underground operations it was found that the summer peaks and winter troughs of surface air temperature could be mitigated by drawing intake air through the mass of fragmented rock in and below the old open pit, via box-holes and the underlying slusher drifts, into the main air intake system for the mine. Over the years, this has been further facilitated by collecting the air from the old slusher drifts into a "gathering loop" of airways at the 800 level. The result is that intake air, at this level, is provided at about 3°C throughout the year. The demand for greater quantities of cool air has increased over time because of greater depths of workings, increased tonnages, additional mechanization and enhanced utilization of diesel equipment. That increased demand for air conditioning has been met, to the present time, by larger quantities of air being drawn through the heat exchange area, coupled with extensions of the 800 level gathering airway in order to utilize additional zones of the fragmented rock.

The planning now under way for mining below the 7200 level has necessitated another long view of the future requirements of ventilation and air conditioning at Creighton, and how these may best be met.

1.5.2 Terminology

Mine level designations at Creighton approximate the distance below surface in feet and the No. 9 shaft collar is 1000 feet above sea level.

Air volumes are expressed at a density of 1.26 kg/m^3, the average density at the top of the main intake airway, 243.8 m below surface. The maximum density is about 1.49 kg/m^3, on 7000 level.

In referring to air temperatures, the wet-bulb temperature in degrees Centigrade is used, as it is directly related to the heat content of the air. With the high humidity encountered in mine workings, the wet-bulb temperature is the best indicator of comfort conditions, especially when correlated with air velocity.

1.5.3 Present Ventilation System

The ventilation system is handling some 448 m^3/s through the lower mining areas for a current daily production of 3175 tonnes of ore. Of this amount, 330 m^3/s is being supplied to the workings below 5000 level.

Figure 1.3 is a simplified diagram of this system showing the main intake and exhaust systems, No. 8 shaft, part of the main intake to 7000 level, and No. 9 shaft, also an intake.

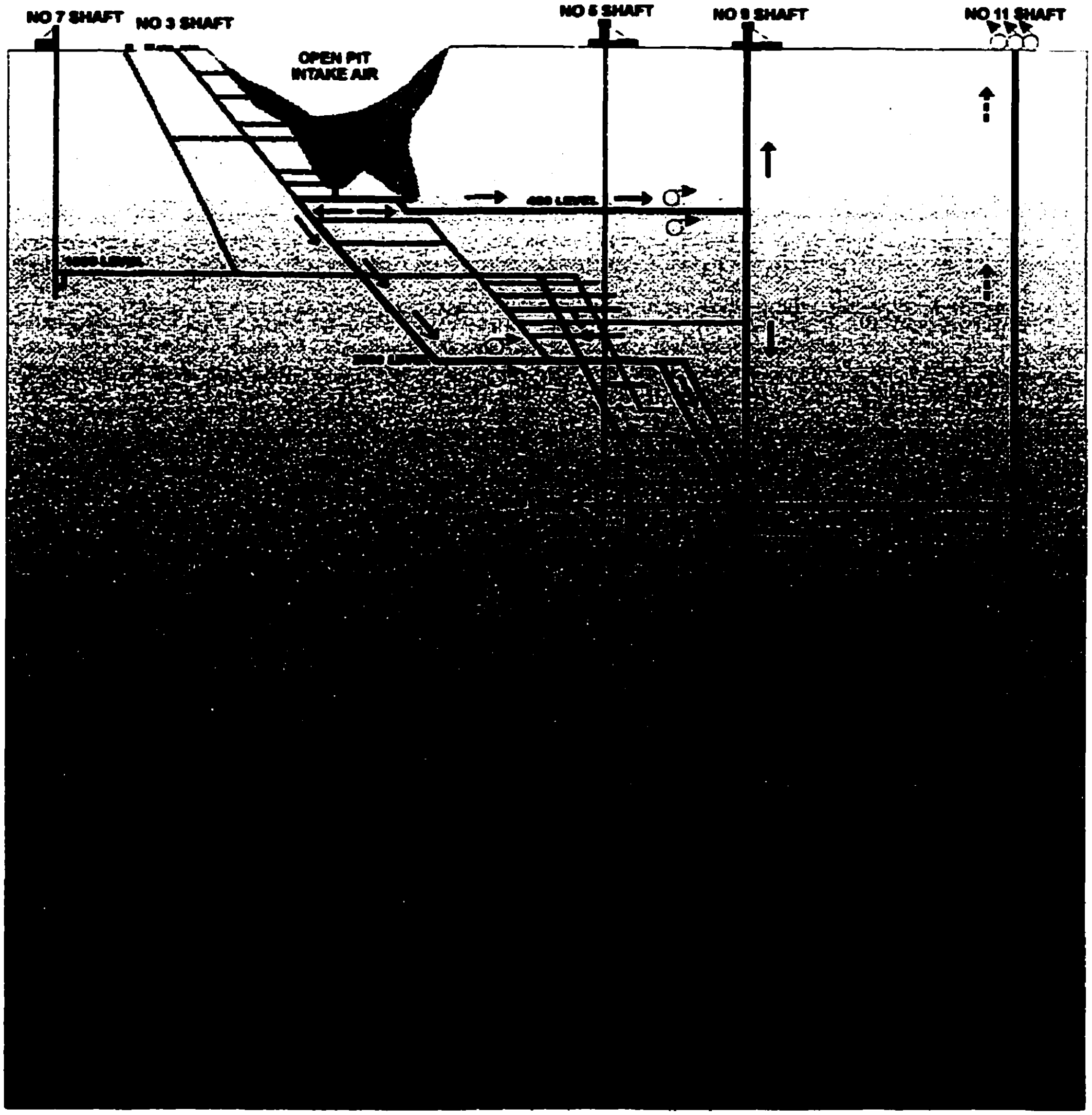

Figure 1.3: Creighton Mine Ventilation Schematic – Present System

The fresh air is drawn down from surface through a large mass of broken rock, measuring 244 m to 183 m in area, and averaging 137 m deep. The broken rock lies above a series of slusher drifts that were formerly used to mine low-grade ore by the caving method in the upper mining area. The air flows down through the drawpoints, across the slusher drifts, into the ore passes, and is collected on 800 level where it enters the main intake airway.

This mass of broken rock acts as a heat exchanger, warming the intake air in winter and cooling the air during the summer. Precipitation causes ice formation in the broken rock mass in winter, and this is melted in the summer; however, the rock itself provides by far the larger heat exchanger effect. Ventilation controls at the ore pass millholes are used to regulate the flow of air to adjust the air temperature seasonally. The average air temperature at the top of the main intake on 800 level is 2.8°C, with a seasonal variation of plus or minus 1.7°C.

The resistance of the main intake system is overcome by two pairs of axial-flow fans in series; one set is on 2600 level and the other on 5000 level. The main fans on 2600 level consist of two 2.44 m diameter fans with a 298 kW motor. There are two intake airways from 2600 to 3800 levels. The two fans on 2600 level are handling 229 m^3/s. On 5000 level there are two 2.29 m fans, each with a 298 kW motor, operating in parallel.

1.5.4 Basis for Designing the Future Ventilation System

It is recognized that the amount of air required to ventilate a mine and provide acceptable environmental conditions depends on the tonnage to be mined, the mining method and the degree of mechanization, among other considerations. Additionally, at Creighton, the major part of the mining would be taking place in areas with rock temperatures over 32 °C, and provision had to be made for adequate heat removal, as well.

Creighton is fortunate to have the large natural heat exchanger to provide cool intake air but increasing amounts of air or mechanical refrigeration are required with depth to contend with higher rock temperatures. In view of the high capital cost of an underground air refrigeration plant and the very high power and maintenance costs, it was decided to defer refrigeration as long as possible through maximum use of the heat exchange area and to increase the volume of air. We estimate that it should be possible to mine to 8000 level and to maintain good working conditions in the mining areas without resorting to major refrigeration plants. However, some small refrigeration units will be required for a few isolated workplaces.

Figure 1.4 shows the increase with depth at Creighton of the virgin rock temperature and of the fresh air intake temperatures for both wet-bulb and dry-bulb, as per late 1970's measurements. The 5000 level fresh air fans are to be eventually removed in the future but it is assumed that wall rock heat pick-up from the newer fresh airways will result in a continuation of the present temperature gradients below 5000 level.

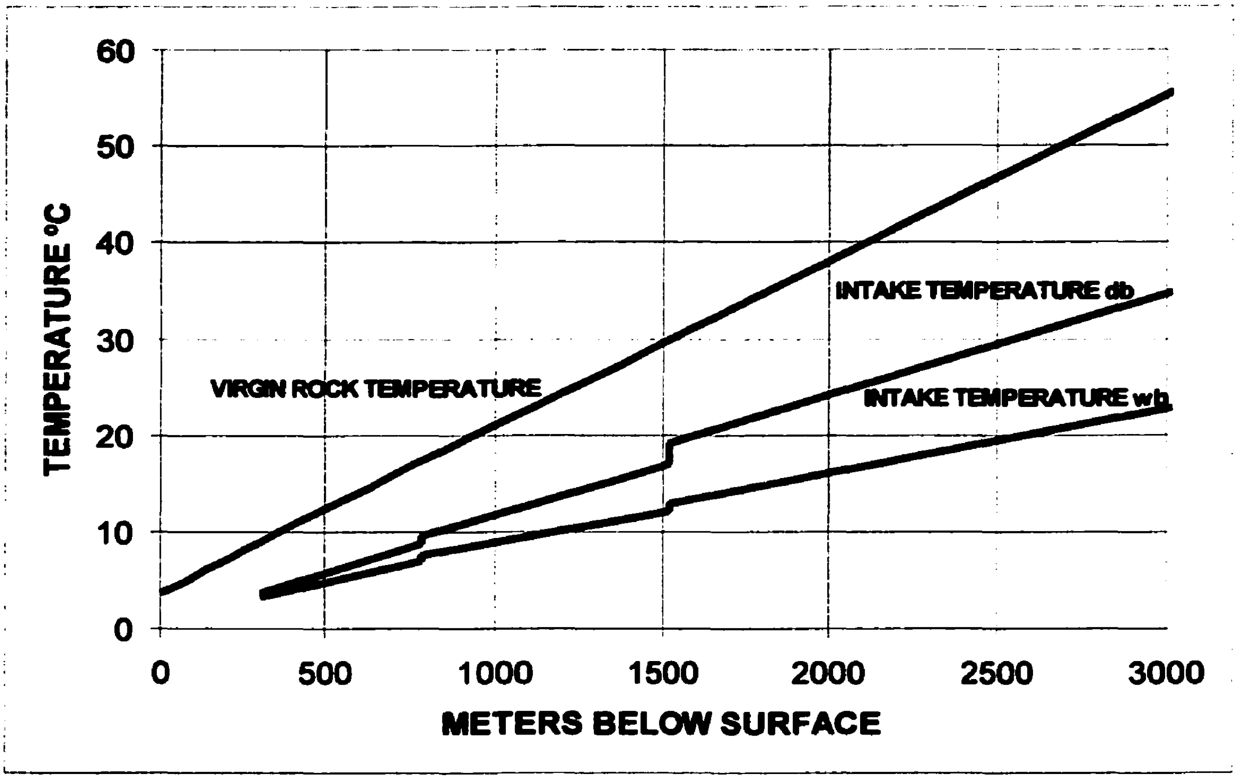

Figure 1.4: Variation of Rock and Intake Air Temperatures with Depth, (After Stachulak, 1979)

Calculations of the heat loads and of the ventilating volumes required were made in 1971, assuming average stope temperatures of both 24°C and 27°C wet-bulb. In 1977, when we had considerably changed production patterns and had increased the ventilation volume, the figures were recalculated and were found to be in substantial agreement.

As mining depths increase, very large quantities of air will be required to remove heat from a mine. Enough heat must be removed to maintain a comfortable work temperature. The fresh air supply will become warmer with increasing depth due to auto-compression, and thus can absorb less heat. Additionally, air at a constant wet-bulb temperature can absorb less heat per kg as the barometric pressure increases. These points are illustrated in Figure 1.5. The lower line shows the heat content per kg of air at the downcast air temperatures. The middle line shows the heat content of air at 24°C wet-bulb, and the top line shows the heat content at 27°C wet-bulb related to depth. Thus, considering a 24°C wet-bulb stope exhaust temperature, a kg of air at 1830m depth will remove 11.8 kJ, at 2438 m, 7.0 kJ, and, at 3292 m, the air will remove no heat and environmental conditions must be met entirely with refrigeration.

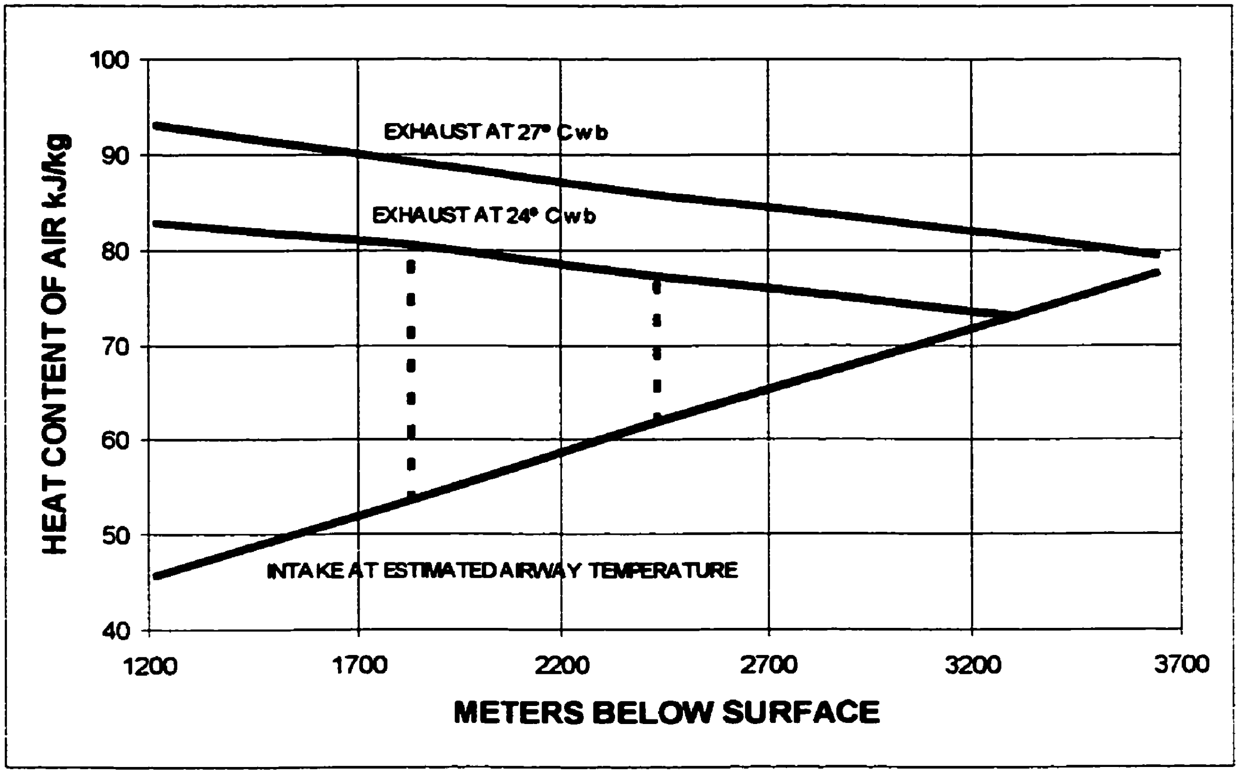

Figure 1.5: Heat Absorbing Capacity of Ventilating Air at Depth, (After Stachulak, 1978)

Figure 1.6 shows the calculated m³/s of air per ton per day required to remove enough heat to maintain average stope temperatures at 24°C and 27°C wet-bulb. The 24°C curve indicates that air quantities will become excessive at about 8000 level (2439 m) below surface; the 27°C curve indicates the same situation at a depth of about 2743 m. Figure 1.6 is based on measurements and data, obtained in the late 70's.

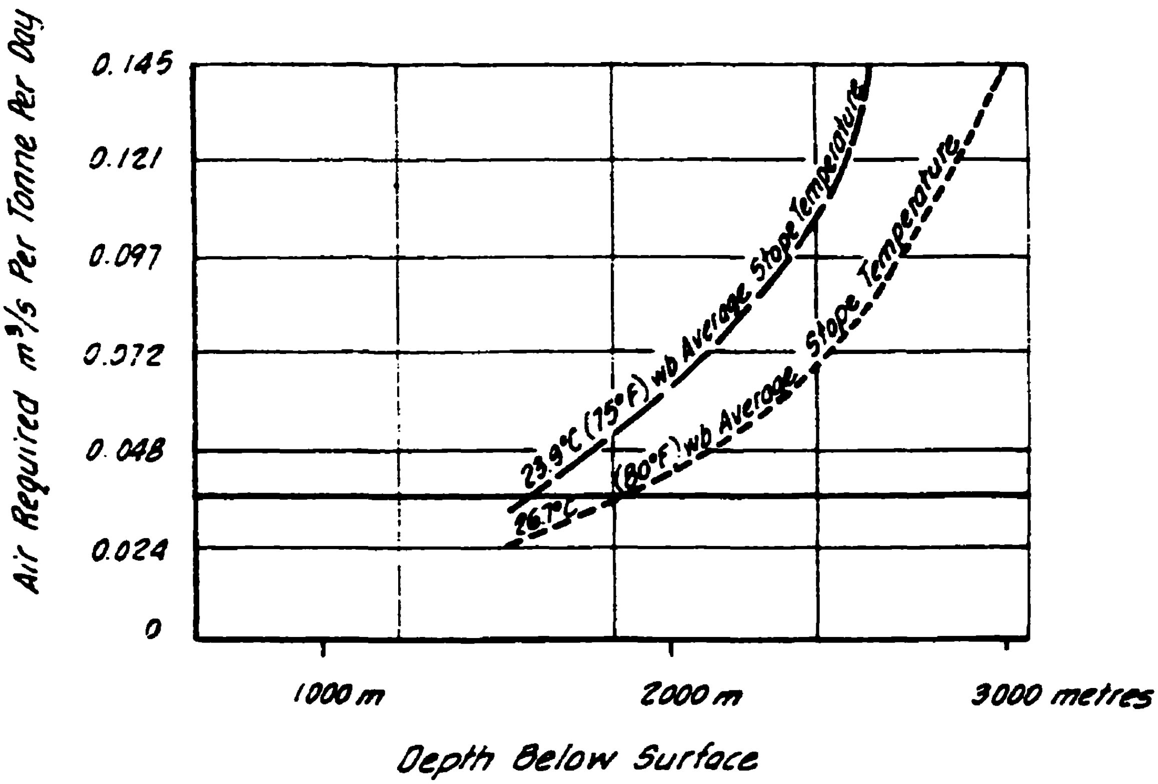

Figure 1.6: Air Requirements Versus Depth

Considering the existing facilities at Creighton, the costs of additional airways, power costs, and the costs for refrigeration, a ventilation system for a final capacity of some 680 m^3/s at a density of 1.26 kg/m^3 was recommended and approved.

- **Intake Airways**

The existing intake raise handles 448 m^3/s.

A transfer system was driven to connect the caved intake area to No. 9 shaft at 1500 level, and fans identical to the new 2600 level fans were installed near the shaft to deliver 200 m^3/s to it. Approximately 11.8 m^3/s is downcasting in No. 9 shaft from surface to 1500 level to keep the steel shaft sets in dry air.

The three main intakes will have capacity of some 708 m^3/s.

The main exhaust fans draw the air down No. 9 shaft to the lowest levels, through the workings, and up the exhaust raise to surface. They also act on the other two intake raises and assist the upper fans in these circuits.

- **Exhaust Airways**

The main expenditure involved the sinking of No. 11 shaft from surface to 6000 level. The shaft sinking commenced in 1977, and was put into operation as the exhaust airway by the end of 1980. It is 6.4 m in diameter with a continuous concrete lining. The shaft is located in the vicinity of the present main exhaust at 5000 level, and is connected to the mine workings at 3800 level and to the present exhaust system at 5000 and 6000 levels. The present exhaust below 6000 level is designed to carry all the air from below that point.

The changes to the Creighton lower mine ventilation system will permit increasing the volume of air below 7000 level to over 490 m^3/s, as compared to the present 95 m^3/s). The system will have low resistance.

- **Future Extensions**

A 3rd intake fresh air raise is being developed currently. It is a 6.1m Alimak raise from 800 level to 7000 level with the design airflow capacity of some 280 m3/s at 1.26 kg/m^3 , as shown in Figure 1.7.

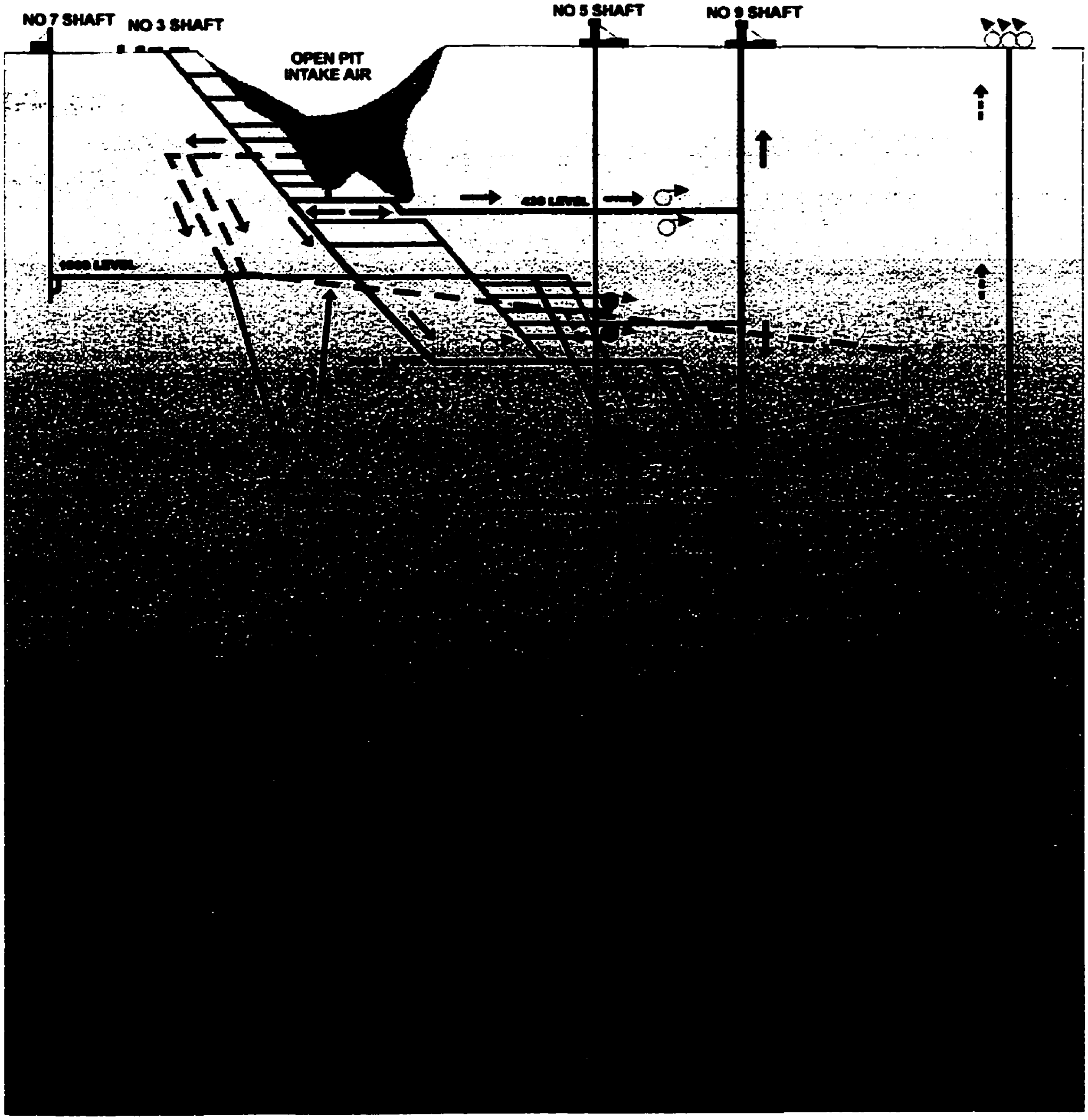

Figure 1.7: Creighton Mine Ventilation Schematic – Third Fresh Air System

- **Main Exhaust Fans**

The three main exhaust fans are located on surface at the collar of No. 11 shaft.

1.6 General

Downcast air for the majority of Canadian mines must be heated in winter. Because of this, the quantity of air necessary for the efficient removal of dust and gas from the production areas normally determines the minimum volume that must be circulated. However, as the workings are deepened, a point is reached where either increased volume circulation or air-cooling (or a combination of both) becomes necessary to ensure that temperature and humidity conditions remain within specified limits. At the Creighton mine, the desired upper limit of wet bulb temperature at the working place seems to be around 24°C (Coulter, Parsons, 1978) (Stachulak, 1979).

There is no legislative standard, however, some guideline is essential to permit design work and a limiting workplace temperature of 24°C wet bulb is assumed for Creighton. This is considered to be a conservative (but realistic) figure based on present knowledge of the mining in Canada and the stage of the art concerning research on heat stress limits carried out in South Africa, Figure 1.8. Furthermore, this is also supported by NIOSH pronouncement of heat stress standard and recommended exposure limits (NIOSH, 1986) refer to Section 4-5.

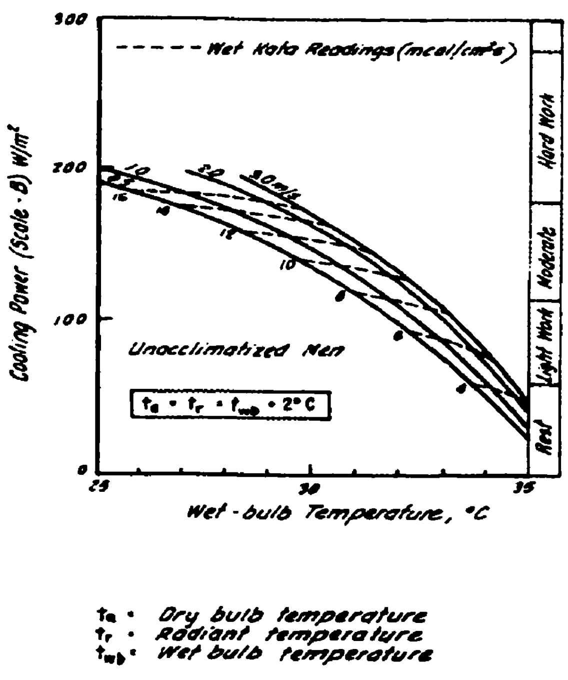

Figure 1.8: Cooling Power as a Function of Wet Bulb Temperature and Wind Speed – After Stewart 1979

The wet bulb temperature is a measure of the heat content of the air, and the principal contributors to the increase in temperature of the intake air are autocompression and wet airways. Typical figures for the increase in station wet bulb temperatures as a function of depth are given in Figure 1.9 for the Witwatersrand gold mines in South Africa. Thus, the theoretical limit of volume circulation is reached when the intake wet bulb temperature approaches the desired upper limit of temperature at the work place. This depth has been defined as the "critical depth" by Whillier and Ramsden (1975) and any additional heat load must be removed by air-cooling.

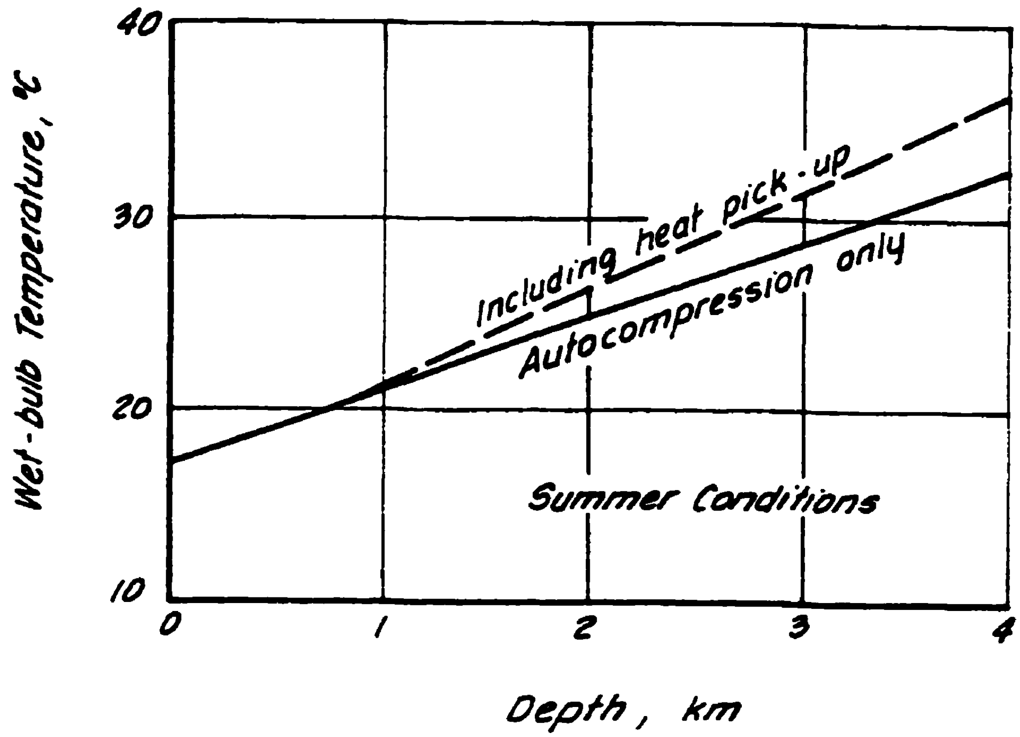

Figure 1.9: Station Wet Bulb Temperatures as a Function of Depth - After Whillier and Ramsden 1975

As far as can be determined, no major refrigeration plant (apart from small capacity units or spot coolers) has been installed in any Canadian mine, although one is now being contemplated for Falconbridge Onaping Depth Project. However, there are many examples of innovation involving techniques designed to reduce the wet bulb temperature at lower level stations. Thompson (1967) describes the work done at Bralorne Pioneer Mines Ltd. In British Columbia to reduce the increase in wet bulb temperature in the intake circuits. Heat problems at a depth of 1800-m (Virgin Rock Temperature 53°C) were alleviated by reticulating intake air through insulated ventilation ducts and drifts. Bischoff

(1947) describes the first use of ice stopes at Noranda's Horne mine in northwestern Quebec. High temperatures were encountered in the mine due to the heat given off by oxidizing sulphides in backfill. In the winters of 1944-45 and 1946-47, ice was made by spraying water into intake air traversing an empty stope located near the surface. This resulted in substantial heating of the air while forming the ice and cooling in the summer while melting the ice. The technique has since been used successfully at Inco's Stobie Mine. It is anticipated that the same principles will be applied in the future to defer the need for mechanical refrigeration at depth.

As far as Inco's Creighton Mine is concerned, refrigeration systems to date have been avoided by using the #3 Mine Open Pit as indicated previously. In the following chapters, we will examine the various factors that require consideration when mining at depth. We will review Creighton's heat exchanger capacity and critical depth.

Chapter 2

2.1 VENTILATION CONSIDERATIONS IN HARD ROCK MINES

2.1.1 Introduction

The primary role of an underground ventilation system is to provide adequate airflow in terms of quantity and quality, in order to maintain an environment, which can support human occupation and work. Ventilation is the "life blood of a mine".

If left uncontrolled, the mine environment can become quite hostile. The combination of heat, dust, blasting fumes and diesel fumes can be not only uncomfortable for workers, but also quite dangerous.

Improvements in ventilation have permitted the productivity of mines to be very much enhanced. Neither the first powered machines, nor the latest LHD equipment could have been employed underground without an adequate supply of air.

The way in which quality and quantity of ventilation is defined varies among different countries, depending on their mining tradition, the contaminants of highest concern, the perceived risk associated with those contaminants, and others, including political or social structure of the country. These basic requirements are incorporated into mining law in those countries that have such legislation.

The major method of controlling atmospheric conditions in a mine is by airflow. The design of an underground ventilation and environmental control system is a complex undertaking. The proposed mine ventilation system must be examined over the life of the mine, as a main ventilation system is costly and relatively inflexible.

The results of inadequate ventilation planning and system design are interruption in production, high expense of redesign, and poor environmental conditions reflected in health and safety of the workplace.

2.2 General network configuration

A standard ventilation system is used for the majority of mining operations. This is basically a fresh air pressure system, in that all, or the greater part of, the mining areas and the approaches to them, are subject to pressure from the intake fans.

It has been Inco's practice since the early 1950's to excavate separate airways to handle all main airflow, both from and to surface, thus leaving the operating shafts basically neutral. One of the advantages of this configuration is that it creates a positive upcast of fresh air to the surface, which serves to prevent freezing the shaft during winter months. This type of configuration also leaves the mine accessible in the event of an underground fire (Rutherford, 1955).

The main intake is located between the operating shaft and the orebody, so that the fresh air can be maintained to either area in the event of a fire emergency Figure 2.1.

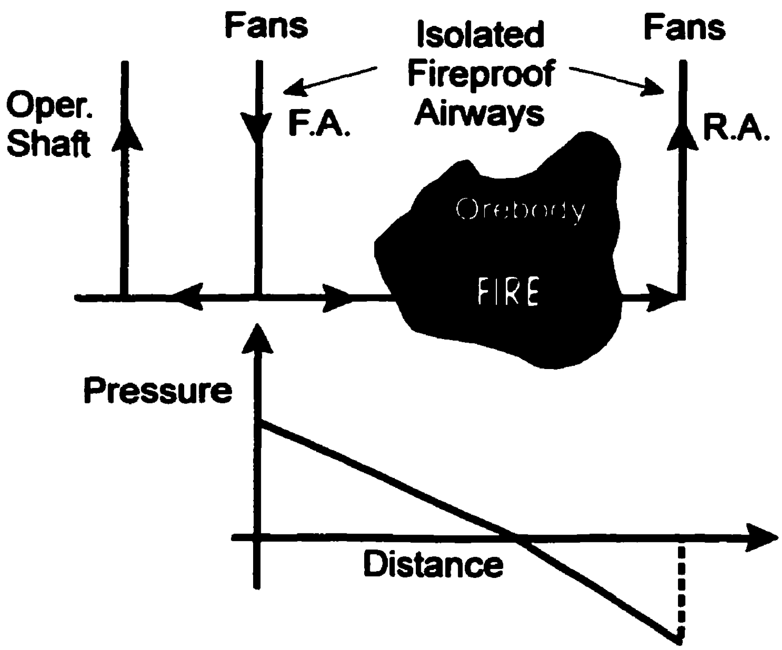

Figure 2.1: Simplified Pressure Diagram

All permanent airways are located in the footwall rock, outside of the area, which might be affected by mining activity. All intake and return airways are fireproof. In the design of ventilation networks, series configurations are avoided if possible to prevent each subsequent working area to be supplied with contaminated air from upstream work areas.

The sizes of the main airways are determined on the basis of total cost; capital cost plus the present value of the power cost for handling the designed airflow Figure 2.2. Trailing hose and barometric pressure surveys are carried out. Computer ventilation network analyses have been the standard practice for all major ventilation planning purposes since the late 1970's.

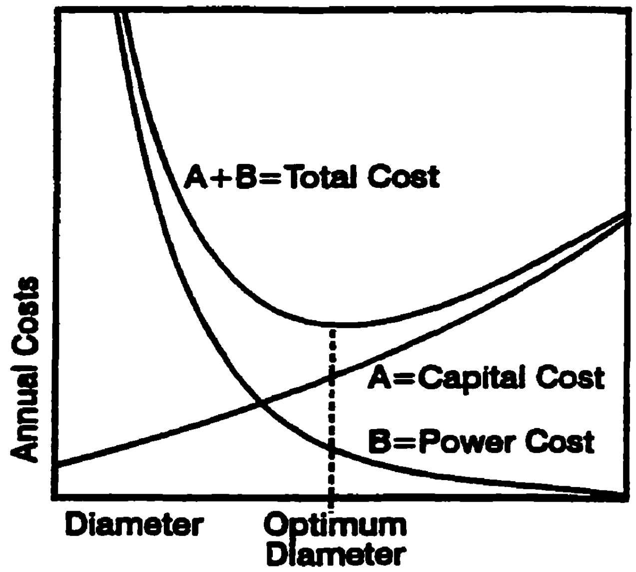

Figure2.2: Variation of Cost with Airway Diameter

Considering the overall picture of present day mining operations, the expenditure required to provide a proper system of mine ventilation should be regarded as a sound investment that will pay back in the form of increased mining efficiency, improved safety, health, and morale of miners.

2.3 Airflow requirements

One of the first questions to be resolved for the ventilation of a development heading, the initial development of a mine, the expansion of a mine, or the development of a new mine, is the quantity of air required. Quantities have to be such as to dilute contaminants to low levels within time limits acceptable to management and to keep contaminants within acceptable limits.

For the expansion or deepening of an existing mine, the environmental conditions of current operations serve as good estimates in determining suitable airflow requirements for new areas. However, the task is more involved when a new mine is to be developed away from existing mines, when new or different mining methods are to be employed, or where the rock and ore constituents are appreciably different.

The daily flow rates (m^3/s) per tonne vary depending on the geographical location and type of mining employed. For example, the large block-caving mines of the southwest United States reach up to more than 0.094 m^3/s per ton per day for m^3/s highly siliceous gold ores. High-grade uranium mines use 0.378 m^3/s per ton per day and more, to control radiation. The deep gold mines in South Africa use an approximate limit of 0.118 m^3/s per ton per day, and then resort to refrigeration.

Determining airflow requirements for new hard rock mine settings require that long-term mine plans be established which allow for the following factors to be considered. A proper ventilation design must consider the number and type of workplaces, production schedules, virgin rock temperature, heat load generation, number of diesel equipment, and respect legal requirements.

2.3.1 Diesel fume contamination

The generation of diesel fumes by mechanized equipment requires that sufficient quantities of air be able to dilute pollutants to acceptable levels. In Ontario, the Ontario Occupational and Safety Act (Reg. 854) requires that at least 0.047 m^3/s for each horsepower of diesel powered equipment operating in the workplace. Inco practice is to design 0.059 m^3/s per each horsepower of diesel working in the area. Not long ago, 4.7 m^3/s was adequate for most development headings, with a general figure of 0.024 m^3/s per square foot of face area being used. Today, Inco is driving single drifts requiring 37.7 m^3/s to meet legal requirements for the diesels being used in the heading. With mechanized mines, the main shaft crosscuts and tail end drifts must be ventilated continuously, although diesels are in them only occasionally.

Extra allowances are required for garages, ramps, crushers, underground hoists, leakages to shafts, major ore haulages by diesels, and for long developments or exploration drifts.

2.3.2 Blast fume contamination

Blasting can significantly impact air quality if diligent practices are not employed such as the use of water blasts and sprays. Another effective way of dealing with blast fume contamination is the proper scheduling of blasts. Inco for example, does limited blasting of misfired holes and secondary blasting during the shift and confines stope and development blasting to the end of the shift, or in the case of large blasts, to the weekend. Any significant move away from this practice would result in a large increase in air requirements to maintain acceptable working conditions.

2.3.3 Dust contamination

Good dust and ventilation control practices at ore and rock passes, crushers, conveyors, transfer points and shaft loading stations reduce ventilation requirements to achieve a given set of environmental conditions. In general dust concentration decreases by dilution as airflow increases. However, large dust particle concentration increases as air velocity rises, due to enhanced pick-up and delayed settling. Therefore an optimum velocity range exists between 1.5 and 3.6 m/s as seen in Figure 2.3. This graph indicates that a minimum total dust concentration is produced at a velocity of about 2 m/s.

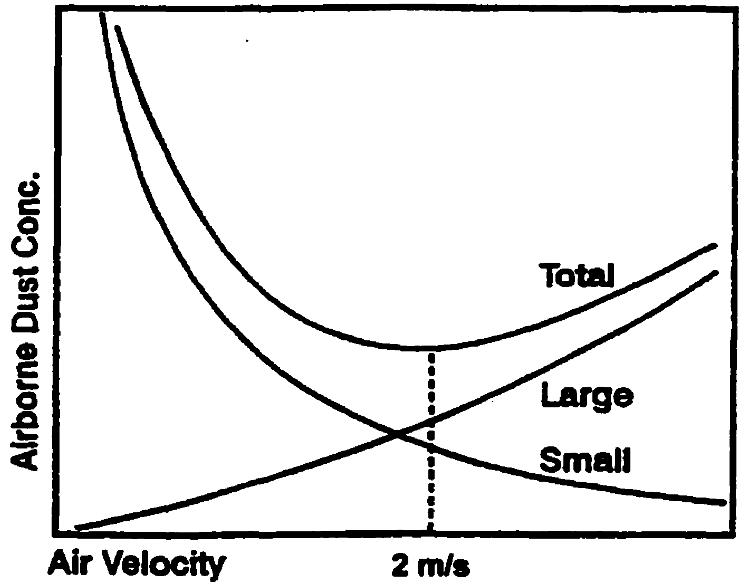

Figure2.3: Variation of Dust Concentration with Air Velocity by Gruszka et al (After Anon, 1974)

The total dust, however, encompasses a fairly broad-based curve and air velocity in the range of 1 m/s are acceptable. The air velocities over 4 m/s are not so much a health hazard, but rather a physical discomfort of large dust particles striking the skin.

2.4 Heat Load Prediction and Airflow Requirements

In deep Inco mines such as Creighton, heat is a dominant environmental problem, and the air quantity depends on the heat removal capacity of the air and becomes a special consideration involving the economics of whether to increase air flows above a certain point, or to resort to refrigeration. The ventilating air that circulates through a mine is heated by various sources. The three principal heat sources in mines are the conversion of potential energy to thermal energy as air "falls" through downcasting airway (auto-compression), geothermal heat from strata, and mechanical and electrical equipment (booster and orebody fans, diesel and electric mobile equipment, electric load centers, pumps). Additional heat sources are explosives, body metabolism and cement in sandfill.

Predicting heat transfer from the large area of rock to air in a mine, as a whole, is still largely a difficult task at best. How much heat will flow from rock to air in a mine depends upon many factors, such as whether the mine is wet or dry, the cooling time of the rock, the method of stoping, production rate, variation in surface temperature, and many other factors.

Two courses of action are possible. One is to claim that the problem is so complicated that it is impossible to calculate anything that will be meaningful. This would be the negative approach of a defeatist.

The other approach is to ignore the complications until the preliminary calculations have been done, and then ask the question: What percentage effect would the complications likely to have on the answer obtained from the idealized calculations? It is quite amazing how often answers to extremely complicated problems can be obtained in this way, by using greatly simplified models of the real situation for the purpose of calculation (Whillier, 1982).

In the late 1970's, Inco had adopted an empirical approach, based on a combination of mining depth and production rates among others. Basically, in a mine, the rate of heat flow is approximately proportional to the difference between virgin rock temperature and air wet bulb temperature, multiplied by a factor which depends upon the many variables which have been mentioned previously.

This factor is obtained either from experience, or from actual measurements. When this factor is known, it is a comparatively easy matter to predict future heat flow conditions. During the last several years, developments in computing power and mine climate simulation software enabled the complex mathematics and heat parametric interactions to be handled relatively easy. These mathematical models allow heat flow predictions to be quantified in detail, provided that the more influential items of input data are adequately specified. Both the empirical and mathematical approaches have their strengths and weaknesses (McPherson, 1993).

2.4.1 Empirical Method

Strengths –

- Simple and easy to apply
- Requires no computational aid
- Gives reliable results for conditions, mining methods and equipment identical to those in which the empirical constant (Heat Load Factor) was established.

Weaknesses –

- Provides estimates of overall heat additions but gives no detailed predictions of climatic quality
- May produce large errors if applied to conditions that deviate from those in which the Heat Load Factor was evaluated

2.4.2 Mathematical Methods

Strengths –

- Gives detailed predictions of all psychrometric and heat stress parameters, as well as heat loads, for any specified routes in the mine
- Very rapid desk-top computation, given the required computer facilities and software
- Gives reliable results provided that correlation tests have been carried out

Weaknesses –

- Requires the acquisition of a significant variety of data
- Some items of data (e.g. internal wetness of airways, in-situ values of rock thermal parameters) may be difficult to ascertain with good accuracy
- Correlation tests are required to fine-tune data that contain uncertainties and before accurate predictions can be expected

The form of the empirical equation used in South Africa and at Inco is not entirely rational. The flow of strata heat into mine workings is, theoretically proportional to the difference between VRT and dry bulb (rather than wet bulb) temperature.

$$\text{Heat Flow} = h \times A\ (VRT - t_d)$$

Where h = overall heat transfer coefficient (W/(m°C)
A = internal surface area of opening (m^2)
VRT = Virgin Rock Temperature (°C)
and t_d = local dry bulb temperature (°C)

In an empirical equation, with a Heat Load Factor determined from actual measurements, this will have little impact on the applicability of the relationship. Furthermore, there is a definite practical advantage to employing the wet bulb temperature ie: the latter behaves in a more stable manner than dry bulb temperature. While wet bulb temperature depends only upon heat additions and changes in elevation, the dry bulb temperature depends, additionally, on evaporation and condensation, and these can vary locally and very considerably along any underground air route.

The increase in the value of air heat content over any given zone and time depends upon:

- Length, cross-sectional area and perimeter of the airways
- Levels of both the inlet and exit ends of the airways
- Roughness of the internal surfaces of airways
- Ages of all openings
- Wetness of surfaces
- Airflow distribution and the psychrometric condition** of the air at intake point(s).
- In-situ values of rock thermal conductivity and diffusivity
- Virgin Rock Temperatures** and geothermal gradient
- Mine production** and, hence, the rate at which new surfaces are exposed
- Type, power and distribution of all mechanized equipment.
- Type and consumption** of explosives
- Rates and temperatures of fluid (water and gas) emissions from the strata
- Methods and rates of transporting water along airways
- Methods and rates** of transporting fragmented ore along airways and through orepasses
- Oxidation processes
- Number and deployment of the labour force
- Local geothermal or radioactive phenomena

Only those factors marked by ** are in the empirical equation. All other factors are accounted for within the Heat Load Factor. It is for this reason that the HLF must be expected to vary between the tests that are carried out to evaluate it. Furthermore, a significant variation in any one of the contributing factors can cause a substantial change in the HLF (a variation of some 30 percent has been observed in values measured over the past 20 years at Creighton Mine). Therefore, it is advisable to continue the practice of re-evaluating the HLF throughout the productive life of the mine.

Currently at Inco, both methodologies, i.e. empirical method and mathematical modeling, are used for predicting ventilation requirements, and securing future adequate environmental conditions at several of our mines.

The following examples provide the assessment of airflow requirements to dissipate heat load in trackless areas, based on a design cut-off wet bulb temperature of 23.9°C in the workplace.

Case 1

The virgin rock temperature is 25.6°C at 1158 m below surface. The surface air temperature is 16.8°C/21.7°C. The four heat sources have been identified in the 3800 level working stope, namely: mobile diesel equipment – 275 HP, explosives – 1688 kJ/min or 28 kW, broken rock – 10,340 kJ/min or 172 kW, and ventilation fans – 30 HP.

Results

21.2 m^3/s is required to dissipate heat load and to maintain a temperature of 23.9°C wet bulb in this workplace, and this compares with 13.0 m^3/s required to meet legal requirements.

Case 2

Creighton – Heat flow and airflow prediction at 2256 m below surface. Fresh air station temperature = 18.0/26.1°C, VRT = 44.4°C.

	Required	(kW)					Airflow
Equipment HP	Airflow m^3/s	Rock Broken	Rock Wall	Equipment	Fans	Total kW / %	Legal Req. m^3/s
2 Diesel Scoop 2 x 277 1 Electric Truck 1 X 350	103.8	(200) 11%	(136) 8%	(1078) 62%	(320) 18%	(1732) 100%	26.0 *
2 Diesel Scoop 2 X 277 1 Electric Truck 1 x 700	115.6	(200) 10%	(136) 7%	(1276) 66%	(320) 17%	(1932) 100%	26.0 *

* Airflow requirement for electric truck is not included.

Note: A diesel LHD produces about three times as much heat as an electric LHD of the same power rating, i.e. almost all of the caloric value of the diesel fuel consumed will appear in the form of heat.

Furthermore, the combustion of unit of diesel fuel will produce about 1.1 units of water liquid equivalent. Unfortunately, this may be multiplied, by several times, by evaporation from engine cooling systems and wet exhaust scrubbers. Estimates by the Chamber of Mines Research Organization, South Africa, of the total moisture released during the operation of diesel equipment, varies between 3 and 10 liters of water per liter of fuel.

2.5 Conclusion

As can be seen, the quantity of ventilation in a mine depends on several factors.

1. Consideration needs to be given to the nature and sources of the pollutants which must be diluted.

   - Flow rates and cooling required to dissipate heat in trackless areas must be calculated based on the cumulative heat loads of the equipment in that area.

   - Allowances must be made for other ventilation requirements such as:

Ramps
Workshops
Crusher Stations
Belt Transfer Points
Loading Pockets
Ore Pass Drifts
Diesel Repair Garages
Diesel Fueling Stations
Refuge Stations
Car Repair Stations
Pump rooms
Switch rooms
Battery Charging Stations
Diamond Drill Stations
Raise Bore Stations
Hoists and Electric Set

2. Environmental conditions must meet legislative requirements.

   Even if the criteria outlined above are met, it is still important that the air velocity is sufficient to remove fumes from the area where trackless equipment is working. Load haul dump (LHD) vehicles drive at about 2 m/s and if an LHD turns around at the end of its run to the ore pass it could return into the workplace before the air has cleared. This is especially true in large airways typical of trackless areas where the design airflow dilution factors can lead to rather sluggish velocities.

   In a long development headings, sluggish velocities could result in gradually increasing concentrations of pollutants with each cycle of the LHD. As a guide it is suggested (LeRoux's, 1990) that air velocities should not be lower than 0,5 m/s. In long developments this may need to be increased to ensure a minimum number of air changes per hour to ensure proper clearance of fumes.

Chapter 3

3.1 MINE ENVIRONMENT AND ITS CONTROL IN DEEP CANADIAN MINES

3.1.1 Introduction

Canadian mines use large quantities of heated fresh air during winter months. The average annual cost of heated air can be as high as $1 million for individual mines (Hall, 1985).

In Sudbury, Ontario, the mean atmospheric temperature is 3.4°C dry-bulb, and 1.6°C wet-bulb. In winter the average temperature over the 150-day heating period is –8.3°C DB, and –8.9°C WB, and minimum temperatures are in the order of –32°C.

The mine fresh air heaters are sized to raise the temperature of the air through some 35.5°C. Direct-fired burners, using propane or natural gas are most commonly used. The burners are rated at 2028 kW per 47.2 m^3/s of ventilating air. The average heating required over the 150 day period is 621 kW for a temperature rise of 10°C per 47.2 m^3/s. However, when the fresh air is supplied to the mine via the main hoisting shaft, the air is usually heated to 4.4°C and the average heating required is increased by 28 percent.

With the steady rise in the cost of energy, more mines are investigating alternative sources of energy to heat and cool the mine fresh air. Various innovative systems were pioneered at Inco (Rutherford, 1960). Examples of such systems and others, past and current practices, are discussed in this chapter.

3.2 Stobie Mine Air Conditioning

In nature, seasonal changes in temperature are modified by the evaporation and condensation of water and the melting and freezing of ice.

Inco's Stobie Mine "air conditioning" is a unique world class example of implementation of one of these methods.

The air conditioning that is being used at Stobie is unique in that it accomplishes both heating during winter, and cooling during summer by one process. This method employs the freezing and melting of ice to adjust year-round temperatures. It was completed in 1955, and has been in operation since then (Stachulak, 1991).

The fresh air from surface is circulated through two large open stopes in series before reaching the main intake airway at the 300 level (Figure 3.1).

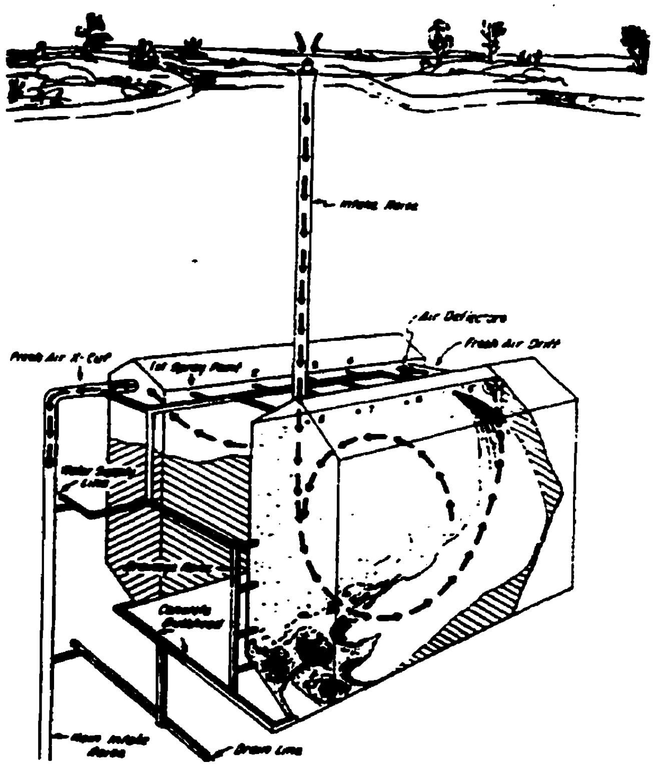

Figure 3.1: Ice Stopes at Stobie Mine (After Rutherford 1960)

During the winter, water is sprayed at the top of the stopes to form ice in the cold air. The ice builds up in the two stopes. The heat, liberated by the formation of ice, 335 kJ/kg, is the main source of heat transferred to the air. About 37.2 kJ/kg is obtained from cooling the water, and there is an appreciable transfer of heat to the air from the large area of rock surface exposed, especially in the early winter. To a lesser extent, there is also heat transfer to the air approaching the ice stopes from the fan and the adiabatic compression of the air.

The two stopes were mined side-by-side by the blasthole method above the 500 level, in a low-grade section of the ore zone between the two main mining areas. Each stope is 24.4 m side, 61 m long and 61 m high. A pillar 21.3 m wide separates the two stopes.

The surface fan, a 5.0 m axial flow unit, mounted vertically, delivers the fresh air through a vertical airway 6.1 m in diameter and 91.5 m long, to the top of one stope at the footwall end. The air circulates through this stope and transfers to the second stope via a large drift through the top of the pillar at the hangingwall end.

Fixed vanes at the discharge of this drift deflect the air down into the second stope. A 5.5-m by 5.5-m crosscut connects the far end of that stope to the top of the main intake airway located in the footwall. With all of the ventilation connections at the top of the stopes, the build-up of ice in the stopes does not restrict the flow of air.

At the same elevation, three small spray drifts are connected to each stope from a service crosscut in the center of the pillar. In each spray drift, a 50-mm water line, with five spray nozzles, is extended 13.7 m out from the pillar across the width of the stope. The pipe is supported by a cable through a short diamond drill hole that intersects the arched back of the stope. A strainer is maintained in each 50-mm spray line.

Mine water from the underground pumping system is supplied to the spray through a 150-mm pipeline from the shaft pump column. The 91.4 m head gives an operating pressure of 758 kPa in the service crosscut and 552 to 621 kPa near the sprays.

The water temperature is 7.8°C. The main pumps are arranged to keep one pump operating continuously during the winter.

Ice formation is very efficient in colder air, from –28.9°C to –6.7°C, the main problem being to raise the temperature to –1.1°C. The most efficient and practical method found to date is to use flat sprays, 19 mm in size, with a 9.5 mm orifice, and to introduce compressed air at 586 kPa into the waterline just ahead of the sprays.

The water particles must be finely divided so that they will quickly turn to ice in the air. Spray orifices must not be too small, otherwise excessive blockage results despite strainers in the water supply lines.

With five sprays on the pipe, each spray line handles 5 ℓ/s of water mixed with 0.019 m^3/s of compressed air. A maximum of six lines are operated during colder periods, handling a total of 30.3 ℓ/s of water. The average volume of water sprayed is in the order of 18.9 ℓ/s. The number of spray lines operating is controlled manually. When a line is turned off, compressed air is bypassed around the valve to blow out the pipe and sprays, removing any water that would otherwise freeze and block the pipe.

The temperature of air leaving the ice stopes during the winter varies from –2.2°C to –0.5°C. During the sub-zero periods, the maximum heat supplied amounts to 11,136 kW. The average heating is close to 3810 kW.

The total quantity of ice formed during the winter is estimated at 145,000 tons, occupying about 75 percent of the storage capacity of the two stopes.

The yearly operating costs, including labour, supplies, and the cost of the extra pumping and compressed air for the ice stopes totals close to $30,000. For an equivalent natural gas air heater, the fuel cost alone would be $230,000 per year.

During the summer, the ice is melted and the air is cooled. While this "air conditioning" is welcome, there is no heat problem at Stobie. The summer melting power is almost four times the winter freezing power, but the efficiency of melting is very low.

During warmer periods, several lines of sprays are operated at the top of ice stopes to increase the rate of ice melting, so that the stopes will be clear of ice before the winter. No problem has been encountered with water drainage from the stopes to the main sump.

The refrigeration provided by these stopes in the summer months does not provide equivalent savings to the heating capacity during the winter months simply because there is not a significant amount of heat generated at the Stobie Mine. However, if a similar system were used in a mine that required air refrigeration during the summer, then the merits of bulk cooling of the fresh air in the ice stopes during the warmer months should be considered. It would require separate storage of the water from the melting ice and a pump installation to supply some of this cold water to the sprays during the summer.

3.3 Creighton Natural Heat Exchange Areas

Creighton is particularly recognized in the world of mine environmental engineering because of its primary heat exchange method utilizing the heat storage capacity of fragmented rock to provide low cost air conditioning throughout the year.

Mining of the Creighton orebody began with an open pit in 1901. Subsequently, several decades later, when this progressed into underground operations, it was found that the summer peaks and winter troughs of surface air temperature could be moderated drawing intake air through the mass of broken rock in and below the old open pit, via box-holes and the slusher drifts (Figure 3.2), into the main air intake system for the mine. Over the years, this has been further developed by collecting the air from the old slusher drifts into a "gathering area" of airways at the 800 level. The result is that intake air, at this level, is provided at about 3°C throughout the year.

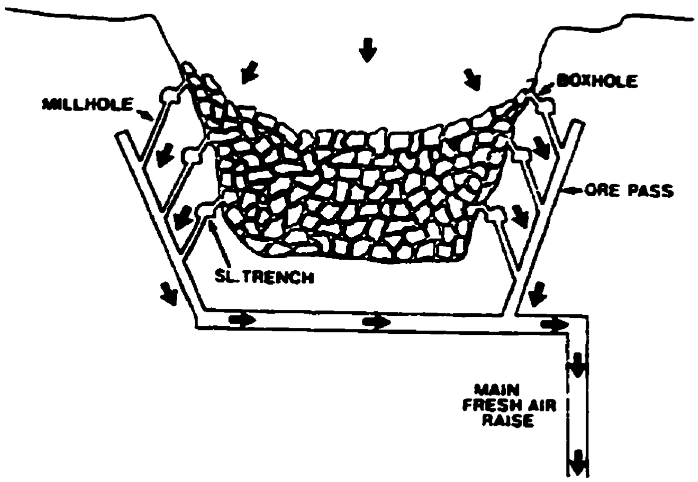

Figure 3.2: Schematic of Creighton Natural Heat Exchanger

Over time, the demand for greater quantities of cool air has increased due to greater depths of workings, increased tonnages, and enhanced utilization of diesel equipment. That increased demand for air conditioning has been met, to the present, by larger quantities of air being drawn through the natural heat exchange area, coupled with enlargement of the 800 level gathering airway in order to use additional volume of the fragmented rock.

By using this "intake" heat exchange area, cool fresh air is supplied at depth at a nearly constant temperature, and a favorable natural ventilation pressure of is developed.

The capital cost of the alternative refrigeration system for the mining at depth would be in the order of $20 million.

3.4 Heat Transfer From Exhaust Mine Air To Cold Intake Fresh Air

Sound engineering practice dictates that recovery of energy from the exhaust mine air should be economically attainable. The examination of the worldwide mining industry would indicate that this source of "low grade heat" is of potential interest to the mines located in northern regions. The effective utilization of this heat energy is, however, a combination of high rock temperature (associated with mining depth) and a cold surface temperature. In Canada, the relatively shallow mining depth and moderate virgin rock temperature do not particularly make heat recovery a viable proposition. As a result, there has been little motivation to implement heat reclamation systems in Canadian mines. The 1980 exhaust shafts heat recovery investigation of 97 underground Canadian mines concludes the following, (Freyman, 1980): "The theoretically reclaimable energy from mine exhaust air, on the basis of existing technology is estimated to be in the order of $20 million per annum. However, the economically recoverable energy is equivalent to a potential saving of $2 million per year. The low volumes, remote locations of exhaust shafts, temperature of the return air below 10 degrees C, and seasonal demand are the main hindrances to economic recovery."

This is well supported by the fact that the first Canadian heat reclamation system (Creighton Mine of Inco Limited, which was in operation from 1955 to 1970) ceased to operate due to constant difficulties to maintain proper operating conditions.

At the Creighton Mine, a heat exchange system was installed on surface (Rutherford, 1960) between the main return from the lower mine, exhausting 123 m^3/s to surface at 16.1°C saturated, and a fresh air fan supplying 59 m^3/s to caving mining operations in the upper mining area.

The two airways were 38 m apart. The heat exchanger consisted of two units, one at each airway, and water was used as the heat exchange medium. At the return airway, the water was heated by direct contact with the exhaust air in a wooden tower adjoining the stack on the airway. At the fresh airway, the air was heated as it was drawn through a large coil that occupied one wall of a house over the vertical fan.

The water was circulated between the two units through insulated pipelines by a 22.4 kW pump handling 88 1/s. The pump supplied the warmed water to the coil and returned the cooled water to the top of the tower. Except for the resistance of the two units to airflow, equivalent to 30 kW, the power to the pump was the only energy required (Figure 3.3).

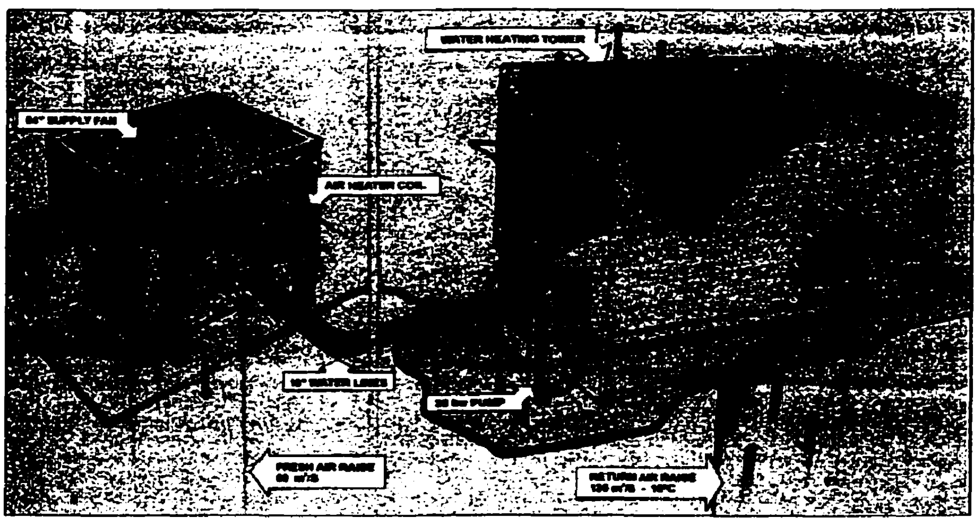

Figure3.3: Schematic of Exhaust Mine Heat Transfer to Cold Intake Air (After Rutherford, 1960)

The rate of heat exchange varied with the outside air temperature, as indicated in Table 3.1.

Temperature – °C				Heat Exchanged kW
Air		Water		
Outside	Heated	Supply	Return	
- 3.9	8.3	14.4	11.9	938
- 17.8	1.7	12.8	8.9	1495
- 28.9	-2.8	11.7	6.4	1993

Table 3.1: Performance of Heat Exchange Equipment – Creighton Mine

The operation of this heat exchanger required considerable attention during severely cold periods. The initial cost of the installation was bout 80 percent of that for an equivalent oil-fired heater.

3.5 Heat Energy from Air Compressors

The application of this "waste" energy to heat mine air was put into operation at the following mines:

3.5.1 Kidd Creek Mine

At the Kidd Creek Mine near Timmins, a total of 406 m^3/s of air was supplied to the mine (Schmidt, 1978).

At the full volume of the air compressors, 16.5 m^3/s at 760 kPa, a total of 5860 kW was removed by the compressor cooling water.

In 1976, a heat recovery system was installed to use this heat to warm the fresh air supplied to the mine in the winter. The heat in the compressor water is transferred to a 50 percent glycol-water mixture in a heat exchanger, and the glycol mixture is pumped through two coil banks in the fresh air tunnel. The glycol coils provide the primary air heating; additional heat is supplied as required by two 6155 kW propane heaters. The estimated saving in fuel costs during the 1979/1980 winter were approximately $215,000, which is about equal to the total cost of the heat recovery system.

3.5.2 Strathcona Mine

At the Strathcona Mine of Falconbridge, a similar system was used to transfer the 2930 kW produced at the full volume of the air compressors, 10.76 m^3/s, to warm the fresh air (McCallum, 1969).

3.5.3 Lockerby Mine

At the Lockerby Mine operated by Falconbridge, a heat recovery system is used to transfer up to 1465 kW produced at the full volume of the air compressors, 5.2 m^3/s, to warm the fresh air in winter.

3.6 Conclusion

The harsh Canadian winters necessitate heating of large quantity of fresh air supplied to underground mines. The cost of heating air is expensive as most operations resort to propane burners. However energy saving alternatives have been developed at Inco such as the use of latent heat generated from the formation of ice. The use of ice stopes for regulating air generates annual savings of approximately $200,000 for the Stobie Mine.

The natural heat exchanger consisting of fragmented rock provides benefits throughout the year for the Creighton Mine whereby air is warmed in the winter months and cooled in the summer months. This natural heat exchanger has effectively replaced the alternative of a $20 million refrigeration system.

The implementation of heat exchangers at exhaust air locations has been deemed uneconomical for most Canadian mines as a result of remote exhaust shaft locations, seasonal demand, and temperature of return air below 10 degrees C. However, heat recovery from air compressors has been economical for some Canadian operations.

Chapter 4

HEAT STRESS

4.1 Introduction

A major challenge associated with mining at depth is the effect of heat on humans. Hard work produces metabolic heat at a rate of 0.5 kW and hard working humans must lose heat to the surroundings at this rate if the body temperature is not to rise and cause heat stress.

It is well documented that heat stress causes discomfort, decreases productivity, increases accident rates and abnormal physiological strain on humans. In its most acute form, excessive heat stress initiates a total collapse of the body's temperature regulation system and results in heat stroke.

Hot mines have posed major problems for operators with respect to maintenance of adequate working conditions. The subject of heat stress has been well researched over the last 30 years and Stewart (1979) has summarized the latest thinking on this subject.

4.2 Environmental Conditions

A permissible environmental condition is not only dependent upon the wet bulb air temperature, but also on the air velocity. Numerous tests have been carried out to determine the heat production of man and the environmental conditions required in order to be able to dispose of the heat produced.

The heat production of a moderately hard working person is about 250 W/m^2 of body surface, and in order to maintain a heat balance between heat production and heat-dissipation, the following environmental conditions are required (Barenbrug, 1971).

Wetbulb Temperature °C	Air Velocity (m/s)
21.4	0.1
25.1	0.2
27.5	0.3
28.6	0.4
29.4	0.5
30.6	0.8
31.3	1.0

Note the great influence of air movement at low temperatures. This is one of the reasons why in the South African gold mines, wet bulb temperatures in the range of 29°C are acceptable, because air velocities in stopes are usually of the order of 0.5 to 1.0 m/s.

Most continental countries in Europe prescribe a minimum velocity of 0.25 m/s and a maximum air wet bulb temperature of 26.7°C" (Barenbrug, 1971).

4.3 Physical Effects

Humans are homeothermic animals who have a highly developed ability to maintain heat balance within very narrow limits around 35.6°C. As previously stated it is very important for the body to lose heat to the surroundings at an acceptable rate if one is to prevent heat stress.

Heat is removed from the body in three ways (Ramsden, 1975):

1. By radiation and convection from the skin
2. By evaporation of sweat (perspiration)
3. By water evaporation in the lungs (respiration)

The above mechanisms come into play as temperature conditions increase. This is known as thermo-regulation. Firstly, radiation and convection will emit the excess heat. If further heat is encountered, vasodilatation will occur and "extra" blood will be passed to the skin of the extremities in order to increase the rate of heat transfer from body to air.

Again, if further temperature rises are incurred, the body will begin to perspire in an attempt to rid excess heat by evaporation. The latter stage is the most important in terms of body cooling while working in deep mines. Therefore, the rate of body cooling is directly proportional to the rate of sweat evaporation and depends on an acceptable balance between metabolic heat generation and cooling of the body. If the body is to maintain thermal equilibrium then metabolic heat has to be removed to the surrounding at a rate equal to that at which it is generated.

Physiological heat exchange between body and ambient surroundings takes place by a combination of heat transfer processes, namely: respiration, convection, radiation and evaporation.

The physical effects of heat stress on humans are well researched and analyzed, and it is well established that excessive heat stress will cause body disorders.

The following are the main body disorders in diminishing order of severity.

- Heat stroke
- Circulatory deficiency heat exhaustion
- Salt deficiency heat exhaustion, including heat cramps
- Heat rash (prickly heat)

The most serious of the heat disorders, heat stroke may occur when the body core temperature rises above 41°C. The major symptoms of impending heat stroke are the following:

- Perspiration ceases, the skin remains hot, but dry, and may adopt mottled or cyanotic condition
- Severe disorientation, a glassy stare and irrational or quarrelsome behavior, acts as if insane
- Loss of control of bodily functions
- Shivering and other uncontrolled muscular contraction may occur

In the absence of immediate, skilled treatment the patient can suffer permanent brain damage. Kidney, liver and circulatory disorders may also develop, leading to death in some cases. Immediate cooling of the patient using water and compressed air is critical. Medical assistance should be sought to ensure correct treatment and temperature control.

4.3.1 Circulatory Deficiency Heat Exhaustion

This disorder is the result of blood pooling in the veins of the lower body shortly after the discontinuation of work. This produces an inadequate return of venous blood to the heart, creating a blood supply shortage to the brain. The patient needs to be relocated to a cool area and given water or salted fluid.

4.3.2 Salt Deficiency Heat Exhaustion

This disorder is caused by excessive loss of salt through sweating and urination. Heat cramps occur due to body salt depletion in the body. Treatment consists of correcting of dehydration by administering salted fluids orally. If the patient is unconscious, then normal saline solution is provided intravenously.

4.3.3 Heat Rash

Heat rash, also known as prickly heat, is a form of damage to the sweat gland from unrelieved periods of constant perspiration resulting in their inflammation and the formation of tiny red blisters in the affected area. Severe cases may require the worker to cease working underground. Heat rash can be prevented by frequent bathing and a cool environment.

4.3.4 Managing Heat Stress

The principle aspects of utmost importance in protecting a workforce against unacceptable heat illnesses is the design, engineering, and control of the ventilation and air quality system of the mine which provides a safe climatic environment.

A workforce that has to work in hot environments should be trained to understand the symptoms of heat disorders and to employ rational work habits. This would include appropriate, but not surplus, rest periods, and ample consumption of cool drinking water. A sound dietary balance should be encouraged, that contains sufficient but, not excessive salt. Vitamin C supplements are also considered to improve heat tolerance (Wyndhatt, 1974).

In conclusion, heat strain, heat exhaustion and heat stroke can be avoided by reducing working temperature to below 26°C WB and by providing training and safety programs for workers required to work in hot environments. The reduction in the incidence of heat stroke cases in the South African gold mines is indicated in Figure 4.1 (Wyndhatt, 1974).

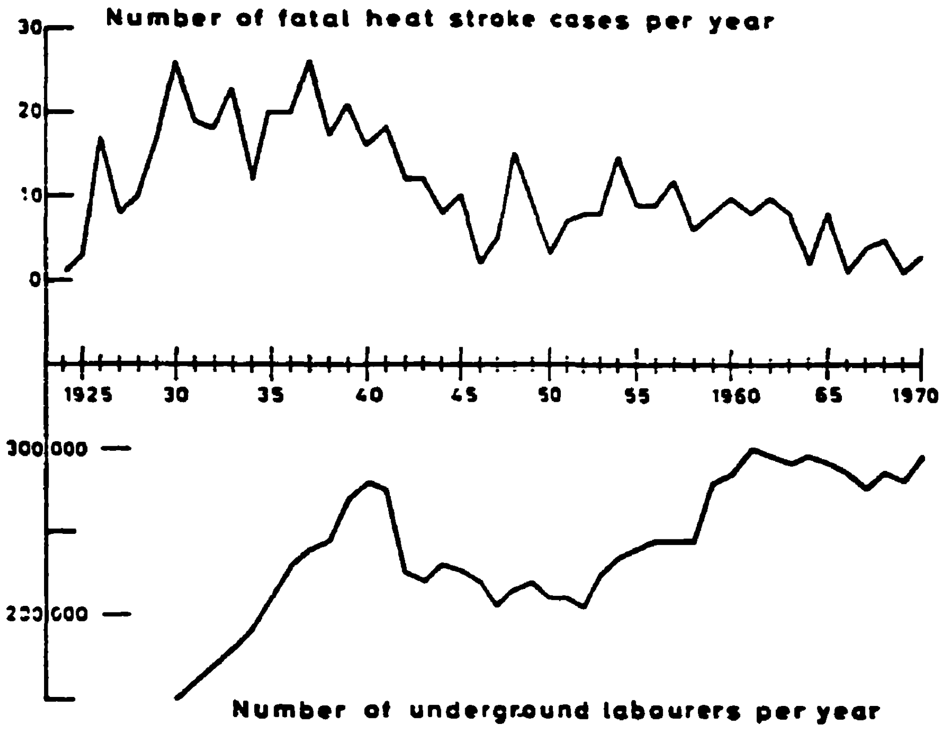

Figure 4.1: Fatal Heat Stroke Statistics (After Wyndhatt, 1974)

4.3.5 Other Effects of Warm Environments

Other effects of hot climatic conditions are increases in accident rates and reduced productivity. These effects are serious as they may result in financial and material losses and increased workforce discontentment. The results of high wet bulb temperature on work performance together with 78 per cent confidence limits are shown in Figure 4.2. Leask (1979) showed the drastic decrease in accident frequency as a result of improvements in environmental conditions. The decrease of accident frequency as a consequence of a decrease in wet bulb temperature from 31.8 to 28°C is depicted in Figure 4.3. The literature verifies the acute consequence of heat on health and safety.

The workforce must be protected from heat illness; specifically heat stroke. Protection can be guaranteed by employing work standards designed to account for the heat stress incurred by underground workers.

Unfortunately, over 90 indices of heat stress have been developed during the twentieth century (Hanoff, 1970) and physiologists are divided about their merits. This demonstrates the numerous intricate variables, the complexity of human thermoregulation, and the magnitude of climatic aspects to be considered.

A further issue in many countries with hot mines is that they do not employ women, and due to this, their physiological response to heat stress is not sufficiently known. Nonetheless, Wyndham et al. (1965) succeeded in acclimatizing four female medical students. The experiment resulted in distress to some participants, but revealed that women can also effectively acclimatize to such conditions. After acclimatization, the female sweat rate is lower, yet they could endure more subjective distress than men (U.S. Dept. Health, 1972).

Age, weight and physical fitness also have appreciable effects on the amount of stress endured. Some people acclimatize readily and can tolerate heat, whereas others are heat intolerant and have difficulties acclimatizing to work in hot conditions.

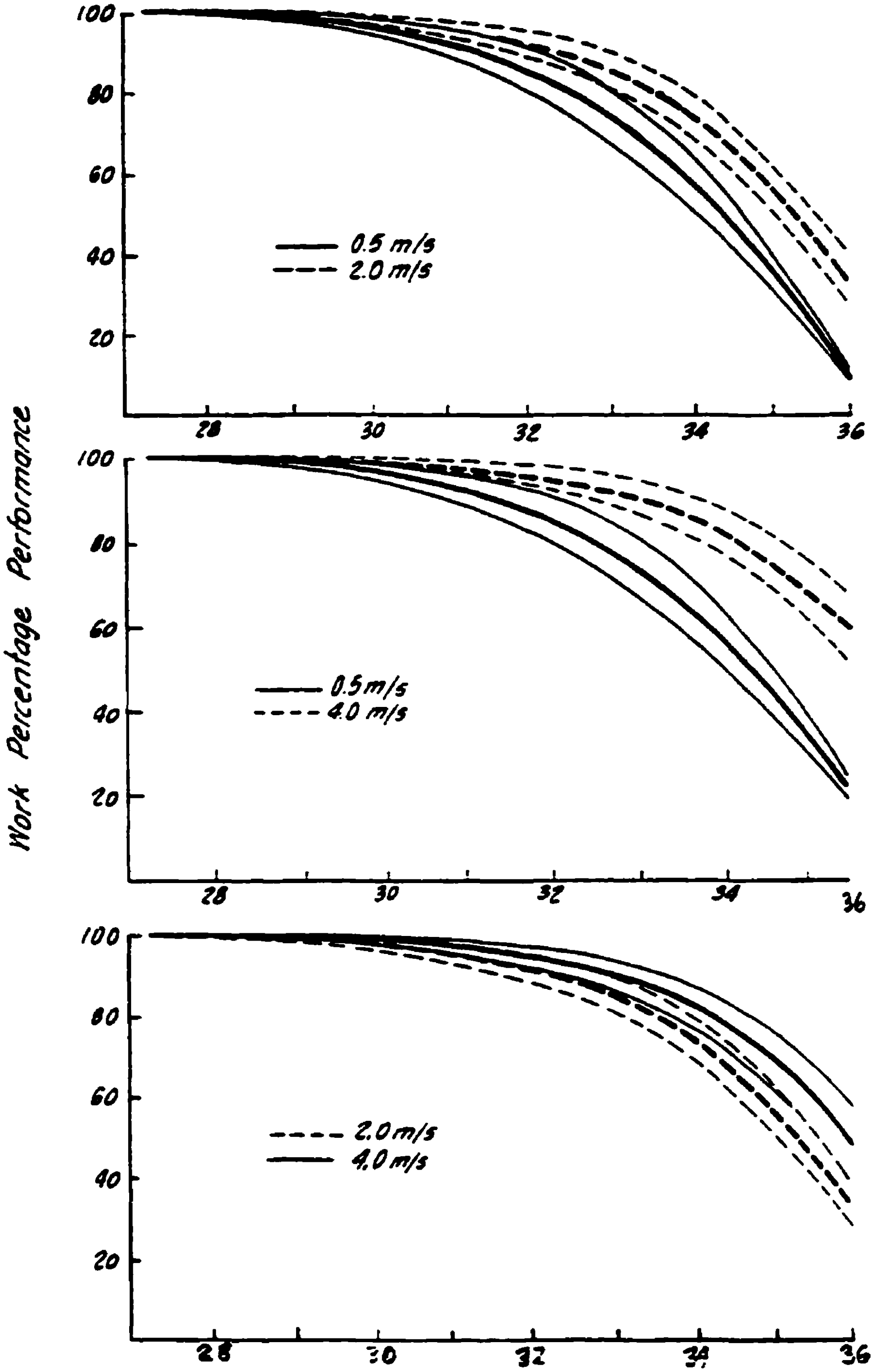

Figure 4.2: Effect of Wet Bulb Temperatures on Work Performance at Various Air Velocities With 78% Confidence Limits (After Wyndham, 1974)

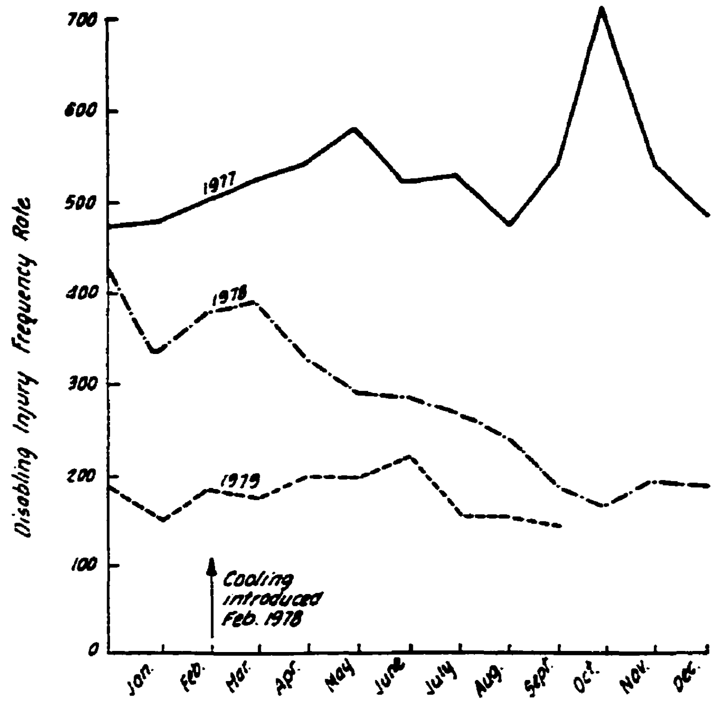

Figure 4.3: Effect of Reduction of Wet Bulb Temperature From 31.8°C to 28°C in South African Gold Mine (After Leask, 1979)

4.4 Indices of Heat Stress For Work Regulation

Heat stress indices are the subject of most disagreement among physiologists due to the complexity of heat stress and strain phenomena. Over 90 indices of heat stress were developed during the twentieth century (Hanoff, 1970).
An acceptable heat stress index should at least fulfill the following criteria, namely: include all essential heat stress parameters, consist of simple measures and calculations, be suitable for industrial use, and be applicable as a regulatory standard.

The in-depth analysis of these indices is outside the scope of this work, and is confined to the following three types of heat stress indices classification: (McPherson, 1993)

1. Single measurements
2. Empirical methods, and
3. Rational indices

4.4.1 Single Measurements

There is no single psychrometric measurement that on its own would provide reliable information of physiological reaction. In hot and humid mines where the principal mode of heat flow in the human body is evaporation, the wet bulb temperature of the air is the most important parameter affecting body cooling. Worldwide, many mines use the wet bulb temperature as the only indicator of climatic acceptability, whereas dry bulb temperature alone has limited outcome on climatic acceptability in hot mines. Nevertheless, dry bulb temperature above 45°C can produce a burning sensation.

The second most important parameter that affects heat stress is air velocity. However, air velocity alone is also a misleading variable and is usually combined with wet bulb temperature as an indication of climatic acceptability.

4.4.2 Empirical Methods

These methods provide heat stress indices that came about from statistical assessment, or are based on observation of individuals under conditions of a controlled climate, utilizing simplified relationship of measurable parameters.

4.4.3 Effective Temperature Index (ET)

The effective temperature scales were contrived in the mid-1920's by Houghton and Yaglou (Golder Associates, 1980). The effective temperature is one of the older, and presumably, most widely used indices of heat stress. In 1946, Bedford suggested a correction to allow for radiant heat and the scales became known as the Corrected Effective Temperature (CET) scales. The scales are shown in Figure 4.4.

The indices is criticized by physiologists, namely Wyndham (1974), for the following reasons:

- They give inadequate consideration to the deleterious effect of low air velocity
- They overstate the contribution of high dry bulb temperatures for air velocities in the range of 0.5 to 1 m/s
- They give insufficient consideration to the harmful effects of high air velocity at air temperatures exceeding 49°C.

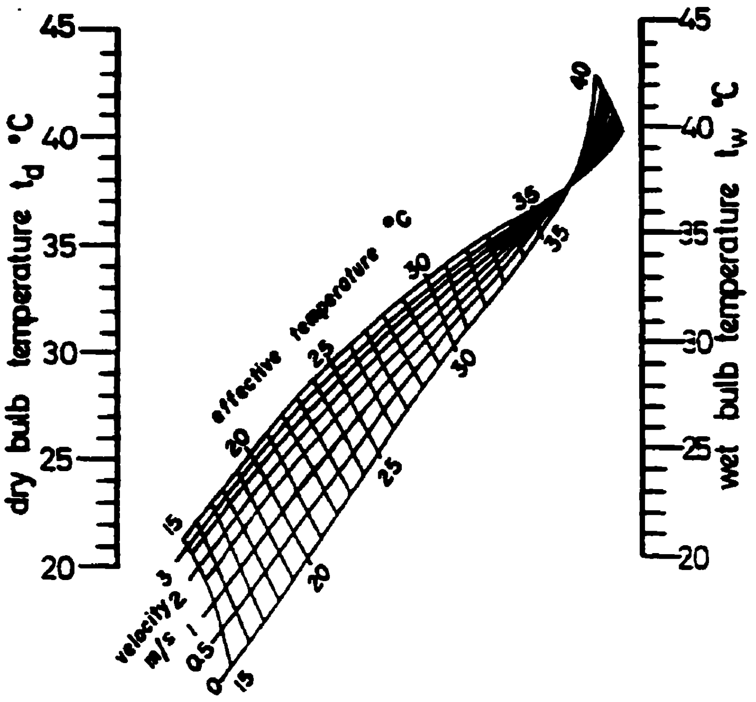

Figure 4.4: Effective Temperature Chart

For the above reasons most mining operations have ceased using the CET as a measurement of heat stress. Effective temperature is however in use in the United Kingdom and Germany, whereas in the USA it is no longer ratified (McPherson, 1993).

4.4.4 Predicted 4 Hour Sweat Rate (P4SR)

The P4SR scale was developed in the late 40's from the results of a large number of experiments. Leithead and Lind (1964) deemed the index to be the most factual of those available. In spite of its sound physiological basis, the index did not get acceptance by the mining industry. The index has been employed at the Mount Isa Mine in Australia, but has most probably been discredited by other mines because the calculations required to obtain P4SR values are complicated.

4.4.5 Heat Stress Index (HSI)

Belding and Hatch (1955) put forward an index for the stress produced by giving a set of conditions for work and climate. The index consists of four factors, namely; air temperature, humidity, radiant heat, and wind velocity.

Leithead and Lind (1964) perceived the index to be principally sound, but not as accurate than the P4SR. NIOSH, U.S. Dept. of Health (1972) did not accept the index on the same grounds as for the P4SR in that the calculation of the index is deemed to be too complicated for general use. This view was also expressed by the MESA report on heat stress, U.S. Dept. of the Interior (1976).

4.4.6 Wet Bulb Globe Temperature (WBGT)

The WBGT is based on temperature measurements alone. When solar radiation is nonexistent, the following formula is employed:

$$WBGT = 0.7\ WB + 0.3\ GT$$

where:

WB = natural wet bulb temperature determined by a thermometer with a wetted wick hung in the environment.

GT = black-globe temperature

The black globe temperature is obtained using a thermometer placed in the center of a hollow sphere, or black globe of standard dimension.

The wet bulb globe temperature is a function of key parameters that play a role in physiological reaction, specifically, wet and dry bulb temperatures, air velocity and radiant temperature. It does not however require a separate measurement of air velocity. For the majority of underground situations, the radiant and the dry bulb temperatures are within a degree or so of each other and the net bulb globe temperature (WBGT) is obtained by weighting the wet bulb by 0.7 and dry bulb by 0.3 (Howes and Nixon, 1997).

Wet bulb globe temperature has the advantage of simplicity and has been employed in the United States Mine Safety and Health Administration, U.S. Dept. of the Interior (1976) in accordance with the NIOSH report, U.S. Dept. of Health (1972). Heat stress levels based on the WBGT have also been incorporated in an international standard (ISO.1982). Leithead and Lind (1964) deliberated that the index is subject to the same constraints as effective temperature and should not be adopted at high levels of climatic heat stress.

4.4.7 Rational Indices

A rational index of heat stress is based on the physiological heat balance approach.

Air cooling power (M scale) or metabolic heat, ACPM.

The Chamber of Mines of South Africa have critiqued the WBGT index and developed an index known as the air-cooling power (M scale) or ACPM. The index was developed for use with acclimatized men working in near-saturated temperatures. Stewart (1979) described the basis of the index, which correlates well with wet kata thermometer readings.

McPherson (1993), based on Stewart's (1982) work developed an air-cooling chart Figure 4.5. The lines on this chart were established from equilibrium conditions, when the ambient environment removes metabolic heat at the same rate as it is produced.

The usage of the chart is a straightforward matter for a rapid means of assessing climatic acceptability of any given mine environmental condition. The wet bulb temperature and metabolic heat generated are plotted as a coordinate point.

If this point is above the applicable air velocity line then, the cooling power of the air is greater than metabolic heat production and personnel are able to obtain thermal equilibrium with the environment for a similar rate of work. On the other hand, should the air-cooling power be less than the applicable limit curve, mine personnel will remove clothing, diminish their metabolic rate by resting, or become vulnerable to the onset of heat strain. Finally, as the contribution of dry bulb temperature, radiant temperature and barometric pressure are rather weak, the ACPM chart is well suited for wet bulb depressions from 2 to 8°C, and the associated range of barometric pressures that prevail in mine workings.

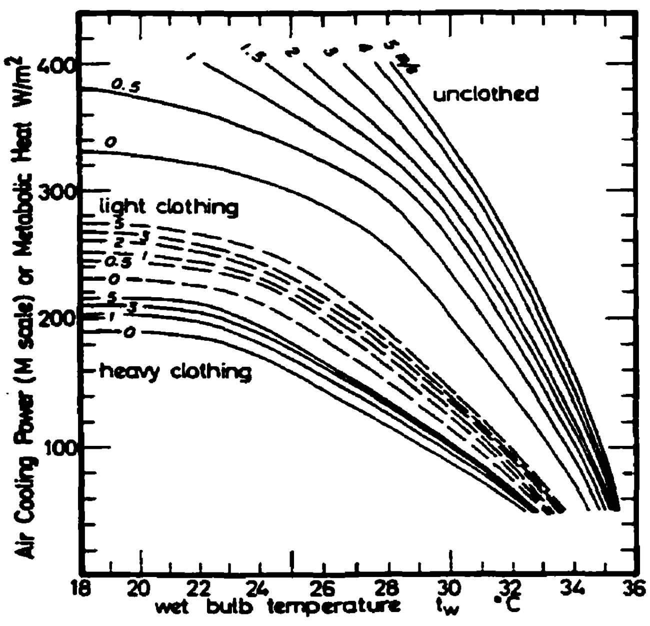

Figure 4.5: Air Cooling Power (M Scale) or ACPM Chart, After (McPherson, 1992)

4.5 Conclusion

Heat stress is a particularly challenging issue for deep mining. With the already difficult economic problems of mining at depth heat stress has the potential of diminishing productivity and rendering work uncomfortable. The physical effects can be numerous and include heat stroke, circulatory deficiency heat exhaustion, salt deficiency heat exhaustion, and heat rash.

One of the limitations in dealing with heat stress is a reliable method of measuring environmental conditions and relating it to different levels of heat stress.

Several methods of heat stress quantification have been developed such as the effective temperature index (ETI), the predicted 4-hour sweat rate (P4SR), the heat stress index (HIS). However, the majority of these methods have been deemed too complicated for general use and the one recognized by ISO and the United States Mine Safety and Health Administration is the wet bulb globe temperature (WBGT) for its simplicity. The National Institute of Occupational Safety and Health of the United States (NIOSH, 1986) and the American Conference of Governmental Industrial Hygienists has employed WBGT as heat stress standard and recommended the exposure limits and stated: "Heat produced by the body and environmental heat together determine the total heat load. Therefore, if work is to be performed under hot environmental conditions, the workload category of each job shall be established and the heat exposure limit pertinent to the workload evaluated against the applicable standard, in order to protect the worker from exposure beyond the permissible limit. The workload category may be established by ranking each job into light, medium and heavy categories on the basis of type of operation. As per Table 4.1, this heat stress standard provides a fairly comprehensive range of permissible values, which are related to the rate of work. In broad terms, the WBGT limits for light, medium and hard work are 30, 26.7 and 25°C respectively. The WBGT is well suited to the rapid assessment of climatic conditions in existing workings. At present it is however less useful for prediction of air cooling power in underground openings that have not yet been constructed, since natural wet bulb globe temperature is not included as output in current subsurface climate simulators (McPherson, 1993). Critics of the WBGT deem that this measurement should not be relied upon under high levels of climatic stress. The Chamber of Mines of South Africa promoted the air cooling power (M scale) or APCM, which is also a quite straightforward method. Yet there is still no general consensus on which method is most preferable and should be widely adopted.

	Workload		
Work – Rest Regimen	**Light**	**Moderate**	**Heavy**
Continuous Work	30.0 °C	26.7 °C	25.0 °C
75% Work – 25 Rest, Each Hour	30.6 °C	28.0 °C	25.9 °C
50% Work – 50% Rest, Each Hour	31.4 °C	29.4 °C	27.9 °C
25% Work – 75% Rest, Each Hour	32.2 °C	31.1 °C	30.0 °C

Table 4.1: Permissible Heat Exposure Threshold Limited Values, °C Wet Bulb Globe Temperature (W.B.G.T.), The American Conference of Governmental Industrial Hygienists (ACGIH, 1986-87)

Chapter 5

5.1 SOURCES OF HEAT IN DEEP MINES

5.1.1 Introduction

At many deep mines, heat is a major problem and the provision of an acceptable working environment requires considerable expenditure on ventilation and cooling systems (Hemp 1979).

Steed (1955), points out that the rock masses, into which mining activities take place, may be considered as vast ovens with an almost unlimited supply of heat, and that the heat transfer from the rock to the air will depend upon the physical properties of the rock, upon the temperature of the air, and upon the type of cooling.

Heat flow from wall rocks and broken rock into airways is only one of the principal heat loads placed on ventilating systems. Other principal contributors to the heating of mine air are as follows:

- Adiabatic compression
- Electrical and mechanical equipment
- Oxidation of sulfides and timber
- Heat flow from water
- Blasting
- Body metabolism

The rational design and operation of ventilation systems in deep, hot mines depends upon an understanding of the above sources of heat in the mine.

5.2 Heat Flow From Strata

The heat liberated from the wall rock is a principal component of the overall heat load in a mine. Rock heat is challenging to assess, because the heat flow from rock to air varies with time. As a cooled, insulated layer of rock is developed around a mine opening, the heat flow into the opening decreases. The rate at which this happens is primarily dependent upon the thermal properties of rock and the temperature difference between the rock and the airway.

The source of rock heat on the continent is predominantly from radioactivity in the rocks of the earth's crust, where heat is produced by the decay of the isotopes U 238, Th 232, and K 40, and not from the cooling of magmatic bodies, a popular belief of the past (Gaskell, 1967).

When taking heat loads into consideration, Canada is advantaged in that the rock temperature near surface is only about 4.4°C compared to 25°C or more in warmer countries. In situ rock temperature is called virgin rock temperature (VRT) and the rate of change of temperature with depth is expressed as the geothermal gradient. Gradients vary substantially in different mining areas in Canada and elsewhere (Golder Associates, 1980), and to a lesser extent within a large deep mine.

Typical geothermal gradient data, from selected mines throughout the world, are given in Table 5.1.

Table 5.1: Examples of Geothermal Gradients Worldwide (After Judge 1972)

REGION		m/°C
Africa		
	South Africa	
	Central Rand	100
	Kerksdorp	98
	Orange Free State	77
	Zambia	
	Mindola	
Australia		
	Broken Hill	40
	Jabiluka	38
	Mt. Isa	52
Great Britain		
	Kolar	75
South America		
	Brazil	
	Morro Velho	77-35
United States		
	Arizona	
	San Manuel	62-42
	Superior	37-28
	California	
	Grass Valley	104-96
	Michigan	69-60
	Montana	
	Butte	34-16

Geothermal gradients measured, and their depths, are provided in Table 5.2 for some of the mining districts in Canada.

Table 5.2: Geothermal Gradients in Canada (After Judge 1972)

Location	Depth Below Surface m	V.R.T. °C	Geothermal Gradient m/°C
British Columbia			
Bralorne	1770	40	38
Ontario			
Kirkland Lk. Area	1220	19	77.4
	1525	23	
	1830	27	
Sudbury Area			
Falconbridge Mine	1220	21	78.5
	1525	25	
	1830	29	
Onaping	1220	21.4	60.9
	1525	26.4	
Lockerby Mine	915	19.4	62.5
	1220	24.3	
Creighton Mine	610	15.3	
	915	20.4	58.7
	1220	25.8	
	1525	31.1	57.6
	1830	36.7	
	2135	42.2	
	3050	58.9	54.9
Frood-Stobie Mine	610	14.3	74.1
	915	18.4	
Levack Mine	610	14.3	65.8
	915	18.9	
Levack Borehole (Inco)	2135	37.2	57.6
	2440	42.5	
	3050	53.1	
Garson Mine	610	12.5	
	915	15.8	91.6
	1220	20.8	
	1525	25.7	61.4
Timmins Area			
Porcupine			107.5

Values of thermal conductivity for different rock types are given in Table 5.3.

Table 5.3: Thermal Conductivities of Some Common Rock Types (After Judge 1972)

Lithology	Thermal Conductivity W m-1 °C-1
Sedimentary Rocks	
Shale	1.5
Limestone	2.9
Sandstone Quartz	4.2
Dolomite	5.0
Evaporites	5.4
Metamorphic Rocks	
Gneiss	2.5
Greenstone	3.3
Slate	3.8
Argillite	3.3
Quartzite	5.9
Intrusive Rocks	
Gabbro	2.5
Diabase	2.1
Granite	2.9

The geothermal flow of heat passing from the earth's core has an average value of 0.05 to 0.06 W/m^2. Heat flow values in the Canadian Shield obtained by several researchers, Sass et al (1968) indicate a value of 0.055 W/m^2 near Elliot Lake, Ontario. Lewis (1969) reports a heat flow of 0.05 W/m^2 in the Eldorado mine. Geothermal heat certainly can be much higher in areas of anomalous geothermal activity. The detail values of heat flow can be found in work done by Jesop & Lewis (1978).

5.3 Heat Transfer Coefficient and Moisture

When cool air flows through a mine, where the rock is at a higher temperature than the air, the temperature of the surrounding rock walls reduces gradually below the VRT. The rate at which this rock cools is denoted as the thermal diffusivity and values of this coefficient for various rock types are given in Table 5.4.

Table 5.4: Thermal Diffusivity of Some Common Rock Types (After Judge 1972)

Location	Rock Type	Thermal Diffusivity cm^2/s
South Africa	Quartzite	0.0065
	Lava	0.0032
Great Britain	Mudstone	0.0034
	Siltstone	0.0026
	Sandstone	0.0158
Butte, Montana	Granite (fresh)	0.0175
General	Granite	0.011
	Limestone	0.007
	Dolomite	0.020
	Sandstone	0.010
	Shale	0.008
	Ice	0.012

Thermal diffusivity undergoes changes with temperature differentials, diminishes with time, and becomes negligible when the temperature differentials approach zero. This layer of cooled rock acts as an insulating barrier inhibiting heat flow from interior rock. From research conducted in South Africa (Rees, 1939), and the United States (Robertson and Bossard, 1970), it was ascertained that the major cooling of a new airway took place in a year or less.

Heat transfer from rock to air is less under dry than wet conditions. Heat emanation by convection in a dry airway can be assessed if the temperature differential and heat coefficient is known. However, wet airways produce a more rapid increase in wet bulb temperature than dry airways, and Steed (1955), quoting from Jeppe and McIntre indicates that the rate of heat flow from rock to air may be increased a hundred times if a thin layer of water covers the rock. Even though water is a poor conductor of heat, the greatly increased rate of heat flow is probably due to evaporation.

The rate of heat flow is associated with dry bulb temperature, although the effects may be masked by the evaporation of water. On the other hand, there is no direct correlation of heat flow to wet bulb temperature. Nevertheless, a number of empirical formulae have been produced which relate the increase in wet bulb temperature to wetness of the wall rocks; air velocity and the difference between virgin rock temperature and wet bulb temperature.

$$\text{Heat Flow} = \frac{\text{Wetness Factor (VRT -- Wet Bulb Temperature)}}{\text{Air Velocity}}$$

Under wet conditions, heat may flow from colder rock surface to warmer air by the evaporation process.

Beadle (1955), summarized the effect of evaporation of water on heat transfer as follows:

(a) If there is no evaporation-taking place, no heat will flow from the rock surfaces to the air if the dry bulb temperature is higher than the temperature of the sources of heat.

(b) If water is available for evaporation, the following occurs:

- Water evaporates; dry bulb temperature of the air reduces
- Dry bulb temperature falls; increase in rate of heat flow from the rocks to the air
- Increase in heat flow; produces increase in heat content of the air
- Increase in heat content; results in rise in wet bulb temperature
- Increase in wet bulb temperature; worsens environmental conditions.

5.4 Adiabatic Compression

A principal cause of high temperature in deep mines is adiabatic compression, which causes both the dry bulb and the wet bulb temperatures to increase as the air flows down the shaft system. The increase is due to the conversion of potential energy into thermal energy. In other words, the air, which is used to ventilate the mine, will heat up as it descends, due to compression upon itself.

As the air flows down through the intake airways, its pressure and temperature increases and this process is known as "auto-compression". When this takes place without any flow of heat into the air from surrounding rock, and with constant moisture, the process is called "pure" auto-compression. The temperature increase due to auto-compression alone, is 0.95°C per 100 m.

However, if water evaporates in the intake airway, "latent heat" is required, and some of this heat will be taken from the air. The dry bulb temperature will fall, masking to some extent the rise of the dry bulb temperature due to auto-compression. The net result is then less than 0.95°C per 100 m.

A common figure in such cases is about 0.55°C per 100 m, but the actual figure may vary and can even be a negative value. The true change in wet bulb temperature is best calculated from psychometric charts.

Air going down the intake raise is also subjected to the air temperature changes caused by seasonal and daily variations. These changes, however, are very much reduced with greater depths, as shown in Figure 5.1.

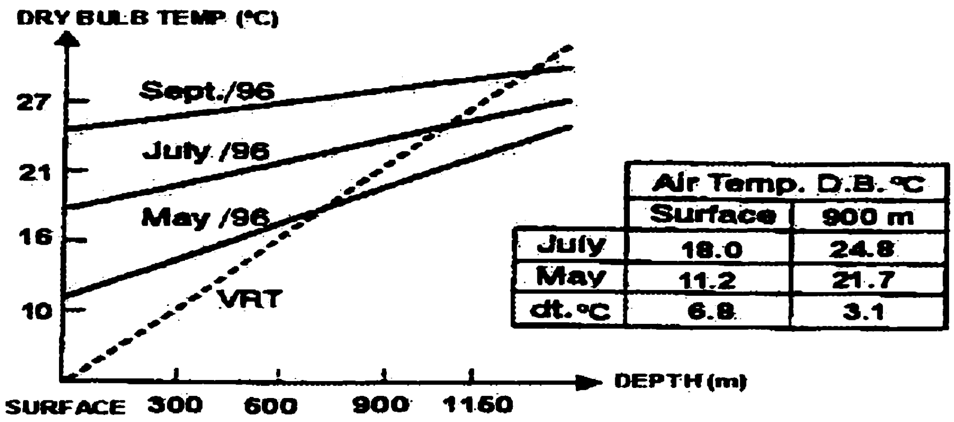

Figure 5.1: Temperature Gradient of Fresh Air System at Inco's Garson Mine as a Function of Depth and Season

5.5 Electrical and Mechanical Equipment

Sources of heat from mechanical and electrical equipment in Canadian mines are:

- Diesel loading and haulage units
- Electric motors and lighting
- Underground hoists
- Fans
- Pumps

5.5.1 Diesel Engines

The wide-scale use of diesel loading and haulage (LHD) units in the last 30 years has made this equipment a significant source of heat in Canadian mines.

Oakes and Hinsley (1955 to 1956), state that heat produced by diesel equipment is equivalent to the calorific value of the total fuel consumed.

5.5.2 Electrical Equipment

Electric motors produce a substantial amount of total heat in the mines when employed to drive fans, underground hoists, pumps, scrapers, crushers, conveyors, battery locomotives and chargers, trolleys, trackless loading and haulage vehicles, Alimak raise climbers, raise boring machines and occasionally compressors for drilling.

Miner's lamps and lighting fixtures also generate heat.

Large quantities of heat (about 50 percent of horsepower) are generated for short periods of time during acceleration and deceleration of underground hoists.

Richardson (1950), indicates that only some 15 percent of pump horsepower is converted to heat.

Fenton (1973), estimates that all the horsepower used in tunnel boring is transformed to heat. However, some is accumulated by the rock and released after boring, decreasing the peak heat load.

The heat given to the air by fans is equal to the thermal equivalent of the electrical energy put into the fan motor. The process is a result of pressure energy being turned into entropy.

Since the practice of adopting after-coolers in compressors to prevent explosions, a negligible quantity of heat is transferred from main compressed airlines in shafts.

5.6 Heat From Oxidation

The heat generated from the oxidation of sulfides in mines is a function of the grade of ore, the type of reactive pyrrhotites present and the time broken ore is left in the workings. Fortunately for Inco, the mining methods used minimize the duration of ore storage in stopes. Therefore, the quantity of heat generated by the oxidation process is negligible.

5.7 Heat From Water Flow

Water in a mine contains ground water from faults, cracks and drill holes, service water used for drilling, spraying to suppress dust, and water used for backfill.

Unsaturated air, flowing over water in ditches and on floors of workings, will increase humidity levels and latent heat by evaporation. Fenton (1973) provides a formula by Ash for estimating the quantity of heat added to air from a free water surface.

Enderlin (1973), commenting on work reported by Hinsley and Morris (1951), recommends that 3 m/s is the most appropriate air velocity for minimum heat gain. He also indicates that in the case where the water is warmer than the air, the sensible heat load will be increased.

One of the principal factors influencing the rate of evaporation is the surface area exposed to air; the actual quantity of water is of less importance. Eliminating the water surface area exposed can prevent evaporation. Steed (1955), points out that the sources of water rates in ease of evaporation are as follows:

- Rain in shafts, water sprays, drips from loading boxes in shafts, haulages and stopes
- Drips from fissures not sealed or covered, leaks in water service mains
- Damp shaft walls (from seepage if concrete lines), stations watered down
- Moist track ballast – drips from cars and capillary action from drains
- Water pools between tracks due to drips from cars
- Open drains.

5.8 Curing from Cemented Backfill

Calculations are based on a heat generation rate of 186.7 kJ/kg of cement during the first 24 hours of curing (data obtained from the supplier). The total heat generated amounts to 376.8 kJ/kg. It is assumed that 30 percent of the heat is removed by air and the remaining 70 percent by water. According to the supplier, the curing gives off 315.3 kJ/kg of cement during the first seven days, and 376.8 kJ/kg (162 Btu/lb.) in 28 days (Fenton 1973).

5.9 Blasting

Fenton (1973), and Enderlin (1973), state that all the explosive energy is converted to heat in an underground blast. However, some of the heat is accumulated in the broken rock and released later. The heat content in various types of explosives is depicted in Table 5.5.

Table 5.5: Heat Potential of Various Types of Explosives (After Cook 1958)

Explosives	Q (kJ/kg)
Nitroglycerin	5943
60% Straight Dynamite	4143
40% Straight Dynamite	3891
100% Straight Gelatin	5859
75% Straight Gelatin	4812
40% Straight Gelatin	3431
75% Ammonia Gelatin	4142
40% Ammonia Gelatin	3347
Semi-Gelatin	3933
AN – 1-0 94.5/5.5	3724
AN – FO 94.3/5.7	3880
AN – Al – Water	4603 – 5022

Hemp and Deglon (1979), also indicate that heat is absorbed in the rock to be subsequently released.

5.10 Body Metabolism

The amount of heat produced by the metabolism of miners varies with the intensity of their work.

Enderlin (1973) quotes Forbes et al (1949), as stating that a miner working at full normal capacity may produce heat at the average rate of 293 W.

5.11 Conclusion

As was illustrated in this chapter, heat generated in a mine originates from many sources.

- Wall rock
- Adiabatic compressor of air
- Electrical and mechanical equipment
- Oxidation
- Blasting
- Human body metabolism

All of the sources contribute, in varying proportion to the overall heat load of a mine. As an example, Table 5.6 illustrates, the weighted source of heat at Creighton Mine (Stachulak, 1978).

Description	Percent
Wall, rock, oxidation, etc.	36
Auto Compression	35
Fans and Diesel Equipment	26
Other	3
Total	**100**

Table 5.6: Sources of Heat in a Mine

Such intimate knowledge of a mine's heat load permits ventilation engineers to focus on areas of concern during the design and operational phase of a mine.

Chapter 6

6.1 HEAT STORAGE CAPACITY OF CREIGHTON'S NATURAL HEAT EXCHANGE AREAS

6.1.1 Introduction

The schematic of the heat exchange area is depicted in Figure 6.1.

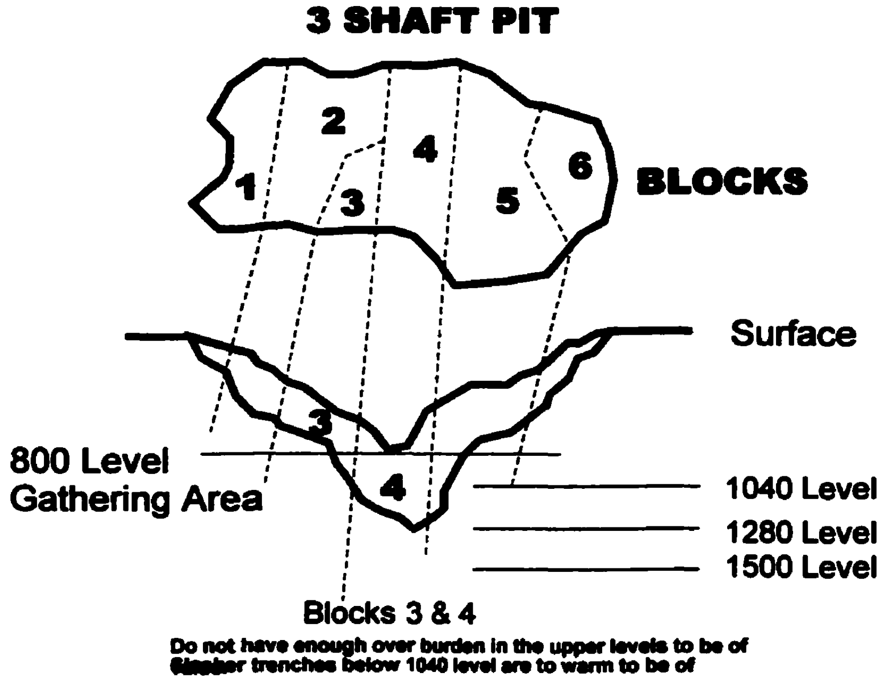

Figure 6.1: Schematic of Heat Exchange Area

The zones encompassed by the old open pit and the fragmented rock beneath it is denoted as six "block". At the present time, intake air is being drawn through blocks 2, 5 and 6. Blocks 5 and 6 are recognized as being the most efficient with respect to heat transfer and, hence, the degree of air-conditioning that they provide. The rock contained in blocks 3 and 4 is insufficiently fragmented to "store heat". This leaves block 1, which is currently being prepared for utilization. Blocks 1 and 2 were mined using similar methods and may be expected to have produced the same degree of fragmentation of the waste rock. However, block 1 is smaller and has only 44 percent of the number of boxholes that connect into block 2. If the additional degree of air conditioning required for the new lower workings cannot be obtained from block 1 then the airflows drawn through the other usable blocks will be increased. This will necessitate additional fan power. A further consequence of drawing larger airflows through any given block(s) is that the amplitude of the seasonal cycle of temperatures in the 800 level gathering airways will tend to increase. When this becomes untenable the useful limit of the heat exchange area will have been reached.

Observations of the air-cooling capacity realized from fragmented rock have served as a guide in assessing the current and projected cooling capacity of specified blocks in the heat exchange area. With the last available block (No. 1) to be brought into operation, a projected airflow of 279 m^3/s at 1.26 kg/m^3 density needs to be cooled and is approaching the limit to which mining can take place without mechanical refrigeration. For this reason, it becomes necessary to maximize the air-cooling capacity of the natural heat exchange areas. This necessitates better understanding of the heat transfer phenomena that take place.

As the air flows from surface down through the broken rock, heat exchange takes place between the rock and the descending air. Ice melting during the summer months also plays a minor role. The mechanisms of heat exchange between the rock and the air are the same regardless of the direction of heat transfer. For the purposes of explanation, let us consider winter condition when air is being heated by the broken rock. Within the rock mass (Whillier, 1982), heat passes by conduction through each discrete fragment until it reaches the rock-air interface in a void. Within the air but very close to the rock surface, a boundary layer exists through which the air velocity increases from zero (actually at the surface) to the full velocity which prevails along the specific flow path. The heat flowing out from the rock passes through the boundary layer by conduction and molecular diffusion. The boundary layer behaves as an insulating layer, suppressing heat transfer. The heat transfer process nevertheless converts into eddy diffusion when the distance from the rock surface has become sufficient to sustain turbulent eddies. This is a much more efficient mode of heat transfer.

Where the air velocity in the flow paths is higher, the thickness of the boundary layers will diminish and, therefore heat exchange will be improved. If the same masses of large fragments and small fragments of rock are exposed to the similar airflow, then the smaller fragments will cool more quickly because of their larger total surface area. However, there could exist areas of rather highly compacted smaller fragments of rock where the air stream is sluggish and laminar. In such zones, heat exchange is not assisted by turbulent eddies and will proceed slowly.

Therefore, the rate at which any given broken rock fragment yields heat is a function of the size of the fragment, the mean velocity of the air in adjoining voids and whether the flow is laminar, turbulent or in a transitional regime (Reynold's Number effects). Due to these factors and processes, there ought to be a non-linear relationship between the rate at which air is pulled through the fragmented mass of broken rock and the rate at which heat transfer takes place.

6.2 Cooling Capacity of Heat Exchange Areas

The assessment of cooling capacity of the heat exchange areas will be carried out with the following assumptions:

- Linear relationship between the rate at which air is drawn through the fragmented mass of the broken rock and the rate at which heat transfer takes place.
- Broken rock (fragmented) consists some 2/3 of rock (Norite) with the remainder being air spaces or ice.
- Broken rock tonnage factor of 0.51 m^3/tonne
- Any ice contributing to the cooling effect substantially must change to water in the summer months, and be replaced by precipitation falling and freezing during the winter months, thus releasing heat
- The range of 45 – 25-degree angle of draw is employed. The 25-degree angle depicts the conservative estimate of the angle at which the rock is broken.

The following data are used in the evaluation of cooling capacity of heat exchange areas, specifically block 5 and 6.

- Sudbury area average temperature for:

  Six summer months:
  May - October = 10.2°C (W.B.)

  and six winter months:
  Nov. - April = -7.3°C (W.B)

- Bottom of heat exchange areas (800L), average temperature for six summer months = 2.7°C (W.B.), and six winter months = 3.1°C W.B.

- The airflow = 283.2 m^3/s percolating through heat exchange areas, namely blocks 5 and 6.

6.3 Example of Calculation

Considering an average surface temperature of 10.2°C W.B. for six summer months, and an average temperature of intake at 800 Level of 2.7°C W.B. (years 73-78).

Σ Heat at 10.2°C W.B. at 98.2 kPa Hg Bar. Press. 48.4 kJ/kg
Σ Heat at 2.7°C W.B. at 101.6 kPa Hg Bar. Press. 32.1 kJ/kg

Σ Heat = 16.3 kJ/kg

Adjustment from 800 Level to Surface = 2.33 kJ/kg

Σ Heat = 18.6 kJ/kg

Average cooling effect of caved area for the six summer months is:

$283.2\ m^3/s \times 1.36\ kg/m^3 \times 18.6\ kJ/kg = 7164$ refrigeration kW

Where:	$283.2\ m^3/s$	=	Airflow
	$1.36\ kg/m^3$	=	Air Density
	18.6 kJ/kg	=	Σ Heat

Considering an average surface temperature of -7.3°C W.B. for six winter months, and an average temperature of intake at 800 Level of 3.1°C W.B. (years 73-78):

Σ Heat at 3.1°C W.B. at 101.6 kPa Bar. Press.	32.6 kJ/kg
Σ Heat at –7.3°C W.B. at 98.2 kPa Bar. Press	-16.3 kJ/kg
	16.3 kJ/kg
+ Altitude Adjustment	2.33 kJ/kg
	18.6 kJ/kg

Using these "global" average temperatures, summer cooling balances winter heating and gives a yearly average temperature at 800L of 2.9°C W.B. (years 73-78).

Furthermore, comparing the heat content of the air on surface with that on 800 Level and correcting for elevation, the fragmented rock in blocks 5 and 6 of the heat exchange area gives an average cooling effect of 7164 refrigeration kW during summer and an equivalent amount of heating during winter. The winter and summer months balance and therefore, the annual cycle can be repeated indefinitely.

6.4 Available Coolth in the Caved Area (Blocks 5 and 6)

Calculated average difference between fragmented rock mass temperature in winter versus summer is some 9.6°C D.B. as illustrated below:

Average surface summer temperature i.e. May – October = 13.2°C D.B.

Average 800L summer temperature	i.e. May – October = 2.8°C D.B.	
		(period 73-78)
Average surface winter temperature	i.e. Nov. – April	= -6.4°C D.B.
Average 800L winter temperature	i.e. Nov. – April	= 3.3°C D.B.
		(period 73-78)

Therefore: $$\frac{(13.2+2.8)}{2} - \frac{(-6.4+3.3)}{2} = 9.6^{\circ}C$$

The fragmented rock heat storage capacity range for blocks 5 & 6 can then be calculated as the product of:

Fragmented rock mass, specific heat and temperature rise, therefore:

10.89×10^6 tonnes x 1000 kg/tonne x (0.879 kJ/kg°C) x (9.6°C) = 9.2×10^{10}kJ

Where:			
	10.89×10^6 tonnes	=	Fragmented rock mass, Table 9
	0.879 kJ/kg°C	=	Specific Heat of Norite
	9.6°C (DB)	=	Temperature Rise, Dry Bulb

There is also surface drainage into the heat exchange area of block 5 and 6.

The plan area for precipitation is 65,030 m^2 (Table 9), the total water per year through the cooling area amounts to:

65,030 m^2 x 0.8026 m x 999.52 kg/m^3 = 52.17×10^6 kg
of this, only that falling in November through April is apt to form ice (Table 11).

Therefore:

65,030 m^2 x 0.3378 m x 999.52 kg/m^3 = 21.96×10^6 kg
Note that snowfall, occurring during the winter, does not provide air heating since the change in state has already occurred before the snow or sleet entered the broken rock mass, but the ice and snow, left at the end of freezing conditions, does cool the air during the summer season.

Ice melted and water heated to an average of 1.67°C provides cooling during summer season, and the amount can be calculated as follows:

21.96×10^6 kg x (334.9 + 6.98) kJ/kg = 7.5×10^9 kJ

Which amounts to estimated total heat storage capacity (blocks 5 and 6) of 9.94×10^{10} kJ.

This amount of coolth is accumulated in approximately six months, and is given up to air in the remaining six months.

$$\frac{9.94\,x10^{10}\,\text{kJ/6months}}{182.5\,\dfrac{\text{days}}{6\,\text{month}}\,x\,1440\,\dfrac{\text{min}}{\text{day}}\,x\,\dfrac{60\text{s}}{\text{min}}} = 6304.0 \text{ refrigeration kW (RkW) or 1792 tons of refrigeration}$$

If however similar assessment of coolth in block 5 and 6 is done based on Table 6.2 then the amount of coolth equals to:

8498 kW or 2416 refrigeration ton.

Tables 6.1, 6.2, 6.3 summarize the Block volumes, tonnages, and surface areas for scenario #1 (25 degree angle), and scenario #2 (45 degree angle). (Detailed spreadsheets are available in Appendix 1.

Block	**1**	**2**	**5**	**6**	**Total**
Volume					
(m^3 x 1.0 E6)	1.78	9.97	2.0	4.12	17.87
Tonnage					
(tonne x 1.0 E6)	3.2	17.8	3.5	7.4	31.9
Surface Area					
(m^2 x 1.0 E6)	0.022	0.082	0.031	0.034	0.167

Table 6.1: Block volume, tonnage and surface area for 25 degree angle of draw.

Block	**1**	**2**	**5**	**6**	**Total**
Volume					
(m^3 X 1.0 E6)	1.81	11.19	2.56	5.63	21.19
Tonnage					
(tonne x 1.0 E6)	3.27	19.96	4.54	9.98	37.75
Surface Area					
(m^2 X 1.0 E6)	0.022	0.105	0.043	0.055	0.223

Table 6.2: Block volume, tonnage and surface area for 45-degree angle of draw.

April	5.8 cm *
May	6.9 cm
June	7.6 cm
July	7.1 cm
August	6.9 cm
September	9.7 cm
October	8.4 cm
November	7.6 cm *
December	5.3 cm *
January	4.8 cm *
February	5.1 cm *
March	5.1 cm *
Total	80.3 cm

*Rain forming ice

Table 6.3: **Average monthly precipitation (calculated as rain)**

6.5 Discussion of the Results

The assumption that the mean temperature of the fragmented rock is the average of the surface and 800 Level dry bulb temperatures involves some uncertainty as it is unlikely that the temperature gradient through the fragmented rock is linear.

The difference between the summer and winter mean rock temperatures is given by this approximation to be 9.6°C.

The volume of the 5 and 6 blocks is estimated to be within the range of 6.12×10^6 to $8.19 \times 10^6\ m^3$ (based on 25 to 45 degree angle of draw), of which two thirds are assumed to be Norite and the rest voidage.

Allowance is made for ice melt, yielding an estimated total heat transfer range of 0.738×10^{10} kJ to 1.055×10^{10} kJ, cooling the air in summer and heating it in the winter months. If the total amount of thermal energy is transferred during a period of six months then the rate of heat transfer would be in the range of 6300 – 8494 kW of refrigeration or 1792 - 2400 refrigeration ton.

The discrepancy between this value and that actually measured (2040 refrigeration tons)may be an indication of the assumptions made in the analysis with respect to the overall porosity of the fragmented rock and variations in the airflow through the heat exchange area.

Neither value is necessarily indicative of the maximum possible heat transfer in the 5 and 6 blocks.

Taking into consideration the above assumptions and the available data in Tables 6.1, 6.2, 6.3 a prognosis of estimated cooling capacity for various airflows is illustrated in Table 6.4. The cursory review of the cooling capacity of the entire heat exchange area indicates, that the limits of natural air conditioning will be reached at an airflow of some 746 m^3/s at 25 degrees of draw of air through fragmented rock, and nearly 896 m^3/s at 45 degrees of draw with a density of 1.26 kg/m^3, therefore, it should be possible to extract additional cooling for planned expansion of 279 m^3/s at 1.26 kg/m^3 density, since the current flow amount to range of 440 – 470 m^3/s. However, the limiting useful level of heat exchange will be indicated by an increasing amplitude of seasonal temperatures in the gathering airways as some additional 279 m^3/s of air increase is sent through the fragmented rock, and this subject matter is examined below.

Blocks	Cooling & Heating Fragmented Rock Capacity at 25 Degree Angle of Draw (Table 8)		Predicted Air Conditioning Air Quantity m^3/s at	
	kW	Tons of Refrigeration	1.26 kg/m^3	1.2 kg/m^3
5, 6	6295	1790	~268	~283
5, 6 & 2	16037	4560	~680	~715
5, 6, 2 & 1	18077	5140	~746	~786

Table 6.4: Natural Heat Exchange Areas – Prognosis of Estimated Cooling Capacity

6.6 Heat Transfer and Temperature Variation in the Fragmented Rock

Another important factor influencing the rate of heat transfer at any given time and location is the temperature difference between the rock and the permeating airflow. The fragments at the top of the rock mass exposed to the surface air vary daily and seasonally. Consequently the mean temperature of exposed rock fragments will also vary rapidly with time. However, as the air percolates downwards and is subjected to heat exchange with the rock, its temperature approaches that of the rock at any given depth. The heat transfer reduces.

Taking into consideration that the thickness of the fragmented rock mass is adequately high or the airflow sufficiently low, then the temperatures of the rock and air will become constant and almost equal in the lower level. The air coming .out from the bottom of the heat exchange area will then remain near constant throughout the year.

As a starting point let us produce conceptual relationships that will assist in understanding the mechanism of heat exchange in this process.

Let the first assumption be, that the complete mass of rock in the heat exchange area is at the same temperature throughout – and let this temperature be the annual mean of 2.8°C.

Now assume that the surface temperature is increased suddenly to its summer mean value of 10.2°C and is maintained at that value. The subsequent behavior of the temperatures through the depth of broken rock is illustrated on Figure 6.2. Immediately after the increase in surface temperature, the temperature gradient in the rock mass is very steep near the surface. However, with the passage of time, the temperature gradient flattens out. Assuming long enough the frame temperature of the rock at the base and, therefore the exiting airflow would increase above its starting point of 2.8°C.

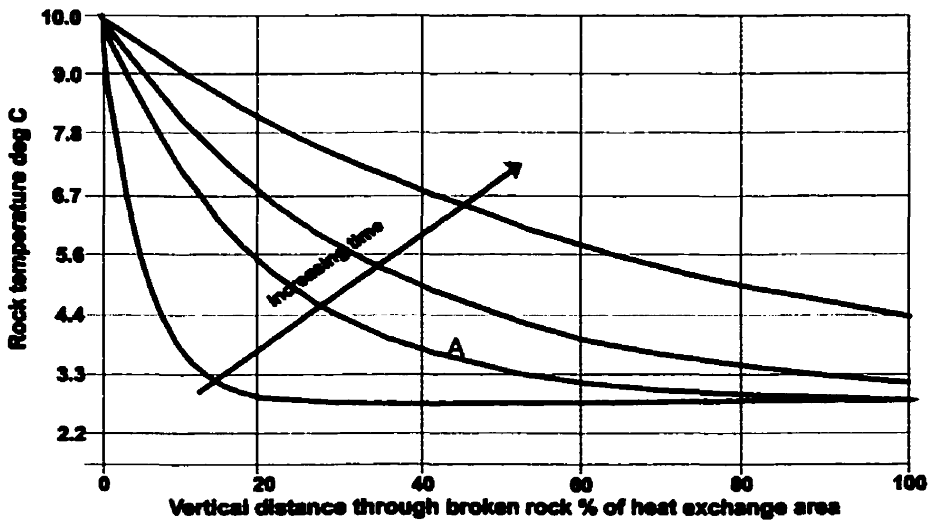

Figure 6.2: Rock Temperature Variation as a Function of Place and Time After Surface Temperature Rapidly Changes from a 2.8°C to 10.2°C.

Now, to expand the conceptual analysis, suppose that at a time when the heat exchange process has reached curve A on Figure 6.2, the surface temperature suddenly changes from 10.2°C to mean winter temperature of –7.3°C. This situation is illustrated on Figure 6.3 near to the surface of the rock temperature is governed basically by the recent change to winter conditions. Further into the rock mass, it remains influenced by the previous summer conditions. As we cycle between summer and winter, a wave of varying temperatures goes down through the rock mass, diminishing in amplitude as it descends.

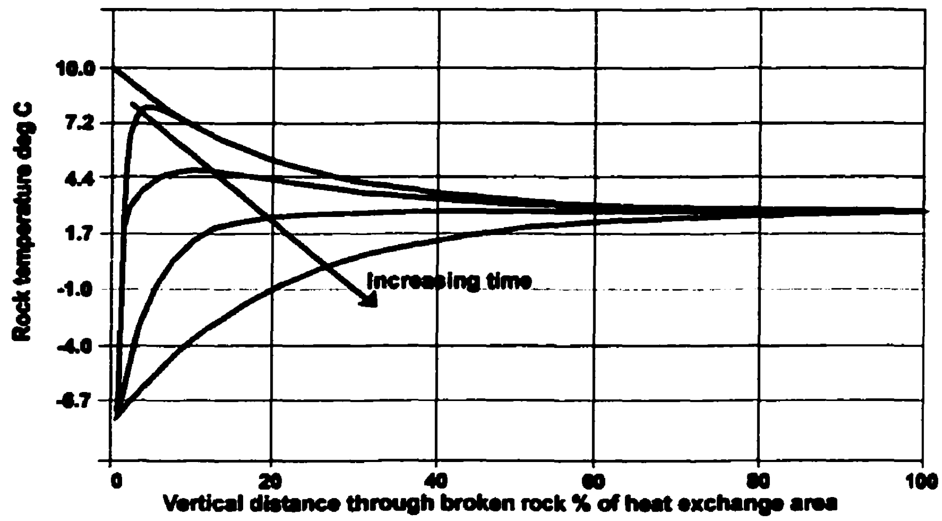

Figure 6.3: Rock Temperature Variation After Swift Change from Surface Mean Summer to Surface Mean Winter Conditions.

In reality, the surface temperature does not change suddenly between summer and winter. Suppose that mean monthly temperatures cycle through the seasons behaves in a manner that may be approximated and described by sine curve. If one applies such a continuously changing situation, then the rock at the top of the heat exchange area will be subjected to seasonal temperature changes that follow the surface temperature. Yearly waves of sinusoidal temperature changes will proceed deeper through the rock mass but they will be of diminishing amplitude as the depth increases.

On Figure 6.4, each sine curve depicts the yearly cycle of rock temperature at a specific distance through the rock mass. This particular figure illustrates a condition where the temperatures of the rock and air approach equilibrium as the air progresses downwards. Therefore, the air leaves the heat exchange area at a temperature that is almost constant throughout the year. If the airflow increases sizably, then the amplitude of the yearly temperature cycle will increase at corresponding points throughout the rock mass. This is shown on Figure 6.5 and indicates that even though the mean yearly air temperature will remain at 2.8°C on 800 level, (at the bottom of heat exchange areas) there will be a larger seasonal variation.

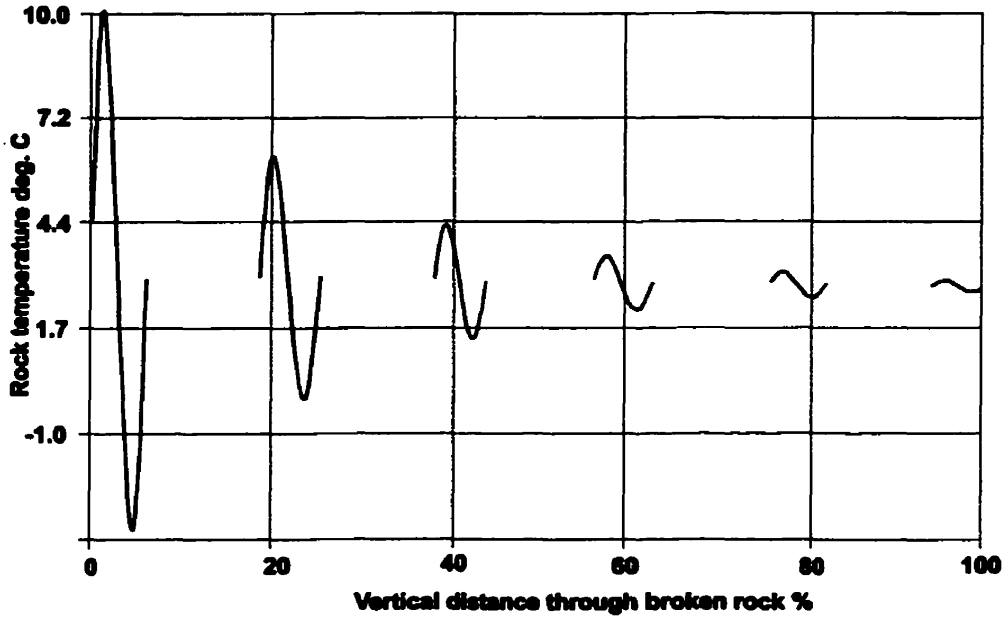

Figure 6.4: Yearly Variation of Rock Temperature Through the Rock Mass

It is certainly rather difficult to come up with a reliable analytical approach of quantifying time-space variations of temperature and thermal transfer for the Creighton heat exchange area. A numerical assessment may be possible but would be hindered by lack of detailed data comprising the variations in non-uniformly fragmented size and voidage that prevails throughout the rock mass. A more workable approach would be to produce empirical relationships encompassing airflow, temperature variations on surface and the 800 level, and time.

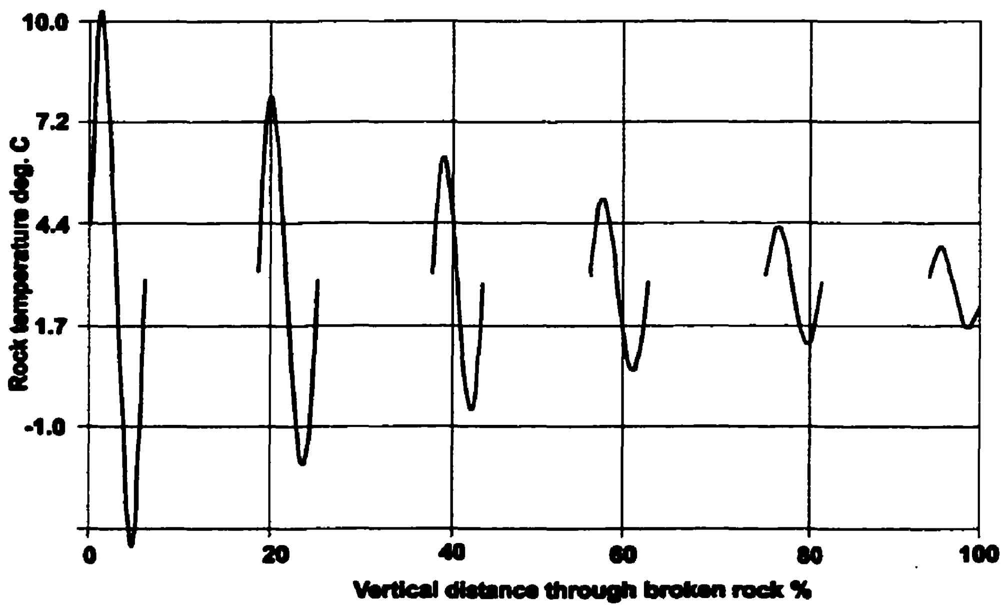

Figure 6.5: Yearly Variation of Rock Temperature Through the Rock Mass With Increased Airflow.

In conclusion, further research into Creighton heat exchange areas could lead into analysis of the heat transfer in a quantitative manner through several numerical models. This type of research could be a subject of a PH-D thesis.

Chapter 7

7.1 REFRIGERATION

7.1.1Introduction

Cooling of the mine environment is required when conventional ventilating techniques do not provide safe and comfortable working temperatures. Mine cooling can be achieved by either natural or mechanical refrigeration methods. The traditional role of mine cooling has been to counteract geothermal heat and the effects of autocompression in deep metal mines. An additional influence has been the escalating amount of mechanization.

One of the earliest methods of temperature control in underground mines was the importation of naturally produced ice from the surface. Blocks of ice were transported in ore cars to cool miners in the Comstock lode under Virginia City in Nevada during the 1860's (McPherson, 1993).

Large centralized refrigeration plants, located underground, became popular in the South African gold mines. Limitation on the heat rejection capacity of return air, combined with the development of energy recovery devices for water pipelines in shafts, and improved "coolth" (chilled water, cooled air) distribution systems, have led to a renewed preference for surface plants.

Until the mid-1970's, centralized plants were usually underground in excavated refrigeration chambers close to shaft bottoms. In South Africa, it was common for large cooling towers to be constructed in, or adjacent to, the upcast shaft bottoms. Chilled water from the centralized underground plant could be transferred to other levels. However, at depth greater than some 500 m below the plant, water pressure in the pipe became excessive.

In the mid-1970's, a number of factors coincided to promote a tendency towards the location of centralized refrigeration plants at the surface of deep mines.

It should be noted that because refrigeration is costly to install and operate, it should only be considered after all possible and practical measures have been exhausted.

7.2 Natural Refrigeration

The simplest cooling method is to deliver cold water from surface into the mine workings to reduce temperatures. This cooling method was implemented in the 19th Century to the Cornish Tin Mines in England and the Comstock Lode in Nevada. At the Gwennap United mines in Cornwall, cold water brought from the surface was poured over the miners they could continue work at a face where fissure water at a temperature of 49°C was intersected. This principle was (Golder Associates, 1980) utilized by Alfred Brandt, who invented a water powered rotary drill in 1876. The drill was used in developing the 19-km long Simplon tunnel between Italy and Switzerland from 1898 to 1905. The rock temperature was 55°C and large water inflows at temperatures of 60°C were encountered. Very large quantities of low temperature water were taken from glacier streams to power three rotary drills, which at the same time cooled the working face. The water released in these systems requires pumping or draining from the workings, resulting in high pumping costs.

To counteract this problem, closed water circuit connected to heat exchangers were developed. Heat exchangers are used to cool the underground air and due to closed circuit water pipes, the pumps are only required to overcome the frictional resistance of the pipes. Unfortunately, this necessitates the use of high pressure piping in deep mines, which increases costs.

A closed loop pipe system of this type has been operating at the Crescent Mine (Hartman, 1982) of the Bunker Hill Company in the Coeur d'Alene district, Idaho since 1967. The system has a cooling rating of 1050 RkW. Eighteen litres per second of water are pumped 1320 m through a horizontal adit and down a 930 m deep shaft in an insulated pipe. The water is then piped 750 m horizontally in another insulated pipe to a cooling coil installation. The water is then returned to surface in a return pipeline. The water pressure at the coils is about 9600 kPa. The cooling coil is comprised of 8 rows of finned horizontal copper tubes having 55 fins per m. The coil is 3 m wide, by 3.6 m high and 0.3 m deep. The surface water has a temperature of 11°C and reaches the coils at a temperature of 12°C; the return water temperature is 28°C. Twenty-two m^3/s of air are cooled from 31.7°C saturated to 23.9°C saturated.

In locations where a low average winter temperature or low relative humidity is encountered, evaporative cooling of the mine water can be utilized. In a system of this type, water or brine is cooled in a surface-cooling tower, and then pumped through an insulated closed loop pipeline to underground heat exchangers and back to the cooling tower. The heat exchangers that are used for air cooling are located in the vicinity of the working areas. The closed pipe circuits produces a balanced hydrostatic head and the pump is only required to counteract the frictional head. In spite of water turbines, pumping cost remains the factor which confines the mass flow of water that can be circulated through the shaft pipelines of deep mines.

7.3 Ice Systems

Latent "Hidden Heat" heat of melting ice is the principle of producing coolth. The use of ice in mines is not new. The first pilot plant using an ice line was constructed in 1982 at Rand Mines in South Africa (McPherson, 1993).

The mass flow of water circulated around a mine for cooling purposes can be diminished by a factor of over 5, if delivered in the form of ice.

There are four principle items to consider when investigating an ice concept for mine cooling:

1. The large scale manufacture of ice
2. The transportation to underground mines
3. How it is best incorporated into a mine cooling system
4. The economics of the system.

7.3.1 Manufacture of Ice

1. Either as particulate ice at subzero temperature, or
2. Slurry of ice crystals within liquid water.

As an example of large-scale particulate ice manufacture, the plant at East Rand Mines Figure 7.1 (South Africa) produces 6000 tons of ice per day from six 1000 tons/day units. Four uninsulated pipes are used. Each pipe is capable of carrying 200 tons/hour.

- The particulate ice falls through shaft pipeline and is delivered into an ice-water-mixing dam.
- Water from hot water dam is also sprayed into the ice silo.

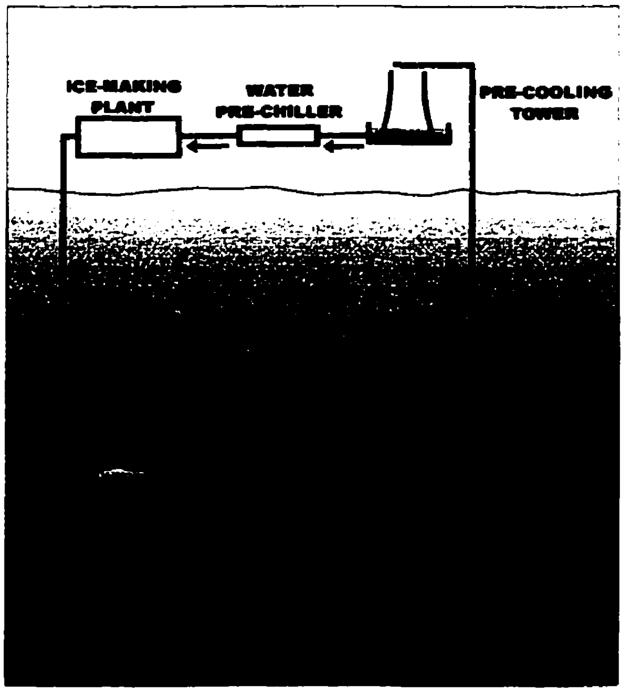

Figure 7.1: Layout of an Ice System (After McPherson, 1993)

7.3.2 Economics of Ice System

1. Large reduction of water flow in shaft pipelines (therefore smaller pipes and lower pumping lost).

2. Underground water from melting ice is at 0°C rather than 3 to 6°C common with conventional chilled water systems (improved performance of heat exchangers).

3. The underground system is simpler (no need for turbines).

The major disadvantage of the system is the considerable increase in both capital and operating costs of the icemaking plant.

7.4 Ice Stopes

This is an economical technique that allows air to be heated to a temperature approaching 0°C at low cost.

7.4.1 Principle of Operation

Water is sprayed into the top of the open stope. Cold air enters the top of the stope, freezing water. The air is heated by three mechanisms:

1. Directly from water droplets
2. By latent heat of fusion
3. By strata heat.

In summer, warm intake air is cooled through the same stopes and melting ice. However, the air-cooling is far from being efficient due to the fact that there is not an "intimate" contact (heat exchange) between ice and air.

7.5 Mechanical Refrigeration

Mechanical refrigeration is the process of absorption of heat from one location and its subsequent transfer to and rejection at another location. As it applies to the cooling of mines, it can be described as removing heat from the intake air at one point, and "pumping" that heat to another "point", where it is released either into the return air as a water cooling system.

A refrigerator is a mechanism, which makes it possible for heat to be transferred from a cooler medium to a warmer medium. This is the opposite direction to the usual flow of heat.

Mechanical refrigeration systems can be classified in several ways and can be combined into one of three groups.

1. Large surface plants
2. Large underground plants
3. Underground spot coolers.

The above three groups normally use vapor-compression machines where refrigerant flows from one part of the circuit to the other, it is first compressed and cooled forming a liquid and later it is expanded and heated forming a vapor.

We note that a change of state is used because latent heat is much greater than sensible heat in a liquid. Also, when a liquid is converted to vapor, a large amount of latent heat must be added to it (without the temperature changing). Similarly, when a vapor condenses to a liquid, an equivalent amount of latent heat must be removed from it.

The principles of refrigeration can be best described by the vapor-cycle.

7.5.1 Basic Vapor Cycle

A refrigeration system consists of a cycle of four basic processes (Hartman, 1982) circulating a refrigerant, the heat transfer medium. The purpose of the refrigerant is to absorb heat from a "source" (evaporation) and discharge it through a "sink" (condenser). Some type of vapor pump must be located between the source and sink so that the energy absorbed by the refrigerant in the evaporation may be transferred to the condenser for discharge; it takes the form of a compressor. The final component of a vapor refrigeration system is an expansion valve, used to control flow rate and permit cooling of the refrigerant in its return to the evaporation. Thus vapor refrigeration is essentially a compression system, involving heat exchange through a change of state of the refrigerant from liquid to gas and then back to liquid.

A vapour refrigeration cycle in schematic is represented by the diagram in Figure 7.2.

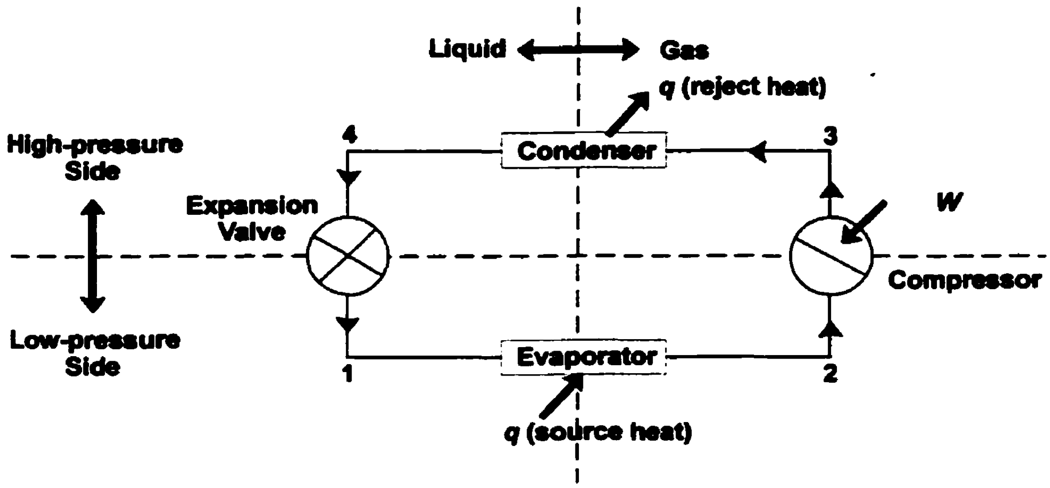

Figure 7.2: Schematic and Block Diagram of Vapour Refrigeration System Using a Vapor Cycle (After Hartman, 1982)

The path of flow of the refrigerant and the changes it undergoes can be traced through the system, as follows:

1. Evaporator: The refrigerant "boils" (evaporates), changing state from predominantly liquid to gas and absorbing heat from the substance to be cooled with no change in temperature.

2. Compressor: In the vapor state, the refrigerant flows to the compressor, where work is done in compressing it.

3. Condenser: Vapor condensed to liquid again, giving up heat without a temperature change.

4. Expansion Valve: The temperature and pressure of the liquid drops during expansion, as the refrigerant completes the cycle.

The state of the refrigerant, liquid or gas, is indicated in Figure 7.2. Notice that the changes of state of the refrigerant are controlled by varying the pressure. This leads to the designation of a low-pressure side and a high-pressure side of the system.

The refrigerant must have chemical, physical and thermodynamic properties that make it both safe and economical to use. No refrigerant exists that is ideal for all situations. Ammonia is a very good refrigerant because it has the highest refrigerating effect per unit weight of any refrigerant, but it cannot be employed underground because of its toxicity. Refrigerants 11, 12 and 22 were usually used underground.

7.5.2 Practical Applications

As will be noted later, using refrigeration to cool mine air, there exist some practical arrangement, which have become accepted. They are as follows:

The refrigerant is usually not used to cool the air directly; rather, it cools a separate water circuit. This cooling is done in a heat exchanger, in which the refrigerant flows in a series of coils immersed in the water circuit (Le Roux, 1990).

The coolth in the refrigerant is transferred to water, which is chilled. This chilled water is then used to cool the air stream, in one of two ways.

1. The water can be sprayed into the air stream through a large number of spray heads, producing fine droplets. The air is cooled and the warmed water is collected in a sump and pumped back to the cooling plant to be used again.

2. To pipe the chilled water to the working places in insulated pipes. Here the chilled water is passed through a series of coils and the air is cooled by big fans over the pipes.

7.5.3 Large Surface Refrigeration Plants

These machines can be arranged to cool air entering the mine and/or water for mine use. Most arrangements cool water in the evaporator and air is normally cooled by water sprays or by water/air heat exchangers commonly known as cooling coils (Golder Associates, 1980).

Early installations of this type were used for air cooling at Morro Velho, Brazil; Robinson Deep and East Rand Proprietary Mines, South Africa; Oorgam Mine, India and Rieu du Coeur Mine, Belgium. These installations used both ammonia and freon refrigerants and cooled air with water or brine circulated through sprays or heat exchangers.

The advantages of good maintenance and easy heat rejection for these installations were neutralized by other factors and no installations of this type have been constructed since the early 1950's. The positional efficiency of these machines, defined as the cooling at the working stated as a percentage of the cooling produced by the plant, was very low. This occurred because of the heating of the air by auto-compression in the shafts and increased heat flow from the strata in intake airways due to the increased temperature difference between wallrock and air. Consequently, by the time the air reached the working faces it had lost most of its cooling capacity.

In locations where low winter air temperatures is encountered on surface, these installations had to be stopped to prevent icing, eventhough the mine could still have underground workings which required cooling. It has been reported that the Robinson Deep South Africa installation was operated for only 3 months of the year.

Consideration was then given to cooling water on the surface and sending it through a closed pipe system to underground air-cooling coils. The limiting constituent for this method was the water pressure in the pipe circuits and coils. A system of this type was installed at the Anaconda Mining Company's Steward Mine in Butte, Montana. The pipelines were designed from grade B low carbon steel weldless tubing of sufficient wall thickness to stand a water head of 1500 m, with a minimum safety factor of 4. A 2900 RkW machine was installed in series with an evaporative cooling tower having a capacity of 4300 RkW. Total underground cooling at the workings was 6400 RkW giving a positional efficiency of almost 90 per cent. The amount of water circulated was 110 litres per second and it entered the mine at 4.4°C and was returned to the cooling tower at 20°C.

At the Zwartberg Colliery, Belgium, an endeavor to combat the disadvantages of high-pressure piping was made by incorporating a Pelton wheel turbine for energy recovery in the downgoing water pipe. This system provided 2400 RkW of cooling on surface for a water flow rate of 33 litres per second. The turbine was connected to a pump for removing the water from the mine.

This system was still fundamentally a closed loop system and it was calculated that for mines exceeding 1000 m in depth, it was less expensive to reduce the water pressure by installing a high to low pressure water/water heat exchanger of the shell and tube type. An installation of this type was installed at Rieu du Coeur Mine but it was found indispensable to use brine instead of water in the primary circuit in order to get acceptable temperatures in the low-pressure water circuit. A similar system was installed at Magma Copper, Arizona and at the Mount Isa Mine, Australia.

Subsequent to these installations, majority of refrigeration plants were installed in South Africa and underground plants were favored. In the mid 1970's, the work of Whillier and Van der Walt at the Chamber of Mines of South Africa resulted in cooling mine service and drilling water on the surface. The primary reason for this was the issue of heat rejection at underground cooling towers.

Due to constraints, it is clear that the availability of cool air on surface for heat rejection is the key reason for the trend to surface positioned water-cooling plants in the South African mining industry. Considerations of service water cooling, pipe sizing and energy recovery have also impacted on the increasing popularity of this type of installation.

7.5.4 Surface Plant

The ease of:
1) Maintenance
2) Construction, and
3) Heat Rejection,

combined with the use of energy recovery devices, have led to a preference for the main refrigeration units to be located on surface.

The Figure 7.3 shows the simplest system that employs a surface plant.

Chilled water from the evaporator passes down an insulated pipe in the shaft to a high-pressure water-to-water heat exchanger at one or more shaft stations. (This heat exchanger is a source of potential loss of system efficiency – problem of corrosion, scaling and erosion).

Note: Should the heat problem exist during the summer months only, then all that may be necessary is to install a surface bulk air cooling system and to utilize it during the warm season.

However, if the workings are far away from the shaft bottoms, and the heat load is concentrated heavily within those workings, then bulk air cooling on surface will have greatly reduced effectiveness.

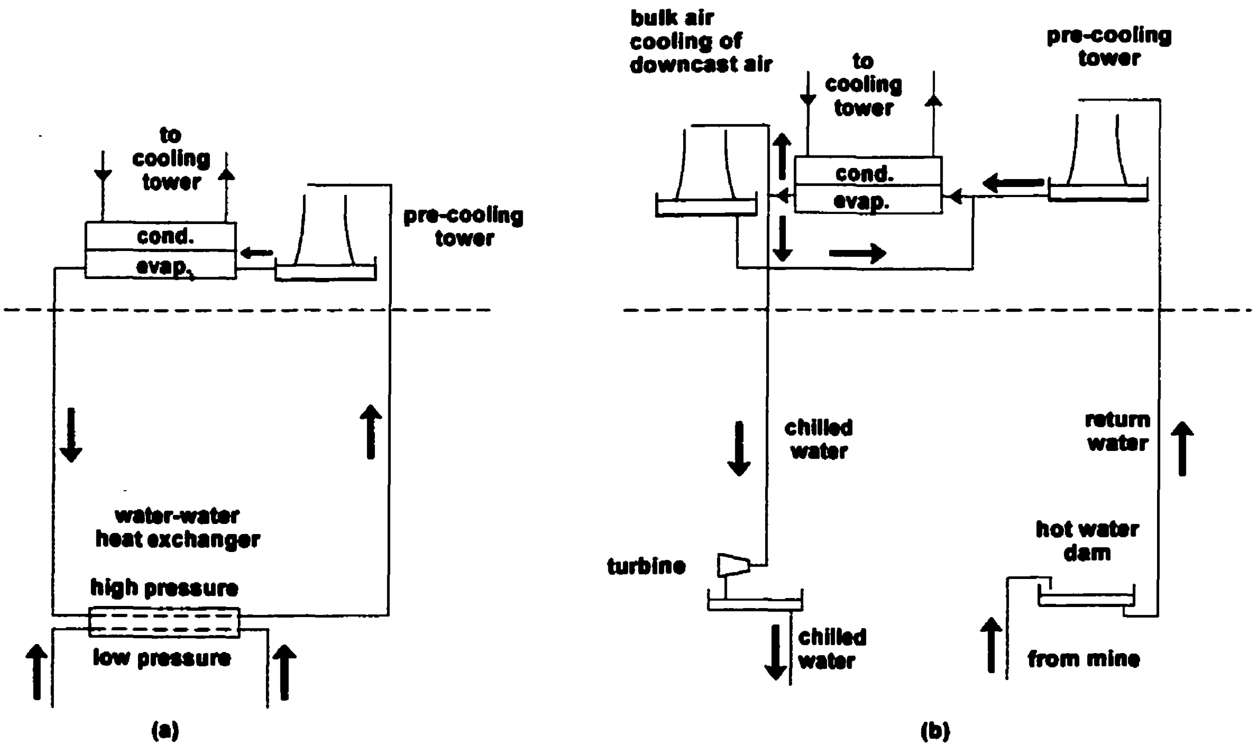

Figure 7.3: Examples of System Configurations Using Surface Refrigeration Plant (After McPherson, 1993)

7.5.5 Large Underground Refrigeration Plants

Large underground plants for air-cooling have been used widely in South Africa and some 225 RMW of this type of cooling was installed by 1975. These are normally over 1000 RkW capacity. A German installation of this type was commissioned at Consolidation Colliery in 1977, Hamm (1979).

The main advantage in locating refrigeration plant underground is that the length and cost of the water pipe reticulation system is reduced largely and much lower pressures are encountered in the pipe system.

Recent South African practice has been to turn some of these existing plants over to cooling service water, Whillier (1978) in place of air cooling.

The major factor curtailing the use of underground plant is the heat rejection capacity available to the condenser. Condenser water is usually potable mine water or mine service water. The water quality has to be controlled as far as possible or excessive corrosion and fouling of the condenser tubes occurs.

Condenser heat can be transferred by using water on a once-through basis and pumping it out of the mine or by transferring the heat to the mine exhaust air using one of the three following systems:

- Vertical cooling towers
- Single or multi-stage spray banks in horizontal or inclined airways
- Closed water to air heat exchangers made from finned tube or plate and tube with the water circulated through the tubes.

Water circulation is of the order of 30 litres per second per RMW of refrigeration. Pumping condenser water to surface is therefore only practical for smaller units. Closed heat exchangers have the same volume limitation and large plants normally use cooling towers or spray banks for condenser heat rejection. Nevertheless, the German Consolidation installation employed direct air cooling of the refrigerant at the condenser for a cooling capacity of 1760 RkW. The air cooled condensers are very prone to blockage by mine dust and consideration should be given to this factor when selecting these units for hard rock mines where blasting is commonly used.

The Figure 7.4 below shows the principles of the system preferred by the South African gold mines prior to the 1980's.

1) The refrigeration plant is located underground.

2) The advantages of the system are that it eliminates the necessity for surface-connecting pipes and the accrued pumping costs.

3) The principal disadvantage of the system is that its duty is confined by the capacity of the return air to absorb heat rejected in the underground cooling towers.

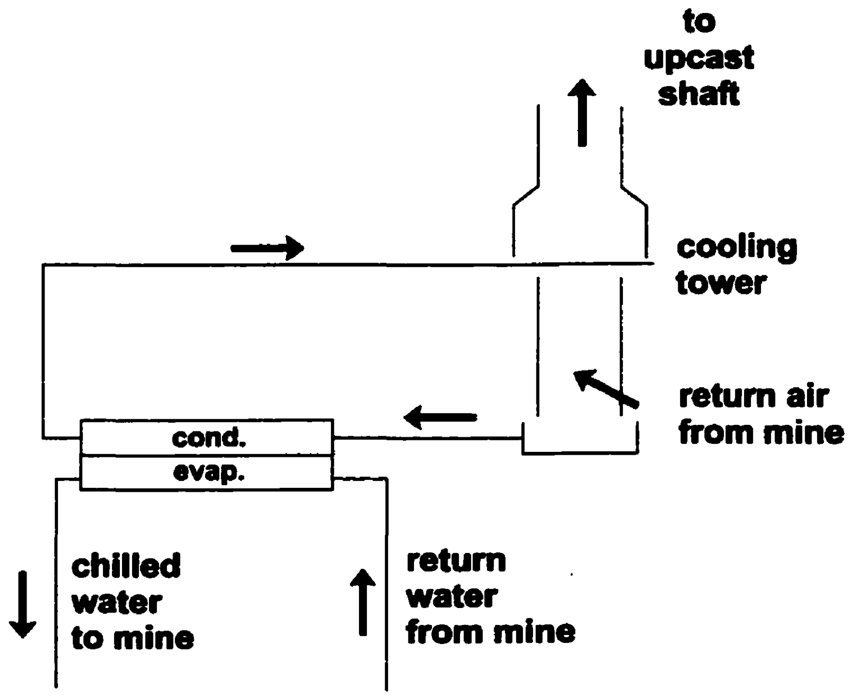

Figure 7.4: System Configuration for Centralized Underground Plant (After McPherson, 1993)

7.6 Underground Spot Coolers

Underground spot coolers are small refrigeration plants commonly having up to 1000 RkW (300 ice tons) cooling capacity. They are portable and are used for cooling small or isolated working areas of a mine. The largest installations of this type are used in the Ruhr coal mines of West Germany (Hamm, 1979). At that location the installed capacity has increased from 10 RMW (35,000 ice tons) in 1970 to 97 RMW (340,000 ice tons) in 1978. The majority of these plants have a capacity of 100 to 600 RkW (30 to 170 ice tons) and are equipped with reciprocating compressors. The most notable features are their relative compactness and portability. They are usually equipped with a so-called "back-cooler". A "back-cooler" is a plate and tube heat exchanger, which transfers the condenser water heat to the return air. This gives a closed condenser water circuit and reduces the amount of water required for heat rejection.

Spot coolers are also used at the Hecla Mining Company's Star Mine, Idaho. These machines are described in detail by Marks (1979). They are placed in close proximity to the working face in order to optimize positional efficiency. Heat is rejected via the mine service water system, which is pumped to surface.

A 530 RkW plant of this type has been utilized in the Champion Reef Mine, Kolar Goldfield, Caw (1968, 1959). They have also been utilized in tandem with tunnel boring experiments at Libanon and Free State Geduld Mines, South Africa, Hall (1979). Other units have been utilized in Zambia, Nelson (1957) and for civil tunnel projects.

7.6.1 Performance and Efficiency

The efficiency of a cooling plant can be expressed by: the coefficient of performance, and the positional efficiency.

The Coefficient of performance C.O.P. shows how efficiently the plant operates i.e. how much heat is removed at the plant for a given expenditure of energy. It is defined as: heat extracted on the evaporator. Heat equivalent of the power input to the plant.

$$\text{C.O.P.} = \frac{T_1}{T_2 - T_1}$$

The theoretical maximum C.O.P of a cooling plant is given by the Carnot Cycle C.O.P.. This is defined as:

$$\text{Carnot C.O.P.} = \frac{T_1}{T_2 - T_1}$$

T_1 = Evaporation Temperature
T_2 = Condensation Temperature

Naturally, this theoretical value is never obtained in practice.

Tables 7.1 and 7.2 show typical dimensions and data for South African mines.

Table 7.1: Typical Dimension and Performance Data of Stope Heat Exchangers in Use On South African Gold Mines (After Environmental Engineering in South African Mines, 1982)

Face Area	0.9 m x 0.6 m
Depth (in Direction of Air Flow)	457 mm
Water Tube Diameter -External - Internal	19 mm 16 mm
Spacing Between Tubes	38 mm
Spacing Between Fins	5.0 mm

Water Flow	227 l/min
Air Flow	2.83 m^3/s
Cooling Duty	(105 kW)
Typical Water Temp - In	17°C
- Out	24°C
Typical Air Temp - In	32°C
- Out	25°C

Table 7.2: Typical Dimension and Performance Data For Underground Cooling Tower (After Environmental Engineering in South African Mines, 1982)

Cross-Sectional Area	14 – 23 m^2
Height	18.3 m
Water Flow	6363 – 9090 l/minute
Typical Water Temp - In	37.8 – 46.7°C
- Out	34.4 – 40°C
Typical Air Temp - In	31 – 35.6°C Wet Bulb
- Out	35 – 37.8°C Wet Bulb

7.6.2The Positional Efficiency

Positional efficiency is an indication of how effectively a plant cools the air at the location where the cooled air is utilized to ventilate the workers.

In many underground applications, the cooling plant may be located too far away from the working place where the cooled air is to be used.

For example, a surface cooling plant may well cool air down to nearly 32°F, however, if this air is then sent down a deep shaft and through long airways, in which it ultimately picks up heat in both, it may then reach the workplace at a much higher temperature.

Because the cool air is at a much lower temperature than the rock through which it travels, there will be much more heat transfer into this air than there would be into the warmer air taken directly from the atmosphere.

In situ (South Africa), it has been found that there is only a marginal cooling of the air temperature in the workplaces when surface air cooling is utilized.

As an example of positional efficiency take a cooling plant that extracts 35 kJ/kg of air at the refrigeration plant, but the decrease in a particular stope is only 10.5 kJ/kg of air, the positional efficiency is:

$$\frac{10.5}{35} \times 100 = 30\%$$

7.7 Distribution Systems

Cooling capacity may be transferred and distributed in mine workings using either water or air as the transfer medium. Air circulation is traditionally used for localized cooling of the working face. Use of air circulation over long distances however, results in increased heat transfer from the strata and reduced positional efficiency as illustrated previously.

Water distribution systems are more effective over long distances and in such cases, it is important to size and insulate the pipe adequately. Whillier and Van der Walt (1977) have provided guidelines for selecting the correct pipe size to prevent air entrainment and the resulting blockages in gravity fed, vertical pipe systems.

Insulation of pipe circuits is paramount and Whillier (1974) has indicated that the heat gained by a 150 mm diameter steel pipe in typical hot underground conditions is reduced from 472 Watts per m of length to 11.4 Watts per m by the addition of a 30 mm thick insulation covering. Torrance (1962) described a method of calculating the economic benefits of insulating pipe circuits and although it is expensive, it has a short payback period. Insulation thicknesses used in South African gold mines are given in Table 7.3.

Table 7.3: Typical Pipe Insulation Details (After Torrance, 1962)

Pipe diameter, mm	Insulation thickness, mm
100	25
150	25 - 28
200	25 - 50
250	25 - 40
300	25 - 40
500	38

According to Howes (1975), the types of insulation used were: Polystyrene 59 percent, Polyurethane 39 percent, Glass Fibre 2 percent.

It was found that pipe flanges were only insulated in 30 percent of installations. In the advent of increasing energy costs and an improved appreciation of the heat pick-up at these locations has resulted in most flanges being insulated in most recent installations. Several factors should be considered in selecting insulation, the resistance to heat transfer is important and two other factors have received considerable attention. The first is to recognize that resistance to heat transfer is normally provided by still air in the cells of the insulation material. This resistance is diminished if the insulation becomes impregnated with water, which replaces the still air and consequently increasing its conductivity. Therefore, a good insulation material will have strong water resistance capacity and adequate vapour barriers. The second factor concerns the products of combustion of the insulating materials resulting from fires. Both these factors have been examined, and reported on in detail by Smit and Kirkman (1979).

The piping systems used must be adequately designed to withstand the static water heads with acceptable factors of safety. In installations where the static pressures have been too high, water pressures have been reduced by installing high pressure to low pressure water to water shell and tube heat exchangers. The main disadvantage of these units is that with high-pressure primary water at 8°C, it is impossible to reduce the secondary circuit water leaving temperature below 15°C. Other disadvantages are the high maintenance costs and excessive fouling. For these reasons, these exchangers have been eliminated from water circuits.

7.8 Heat Exchange Systems

Air is cooled by chilled water in two ways by: (a) indirect cooling methods, or (b) direct contact cooling methods.

7.8.1 Indirect Cooling Methods

Most indirect cooling installations use heat exchangers similar to tube and fin radiators, e.g. South African and American practice. In West Germany the loss of heat transfer due to contamination on finned tubes has encouraged research into other designs and as a result strip-tube and plate-tube coolers are now used exclusively.

Strip-tube coolers consist of parallel and horizontal, spiral tubes of copper with vertical copper strips to increase the heat exchange surface. Air flows longitudinally along the tubes resulting in partial counter flow and parallel flow with the water.

Plate-tube coolers also use spiral cooper tubes but in the vertical direction. The tubes are pressed and soldered onto copper sheet plates to increase the heat transfer area. The cooler configuration results in a crossflow of the air and water.

7.8.2 Direct Contact Air Cooling Method

Direct contact cooling consists of spraying chilled water directly into the path of flowing air. Large-scale installations spray the water in multiple stages, arranged in counterflow to the airflow. Pretorius (1978) details the performance of the three stage cooling pond sprays at the Loraine Mine, South Africa.

The challenge with spray arrangements in deep mines is that the hydraulic heads are not balanced and the water has to be pumped back to surface. This can be partially overcome by installing an energy recovery system in the circuit. The economic advantages of sprays over cooling coils coupled with the reduced piping costs, have made spray systems attractive (no return water line is required).

The use of sprays in place of cooling coils has created a demand for small capacity units and work has been done in the United States and in South Africa to develop suitable systems. The U.S. Bureau of Mines has published details of a trial unit, which achieved a heat transfer of 22 RkW during underground testing at the Homestake Mine, South Dakota, Thimons et al. (1979). Venturi sprays have been tested in South Africa and it was found that the performance of these sprays can reach 400 RkW.

Open water circuits cannot however be utilized in mines where water will effect surrounding strata. Potash mines and mines with a clay roof or floor that require water control will continue to use indirect cooling to prevent strata deformation. Other mines are likely to use more service water cooling and direct contact sprays in the future.

Chapter 8

8.1 NATURAL SOURCE OF REFRIGERATION VERSUS MECHANICAL REFRIGERATION FOR MINING BELOW 7000 LEVEL AT CREIGHTON

8.1.1 Introduction

In the previous Chapters, an empirical formula was given to calculate the heat load of the mine.

The heat load assessment methodology used at Creighton takes into consideration the following data, the daily tonnage to be mined, the difference between the virgin rock temperature and the fresh air station wet bulb, both at the mean mining depth, and a factor that is determined from the heat load of past measurements and current mining operations. The factor is applied together with the predicted temperatures to determine future heat loads at deeper levels.

Over the last two decades, numerous checks were made at Creighton in order to quantify the heat load of the mine, and the sources of that heat. Table 8.1 provides comparison of various heat sources measured in years 1979 and 1993. The total heat extracted for the mine in 1993, amounted to 14.1 MW, and 8.2 MW in 1979 (Appendix 2).

	1979 MW	1993 MW	Increase MW	Percent Increase
Whole Mine	8.21	14.07	5.86	71
Diesel Equipment	0.80	1.50	0.70	87
Electric Load Centers	0.18	0.55	0.37	198
Fans	2.55	3.87	1.32	52
Compressors		2.97	2.97	n/a
Other Identified Sources (net)	0.05	0.12	0.07	141
Strata Heat (by difference)	4.63	5.06	0.43	9

Table 8.1: Heat Load Comparison

Separate evaluations were also made in 1993 and 1979 (Appendix 2) for the contributions of fans, compressors, diesel and electric equipment, body metabolism, cement in the sandfill pumps and the cooling effect of water supplied to the mine.

The net value of these sources in 1993 was (9.0 MW). The remainder (5.1 MW), was assumed to be strata heat, whereas 3.6 MW, and 4.6 MW respectively in 1979.

The evaluation of 1993 heat sources, production parameters, virgin rock, fresh air station temperatures for various mining depth, produced the following heat load factors Table 8.2.

	Heat Load Factor kW/tonne/day per °C
Whole Mine	0.098
Levels 4600 to 7200	0.088
Levels 6200 to 7200	0.084

Table 8.2: Evaluation of 1993 Heat Sources

The values obtained showed a significant increase between years 1979 and 1993. The dominant increases in heat load were due to equipment rather than the strata. The mining methods had also changed significantly between 1979 and 1993. If the vertical retreat method of mining, and the type, capacity and distribution of equipment remains unchanged in the planned future workings then the value of heat load factor established in 1993 should give useful estimates of the corresponding heat loads. However, it is prudent to check the heat load factor as the mine continues to deepen.

8.2 Airflow Requirements

Operating diesel equipment results in the generation of heat, gases and diesel particulate matter.

Nearly all of the caloric value of the diesel fuel consumed by equipment appear in the form of heat. It is common practice to establish airflow requirements based on the quantity of air necessary to adequately dilute diesel exhaust emissions and dispels them from the workings. A dilution factor for the mine used at Inco is about 7.9 m^3/s per 100 kW of equipment used.

For the Creighton analyses, the dilution of diesel emission is not the only critical factor, but principally the removal of heat from the workings.

If mechanical refrigeration is not used in the Creighton Mine, it is then necessary to pass enough airflow through the workings to remove the heat. The principal criterion used for determining the airflow requirement is that, the airflow temperature in the workings is in the range of 24°C, as discussed in previous chapters.

The predicted heat load and airflow requirements as the mine progresses to greater depths is shown in Table 8.3. The cursory examination of Table 8.3 indicates that the required airflow becomes excessive at mean working depths greater than 8400 Level if cooling is to be achieved by ventilation alone.

Mean Mining Depth Level	(m)	VRT °C	Station Intake Temp. W.B. °C	Barom Press. KPa	ΔΣ Heat for 24°C Stope Exhaust and Fresh Air Temp kJ/kg	Heat Load kW	Volume m^3/s of Air at Density 1.26 kg/m^3	Volume m^3/s per tonne/day of Air at Density 1.26 kg/m^3
7000	2134	42.2	17.2	121.9	19.5	5556	226	0.100
7200	2195	43.3	17.7	122.6	18.6	5689	243	0.107
7400	2256	44.4	18.1	123.3	17.4	5845	266	0.117
7600	2317	45.6	18.6	123.9	15.4	6001	309	0.136
7800	2378	46.7	19.0	124.6	14.2	6157	344	0.152
8000	2439	47.8	19.5	125.3	13.0	6290	384	0.169
8200	2500	48.9	20.0	126.0	11.6	6423	439	0.194
8400	2560	50.0	20.4	126.7	10.7	6579	488	0.215
9000	2744	53.3	21.8	128.7	6.5	7001	854	0.377

Table 8.3: Predicted heat loads at lower level stations Creighton Mine for 2268 tonne/day, heat factor 0.098

The above Table is based on field measurements taken at Creighton Mine over several months in the 1993 survey (Appendix 2) and the use of the empirical approach to heat load assessment, namely,

Heat Load (kW) = Heat Load Factor x tonnes/day x (VRT – tw,in)

Air Mass Flow Requirement = Total Heat Load [kw]/(Σout-Σin) [kJ/kg]

Where:

HLF = Inco Ltd. Heat Load Factor (kW/tpd/°C)

tonnes/day	=	Production Tonnage for a 24 hour period
VRT	=	Virgin Rock Temperature at Relevant Level (°C)
tw,in	=	Wet Bulb Temperature (°C)
Σ	=	Sigma Heat for the Air (kJ/kg)

8.3 Estimation Of Refrigeration Requirements For Creighton Mine

In order to determine whether mechanical refrigeration is a viable option for the Creighton Mine, a detail costing analysis would need to be conducted, coupled with ventilation network analysis, which is outside of the scope of this report. Nevertheless an Order of Magnitude Capital and Operating Cost for the installation and operation of refrigeration system is provided and compared with using of natural heat exchanger as the source of cooling, as described in Chapter 6.

It is estimated that ultimately some (2000 refrigeration ton), 7034 RkW range of 5200-8800 RkW. capacity would be required between 7200 and 8200 Level, to cool incoming fresh air to 15.6°C W.B. range 12.8 – 18.3°C W.B.

Example of refrigeration requirements for 8200 Level (2500 m below surface), based on the following parameters:

VRT	= 49°C, Station intake temperature 20°C, W.B.
Production Rate	= 2268 tonne/day, and heat load factor = 0.098
Assumptions:	- Station intake temperature is to be cooled to 17.2°C W.B.
	- Refrigeration positional efficiency = 60%
Determine	- Refrigeration heat load, and volume of airflow to dissipate heat load

Assessment of Heat Load

By applying empirical equation described previously namely, heat load (kW) = heat load factor X tonnes/day X (VRT – tw,in)

Where in these units, temperatures are in °C and the measured heat load factor is 0.098 kW per tonne/day per °C.

Substituting we obtain:

$$0.098 \frac{\text{kW}}{\text{tonne/day per °C}} \text{ X } 2268 \text{ tonne/day} \times (49 - 17.2)\ °\text{C} = 7068 \text{ kW}$$

The issue now is to dissipate or absorb this heat load by the fresh air at the temperature of 17.2°C W.B., until it heats up to cut off designed stope temperature of 24°C W.B.

The amount of air required to absorb estimated heat load of 7068 kW produced by mining operation is calculated as follows:

7068 kW : (Σ 78.85 - Σ 59.78) (kJ/kg) = 371 kg/s of air

Where: 78.85 kJ/kg = Σ Heat of Air at 24°C W.B.
59.78 kJ/kg = Σ Heat of Air at 17.2°C W.B.

Where: 371 kg/s of air = 294 m^3/s at 1.26 kg/m^3 density.

The total airflow requirements for mining operation at 8200 Level comprises the following components:

(294 m^3/s	+ 30 m^3/s)	+ 10% = 356 m^3/s
(as above)	(airflow for garage)	

As indicated previously, the incoming fresh air temperature is 20°C W.B. on 8200 Level. Therefore, 356 m^3/s need to be cooled from 20°C to 17.2°C W.B.

The amount of cooling required:

$$356 \frac{m^3}{s} \text{ X } 1.26 \text{ kg/m}^3 \text{ X } (\Sigma\, 67.2 - \Sigma\, 59.8)\ [\text{kJ/kg}]$$

= 3319.3 kW ; 3319.3 : 0.6 = 5532.2 RkW installed refrigeration kW
or 1573 refrigeration tons

Where: 67.2 kJ/kg = Σ Heat of air at 20°C, W.B.
59.8 kJ/kg = Σ Heat of air at 17.2°C, W.B.
0.6 = Refrigeration positional efficiency of 60%

Capital Cost

1573 refrigeration ton x 4250 \$/Rt ~ \$6,685,000

Note: 1 refrigeration ton = 4250\$

Power Cost

2 hp/refrigeration ton, 400 \$/hp – yr

1573 refrigeration ton x 2 hp/refrigeration ton x 400 \$/hp ~ 1,258,000/year

Whereas: Capital Cost of 2000 R.T. = \$8,500,000 and Operating Cost = \$1,600,000 \$/year

By using the open pit caved area as per Chapter 6 as a natural source of refrigeration, the total airflow requirement for the mine is in the range of 710 m^3/s of which some 439 m^3/s is required for mining below 7000 Level mean mining depth 8200 L, to remove heat load as per Table 8.3. This compares with 565 m^3/s at 1.26 kg/m^3 total mine requirement for mechanical refrigeration case, where the airflow of 356 m^3/s destined for mining below 7000 Level, mean mining depth of 8200 Level would need to be cooled.

As can be deduced from the above it is possible to dispense with underground refrigeration by providing extra 83 m^3/s below 7000 Level.

Mechanical refrigeration therefore reduces airflow requirements below 7000 Level from some 439 m^3/s to 356 m^3/s. The currently driven at Creighton new fresh air system would still be required, but could be downsized. Offsetting this downsizing is an extra of some \$8,500,000 capital for the refrigeration plants. The operating cost of refrigeration in the range of \$1,600,000 – 2,000,000 per annum, while operating cost of fan power requirement for non refrigeration case amounts to about 75% of the above cost.

8.4 The Review of Current – 1999 Ventilation Expansion

8.4.1Basis For Designing the Current Ventilation System

As mining depths increase, large quantities of air are required to remove heat from a mine. There is a maximum working temperature and enough heat must be removed to maintain this temperature. The fresh air supply becomes warmer with increasing depth due to autocompression, and thus can absorb less heat (produced by mining operations, etc.) or remove heat. Additionally, air at constant wet-bulb temperature can absorb less heat per kg of air. These facts are illustrated in Tables 8.4, 8.5 and Figure 8.1.

The major ventilation expansion that is currently taking place at Creighton Mine, comprising of the development of third fresh air system which will provide additional 280 m^3/s at the density of 1.26 kg/m^3, the primary booster fans installation, and the replacement of the main exhaust surface fans installation has been in planning stages for the last number of years.

The airflow requirement of some 490 m^3/s Table 8.3 for mining below 7000 Level (7000 – 8400 Level) was based on extensive heat load studies and assessment done in the late 70's (Coulter et al 1978, Stachulak 1979) and large field data investigation, carried out in 1993, Table 8.3, Appendix 2, and climatic computation – computer program, Climsim (McPherson, 1996), was employed to check and further validate the heat load assessment and airflow requirements. These simulations outside of the scope of the report were in close agreement with empirical method used.

8.5 The Current Challenges and Observations

The additional heat load measurements conducted during the period of this year (1999) revealed that the fresh air coming to 7000 Level has increased by 2°C, since 1993 Table 8.4, and is now 19.1°C, similarly heat load factor has increased during that same period and is now 0.14 versus 0.098 in 1993 which amounts to some 43% rise.

The analysis of Table 8.4 illustrates that if the above parameters i.e. 2°C temperature of fresh air increases between levels 7000 – 8400, and 43% rise in heat load factor would become reality then calculated airflow of 488 m^3/s for 8400 level 2560 m mean mining depth cannot cool the workplace to designed temperature of 24°C. The airflow would have to be increased to 1517 m3/s to maintain the designed temperature as shown in Table 8.4 and Figure 8.1. Such enormous increase in airflow would not be possible. A possible option would be to allow the temperature in the workplace to rise above 26°C W.B., and then the volume requirement would drop drastically from 1517 to 587 m^3/s as depicted in Table 8.4 and Figure 8.1. This temperature may, however, be not acceptable as discussed in Chapter 4. The only viable option then would be mechanical refrigeration.

8.6 Conclusion

It is clear that temperature of fresh air coming to 7000 L (2137 m, mining depth) must be reduced by means of closer control of 800 Level gathering area's natural heat exchange, and water elimination from fresh air.

The current 43% rise in heat load factor (0.014 versus 0.098 kW per tonne/day per °C in 1993) is a function of high degree of mechanization, elevated fresh air temperature, and possibly temporary deviation from one pass ventilation. This in turn directly impacts on extensive reduction in heat removal capacity of fresh air as documented in Table 8.5, which shows 28 – 59% decrease in heat removal capacity of air for various mean mining depth, namely 7000 to 8400 Level.

In summary, the calculated airflow requirement of 490 m^3/s for mining up to 8400 level will remove heat load produced by mining operation, and maintain

temperature of 24°C W.B. in the workplace only if current fresh air intake temperature and heat load factors are reduced to original 1993 values. However, if this does not take place, then either temperature in workplace is allowed to increase to the range of 26°C W.B., at the expense of additional airflow requirement as indicated on Figure 8.1, and Table 8.4, or installation of mechanical refrigeration at lower levels will become necessity.

		1993 Heat Load Factor 0.098 Production 2268 tonne/day					1999 Heat Load Factor 0.14 Production 2268 tonne/day						
					Airflow at 1.26 kg/m^3					Airflow at 1.26 kg/m^3		Percent Increase	
Level	Mean Mining Depth (m)	Stat'n W.B. Temp °C	Stope Exhst W.B. Temp °C	Heat Load kW	m^3/s	m^3/s per tonne/ day	Stat'n W.B. Temp °C	Stope Exh. Temp °C	Heat Load kW	m^3/s	m^3/s per tonne/ day	Heat Load	Airflow
7000	2134	17.2	24	5556	226	.100	19.1	24	7139	404	0.178	28	79
								25		315	0.139		
								26		258	0.114		
7200	2195	17.7	24	5689	243	.107	19.6	24	7315	444	0.196	29	83
								25		344	0.152		
								26		273	0.120		
7400	2256	18.1	24	5845	266	.117	20.1	24	7508	510	0.225	28	92
								25		386	0.170		
								26		300	0.132		
7600	2317	18.6	24	6001	309	.136	20.6	24	7702	583	0.257	28	89
								25		422	0.186		
								26		349	0.154		
7800	2378	19.0	24	6157	344	.152	21.1	24	7895	708	0.312	28	105
								25		487	0.215		
								26		394	0.174		
8000	2439	19.5	24	6290	384	.169	21.6	24	8106	861	0.380	29	124
								25		562	0.248		
								26		444	0.196		
8200	2500	20.0	24	6423	439	.194	22.1	24	8300	1043	0.460	29	138
								25		642	0.283		
								26		495	0.218		
8400	2560	20.4	24	6579	488	.215	22.6	24	8475	1517	0.669	29	211
								25		802	0.354		
								26		587	0.259		

Table 8.4: Heat Load and Airflow Requirement Based on 1993 and 1999 Field Data

		1993 Heat Load Factor 0.098 Production 2268 tonne/day			1999 Heat Load Factor 0.14 Production 2268 tonne/day				
Level	Mean Mining Depth (m)	Fresh Air Stat'n W.B. Temp °C	Workplace Air (Stope Exhaust) W.B. Temp °C	ΔΣ Heat Stope Exhs't & Fresh Air kJ/kg	Fresh Air Stat'n W.B. Temp °C	Workplace Air (Stope Exhs't) W.B. Temp °C	ΔΣ Heat Stope Exhs't & Fresh Air kJ/kg	Fresh Air Temp. Incr. °C	Percent Decrease In Heat Removal Capacity
7000	2134	17.2	24	19.5	19.1	24 25 26	13.96 17.91 21.86	1.9	28
7200	2195	17.7	24	18.6	19.6	24 25 26	13.03 16.75 21.17	1.9	34
7400	2256	18.1	24	17.4	20.1	24 25 26	11.63 15.35 19.77	2.0	33
7600	2317	18.6	24	15.4	20.6	24 25 26	10.47 14.42 17.45	2.0	32
7800	2378	19.0	24	14.2	21.1	24 25 26	8.84 12.79 15.82	2.1	38
8000	2439	19.5	24	13.0	21.6	24 25 26	7.44 11.40 14.42	2.1	43
8200	2500	20	24	11.6	22.1	24 25 26	6.28 10.23 13.26	2.1	46
8400	2560	20.4	24	10.7	22.6	24 25 26	4.42 8.37 11.40	2.2	59

Table 8.5: Fresh Air Temperature and Heat Removal Capacity Comparison Based on 1993 and 1999 Field Data

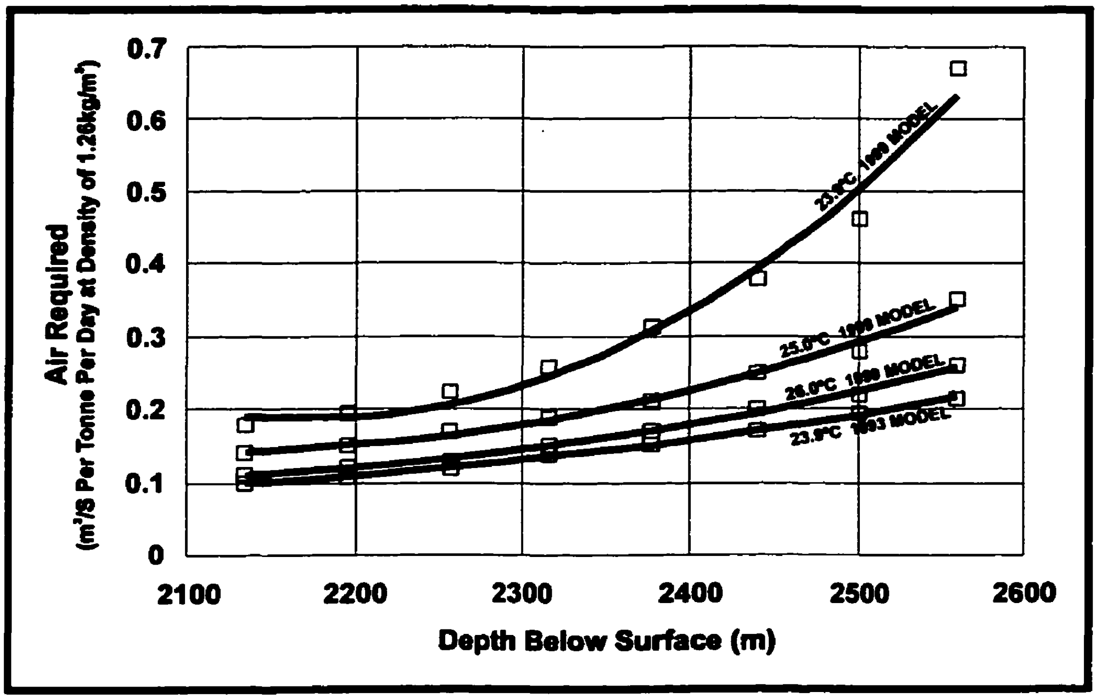

Figure 8.1: Creighton Mine Air Requirement versus Depth Based on 1993 and 1999 Field Data

Chapter 9

9.1 Conclusion and Future Work

The principle aim of this study were threefold, namely:

To examine the environmental parameters, ventilation criteria, and their challenges when mining at depth, to determine, the cooling capacity of natural phenomena of essentially free form of air conditioning at Creighton and to ascertain the depth and associated airflow requirement without mechanical refrigeration.

To accomplish these ambitious objectives, several partial goals of research were set up, principally:

- investigate the aspect of heat stress
- identify and quantify sources of heat in deep mines
- review the methods used for the control of heat in deep mines
- determine the cost to cool deep mines by mechanical refrigeration
- examine the practical aspects of heat load prediction methods and airflow requirement.

Subsequently, the literature search, into the subject matter was undertaken coupled with extensive in situ field data investigation, and analysis.

The main accomplishment of this research can be summarized as:

1. The single most important outcome of this study is the quantification and prognosis of heat storage capacity of natural heat exchanger of fragmented rock, principally the estimated cooling and heating capacity was produced by simplified model analysis (refer to Sections 6.2, 6.3, 6.4 and 6.5, Table 6.4).

   Furthermore, the mechanism of heat transfer and temperature variation in the fragmented rock as a function of yearly variation with increased flow was put forward and described in a conceptual manner (refer to Section 6.6, Figure 6.5).

   The results shown in Table 6.4 and Figure 6.5 are important, since they define the limits of free form of air conditioning at Creighton.

2. The ultimate mining depth and associated air quantity without utilization of mechanical refrigeration for design rejecting temperature range of 24 – 26°C W.B. was produced by means of experimental field data, and empirical methods (Sections 8.2, 8.5, Tables 8.4 and 8.5).

3. The viability of mechanical refrigeration versus current utilization of free air conditioning were outlined, and cost comparison documented Section 8.3.

   This study implies that, by using natural source of free air conditioning, some 440 m^3/s of air is required to mine to 8200 level, based on 1993 heat load factor, fresh air temperature and associated design parameters as per Section 8.2, Table 8.3, this compares with about 356 m^3/s for mechanical refrigeration. As can be deduced from the above, it is possible to dispense with underground refrigeration at a capital cost of some $6,685,000 by providing extra 85 m^3/s. Furthermore, this work infers that the refrigeration option would still require 3rd, intake airway and eventual replacement of main exhaust fans at #11 shaft, however, smaller size for both the airway and fans (the cost analysis is outside the scope of this report).

   The power cost of running refrigeration, breaks about even with the expense of providing larger quantity of air to depth to combat heat with natural air conditioning option and finally, natural air conditioning is preferred and less expensive option and avoids hosts of maintenance issues associated with mechanical refrigeration.

4. This study also concludes that, if current 1999, elevated heat load factor, and elevated fresh air temperature (as compared with field data and designed parameters Section 8.2, Table 8.3, Appendix 2) is maintained, then only 7800 level, 2378 m mining depth can be reached at 25°C W.B. and 487 m^3/s of air circulation Table 8.4 will be required. This implies that below 7800 level, mechanical refrigeration would need to be considered.

   This is attested to by "enormous" quantity of air requirements (beyond the capability of ventilation infrastructure) in order to maintain workplace temperature range of 24 – 25°C W.B., and is documented in Sections 8.5, 8.5.1, Tables 8.4 and 8.5.

   Furthermore Figure 8.1 depicts very clearly the airflow requirements at various mining depths, for 1993 model at which Creighton ventilation expansion system is based, and compares it with 1999 models requirements for 24 – 26°C W.B. temperature range.

Suggestion For Future Work

The research work conducted at Creighton during this field and experimental study has revealed the need for:

- Further research into Creighton heat exchange areas that could lead into analyses of the heat transfer in a quantitative manner through several numerical models.

- More frequent reevaluation of the HLF (heat load factor), upon which empirical assessment of mine heat load airflow requirements, and refrigeration assessments are based.

- Closer evaluation of correlation between empirical and mathematical methods of heat load airflow and refrigeration prognosis should be undertaken.

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APPENDIX 1

Comparison of Datamine Versus Manual Calculations
For Broken Rock In The 3 Shaft Pit

In November of 1998 a Datamine model was constructed and compared to a manually calculated volume assessment of the 3 shaft pit area which was done in a 1996 study. The results of which and data are as follows.

For 25 Degree Draw (Volume in millions of cubic feet)

	Manual Volume	Datamine Volume	% Difference
Blocks 1 & 2	414.9	425.3	-2.4%
Blocks 5 & 6	216.4	235.5	-8.1%
TOTAL	631.3	660.8	-4.5%

For 45 Degree Draw (Volume in millions of cubic feet)

	Manual Volume	Datamine Volume	% Difference
Blocks 1 & 2	459.3	461.2	-0.4
Blocks 5 & 6	289.1	289.1	-4.4%
TOTAL	748.4	748.4	-2.0%

There is good correlation of the volumes in both calculations with the Datamine calculations considered to be more accurate.
If the Datamine numbers are used then the other tonnages and other numbers calculated in the 1996 study should be modified by the percentages shown above.

The other parameters in the original report are still valid for calculations based on the new Datamine volumes.
The broken rock is norite(gabbro).
The in situ solid specific gravity for norite is 2.95.
The broken rock tonnage factor is 18 cubic feet per ton.

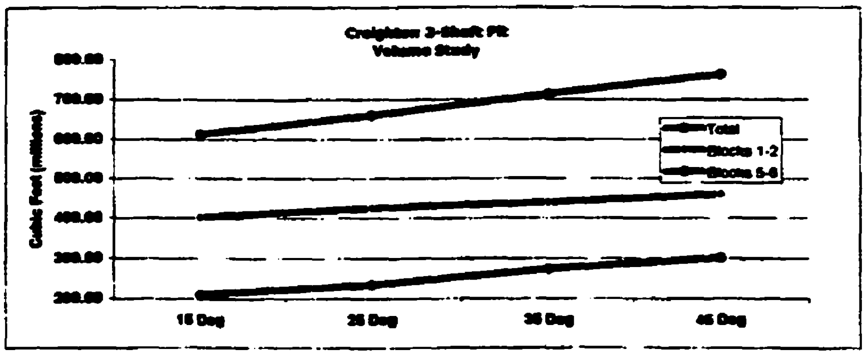
Shown below are additional Datamine calculations for 15 and 35 degrees along with the 25 and 45 degree volumes.

CREIGHTON

3 SHAFT PIT

VOLUMES FOR VARIOUS AIR DRAW ANGLES

	VOLUMES				VOLUMES (millions)			
	15 Deg	25 Deg	35 Deg	45 Deg	15 Deg	25 Deg	35 Deg	45 Deg
BLOCKS 1&2	402,613,000	425,282,700	439,623,000	461,150,900	402.61	425.28	439.62	461.15
BLOCKS 5&6	209,940,000	235,549,900	273,423,700	302,297,200	209.94	235.55	273.42	302.30
TOTAL	612,553,000	660,812,600	713,046,700	763,448,100	612.56	660.81	713.05	763.45



1.0 Introduction

The following outlines the method in which the volume and tonnage of broken rock, and the corresponding surface area, above the slusher trenches in the 3 Shaft Pit was calculated. It is based on an aerial survey of the 3 Shaft Pit performed in 1995.

The results from this report will be used to determine the 'cooling capacity' of the broken rock above 3 Shaft Pit. The 3 Shaft Pit is divided into six Blocks (1 to 6), and presently air is being delivered to the mine workings through Blocks 2, 5, and 6. The planned expansion of the Creighton Mine ventilation system from 1.0 million cfm to 1.5 million cfm will mean that Block 1 must also be utilized.

As Blocks 2, 5, and 6 are currently being used, we can develop relationships between the calculated surface areas and broken rock volumes and tonnages, and the known temperatures and volumes of the air being delivered through those Blocks. These relationships can then be applied to Block 1 to estimate the temperature and volume of the air that would flow through that area and into the mine workings.

2.0 Methodology

As previously noted, an aerial survey of 3 Shaft Pit was completed in 1995. From this survey, a 1:300 scale topographic map was created, and then sections were cut at 50 foot intervals parallel to Creighton Mine's World Coordinate system, (attached).

Each section (in AutoCAD) contains its pit topography and the location of the available slusher trenches beneath the pit. On each section a line was drawn from trench floor to trench floor, and the most easterly and westerly trenches have lines drawn up at 25 degrees from vertical (scenario 01), and also 45 degrees from vertical (scenario 02) connecting to the pit floor. In cases where the available trenches are situated east or west of the pit limit for the particular section, a line was drawn from the most easterly (or westerly) trench within the pit limit up to the-pit floor. These lines were then traced using AutoCAD's draw polyline subroutine to create a polygon, and the area of each section (polygon) was then calculated using AutoCAD's area of polygon subroutine (see sections attached).

Multiplying the areas determined from AutoCAD by the width of influence for each section (50 feet) will provide the volume of broken rock for each section. The volume is then converted to tonnage by dividing the volume by a broken rock tonnage factor of 18.0 cu ft/ton. Each section is allotted to a certain Block according to the topographic map (attached), so that a Block volume and tonnage can be determined.

The surface areas are determined in a similar manner. On each section, the true length of the pit floor was measured and then multiplied by the section influence (50 feet) to give the surface area of the particular section. Each section is allotted to the proper Block, and Block surface areas are calculated.

Note: The 45 degree angle of draw is used because the most easterly and westerly holes in each trench were fanned to 45 degrees from vertical. The 25 degree angle is used because it is a conservative estimate of the angle at which the rock broke.

3.0 Results

The following tables summarize the block volumes, tonnages, and surface areas for scenario #1 (25 degree angle), and scenario #2 (45 degree angle). Detailed spreadsheets are attached.

Block	1	2	5	6	TOTAL
Volume					
(cuft X 1.0 E6)	62.7	352.2	70.7	145.7	631.3
Tonnage					
(tons X 1.0 E6)	3.5	19.6	3.9	8.1	35.1
Surface area					
(sqft X 1.0 E6)	0.22	0.88	0.33	0.37	1.8

Table 1: Block volume, tonnage and surface area for 25 degree angle of draw.

Block	1	2	5	6	TOTAL
Volume					
(cuft X 1.0 E6)	64.1	395.2	90.3	198.8	748.4
Tonnage					
(tons X 1.0 E6)	3.6	22.0	5.0	11.0	41.6
Surface area					
(sqft X 1.0 E6)	0.24	1.13	0.46	0.59	2.4

Table 2: Block volume, tonnage and surface area for 45 degree angle of draw.

4.0 Discussion

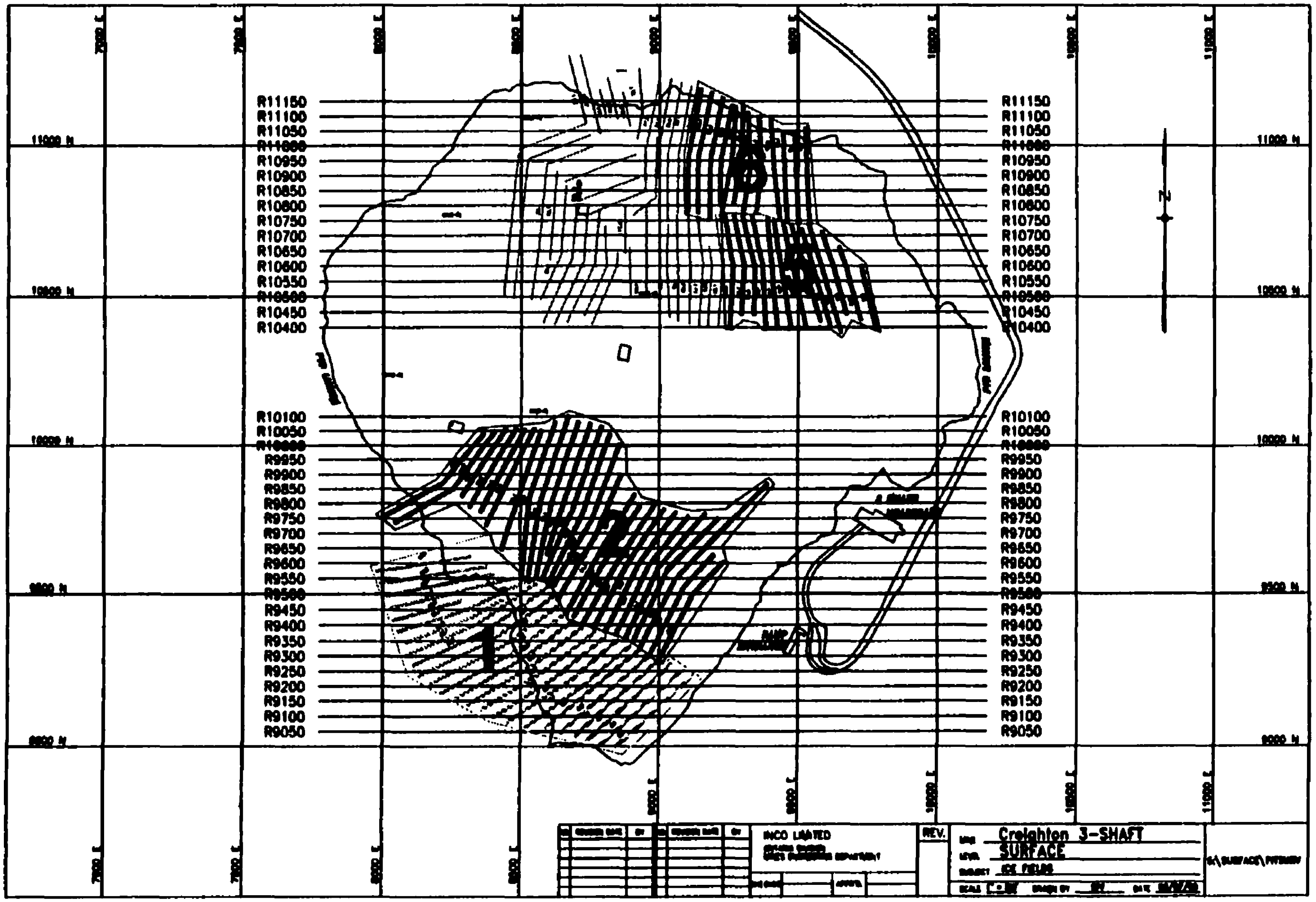
The percent difference in volume, tonnage and surface area between the two scenarios is quite noticeable, as table 3 indicates.

Block	1	2	5	6	TOTAL
Volume	2.2	12.2	27.7	36.4	18.5
Tonnage	2.2	12.2	27.7	36.4	18.5
Surface area	9.1	28.4	39.4	59.4	33.3

Table 3: Percent difference for block volume, tonnage and surface area between 25 degree and 45 degree angle of draw.

5.0 Conclusion

For a 25 degree angle of draw, the total pit volume is 631.3 million cubic feet, total pit tonnage is 35.1 million tons, and the total pit surface area is 1.8 million square feet. For a 45 degree angle of draw, the total pit volume is 748.4 million cubic feet, total pit tonnage is 41.6 million tons, and the total pit surface area is 2.4 million square feet.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R10850
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R10400
R10100
R10050
R9950
R9900
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R9550
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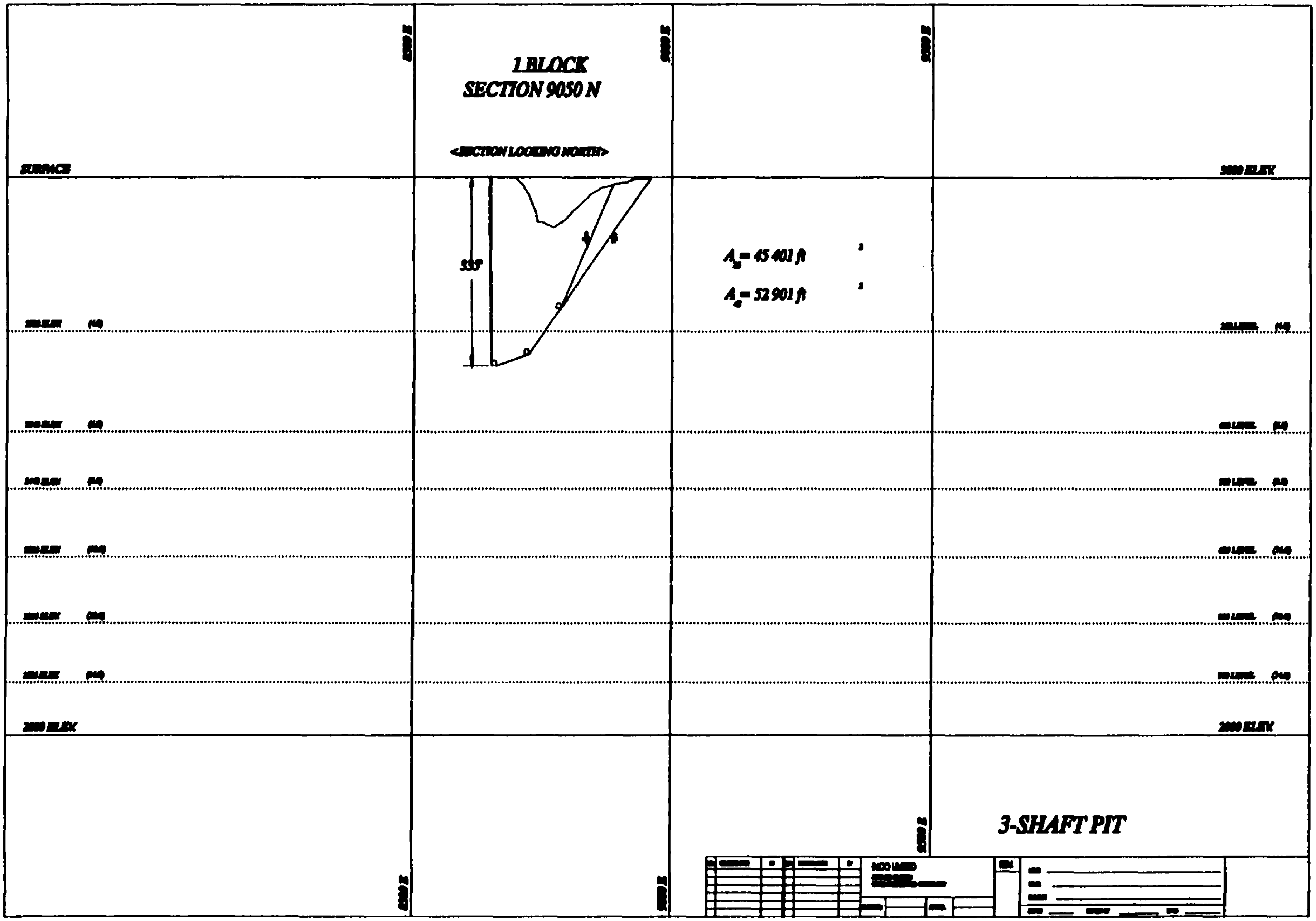
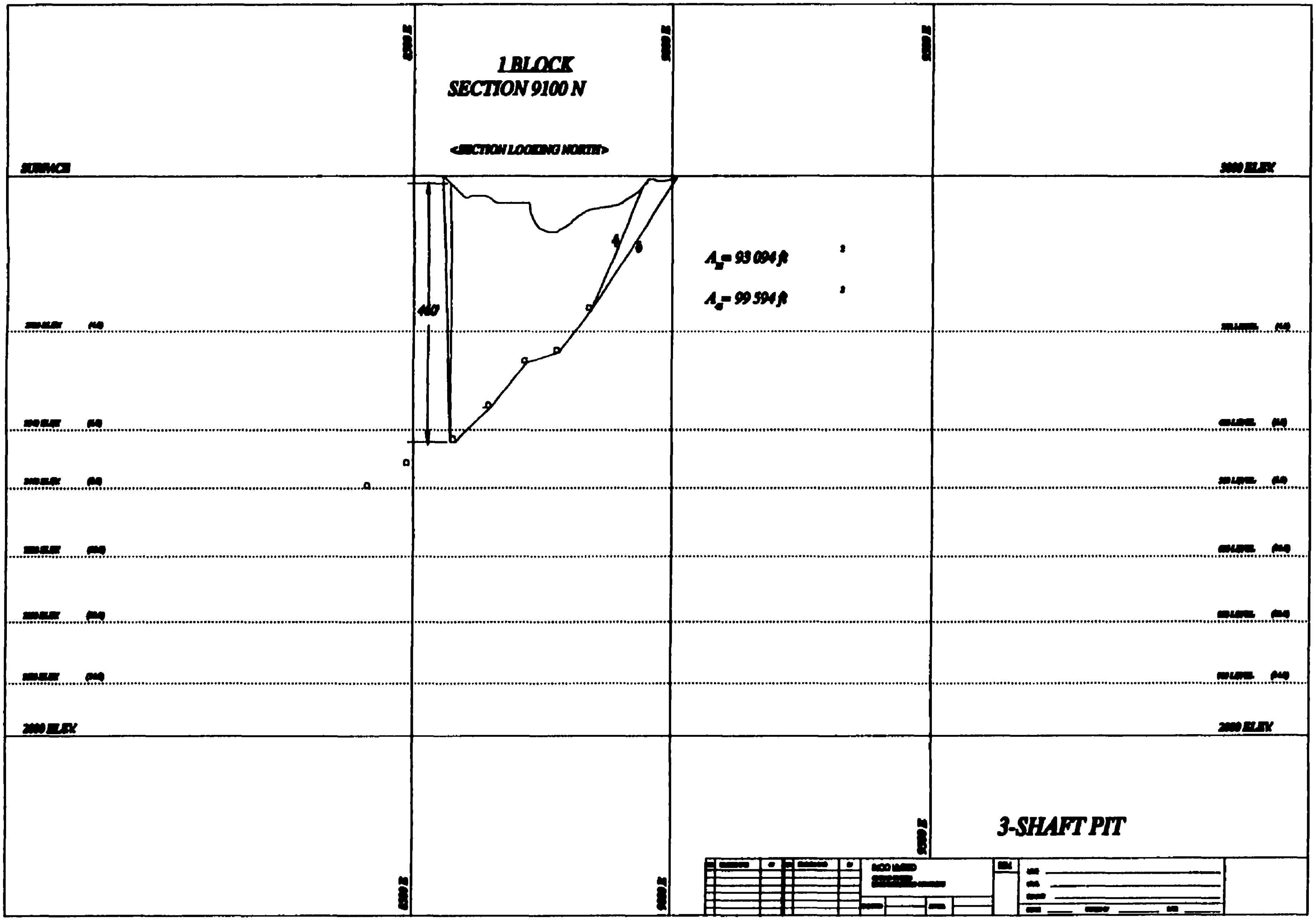
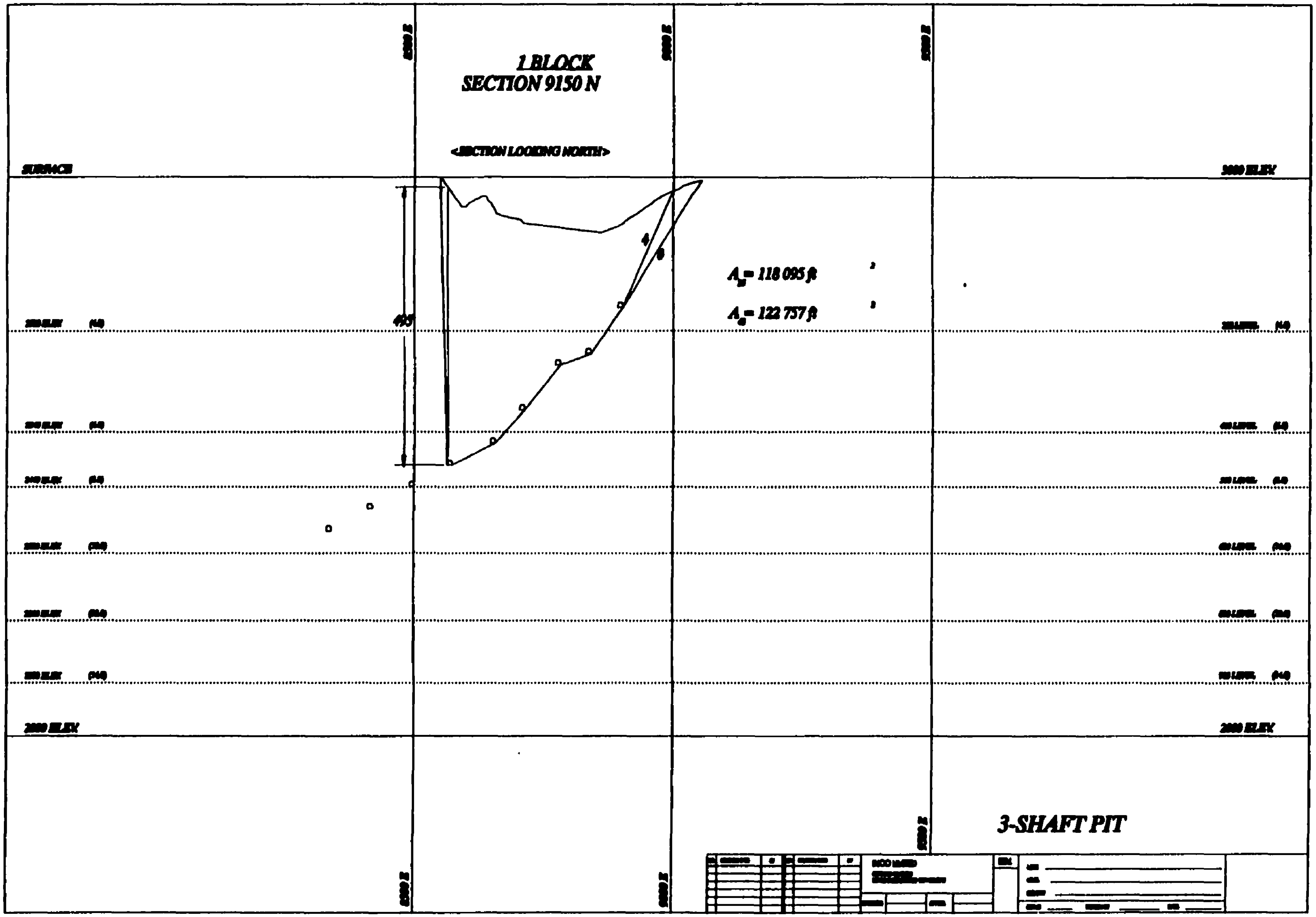
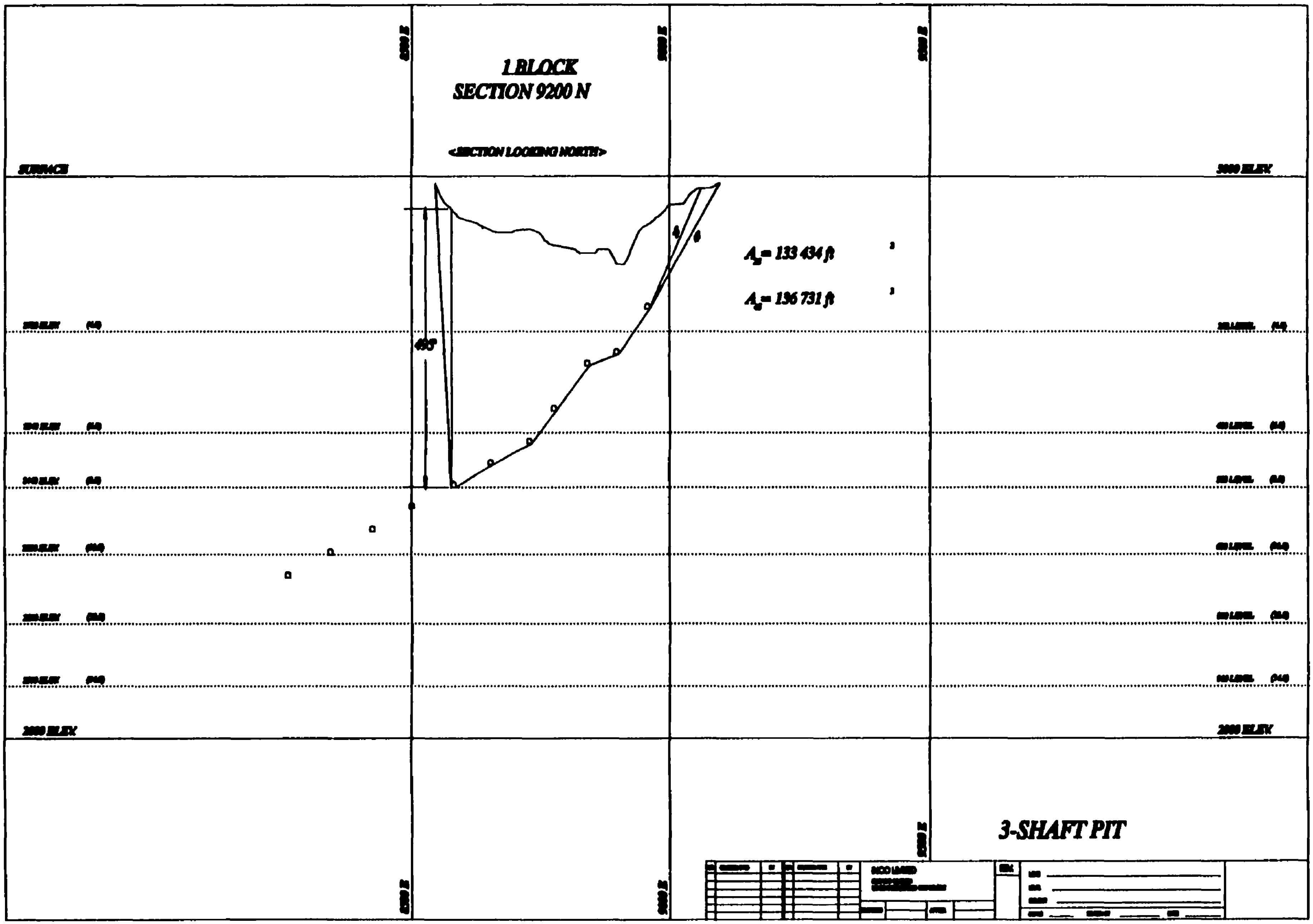
N
INCO LIMITED
REV.
Creighton 3-SHAFT
SURFACE
ICE FIELDS

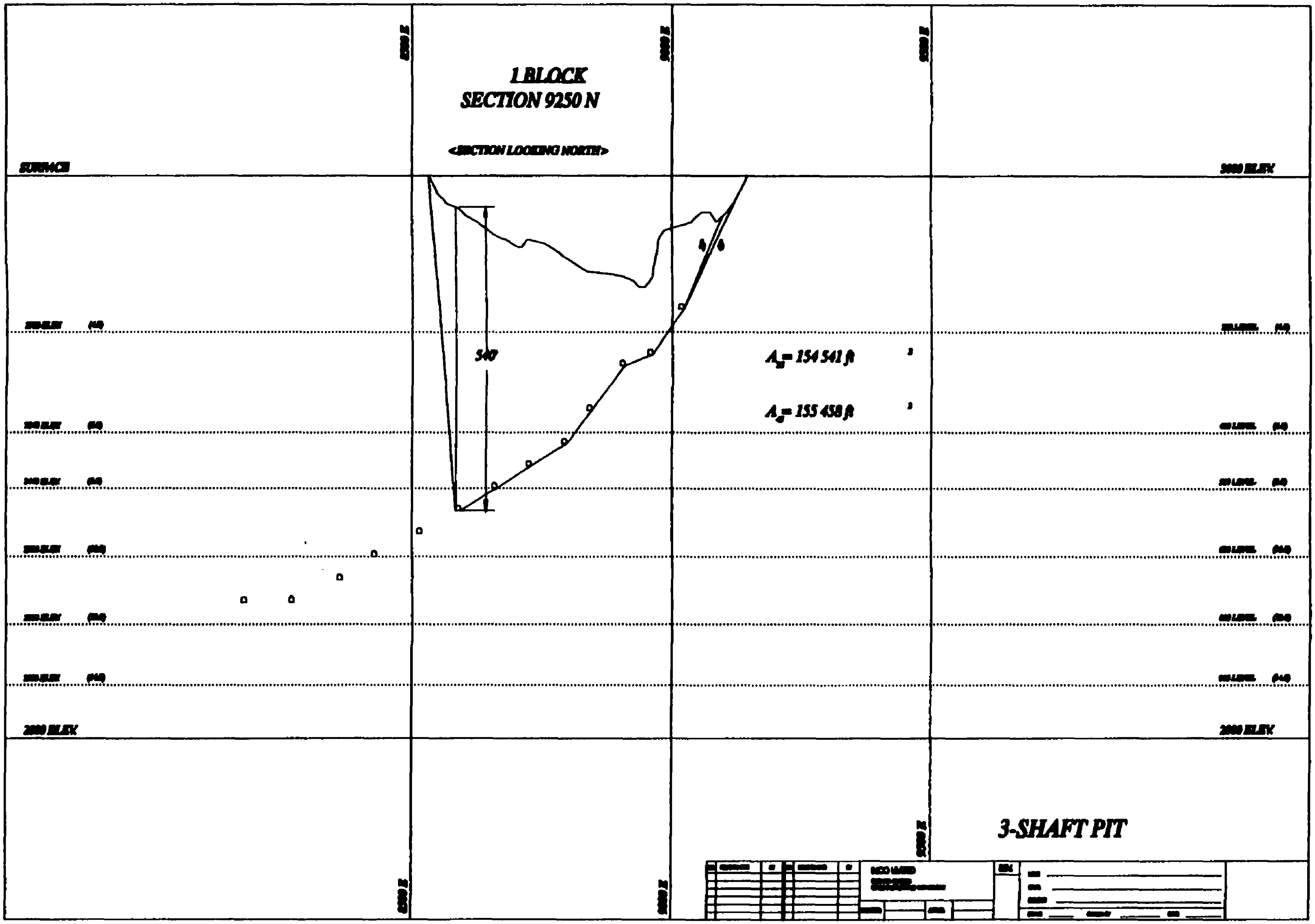


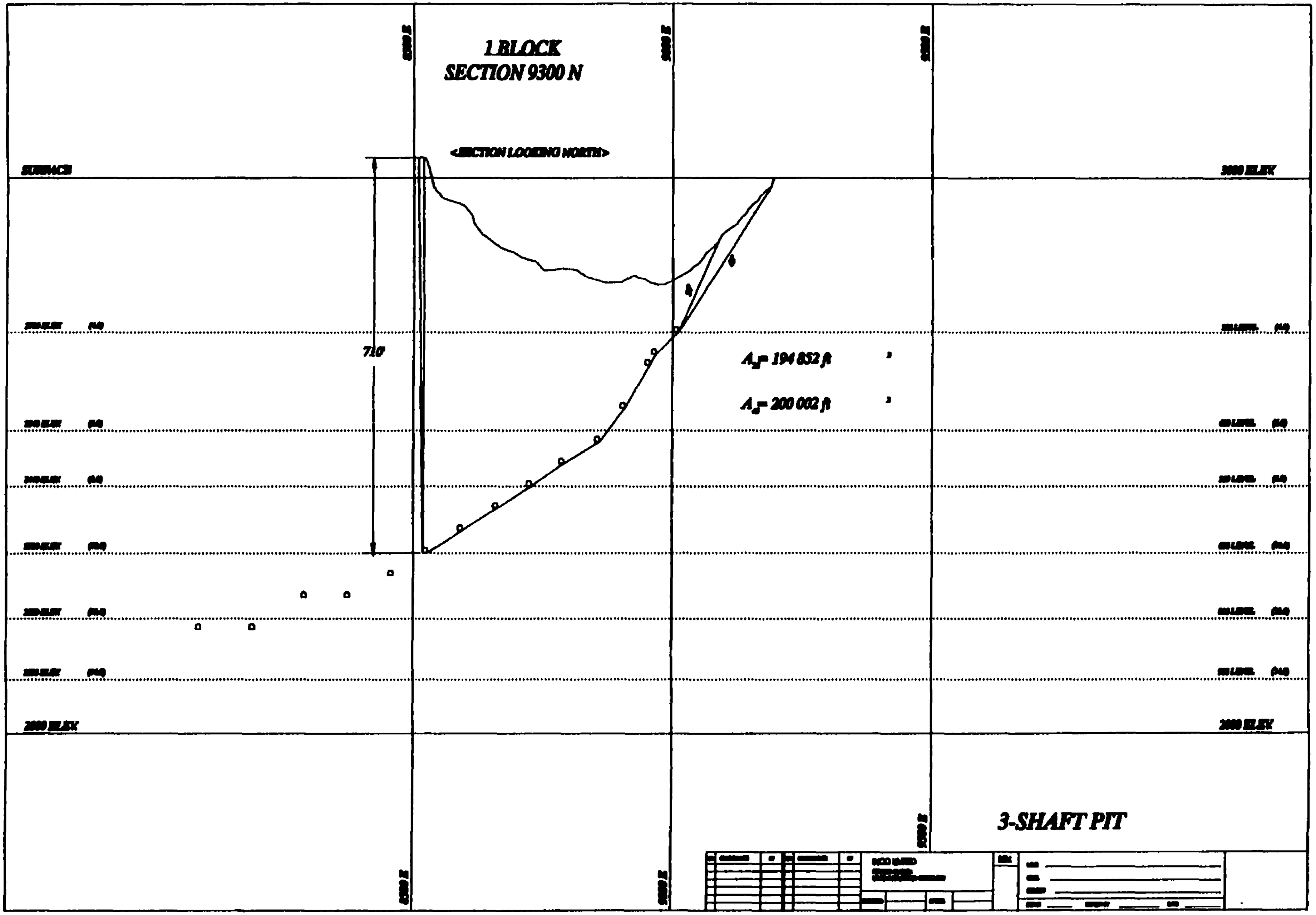
1 BLOCK
SECTION 9050 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
A_m = 45 401 ft²
A_a = 52 901 ft²
2000 ELEV
2000 ELEV
3-SHAFT PIT
INCO LIMITED

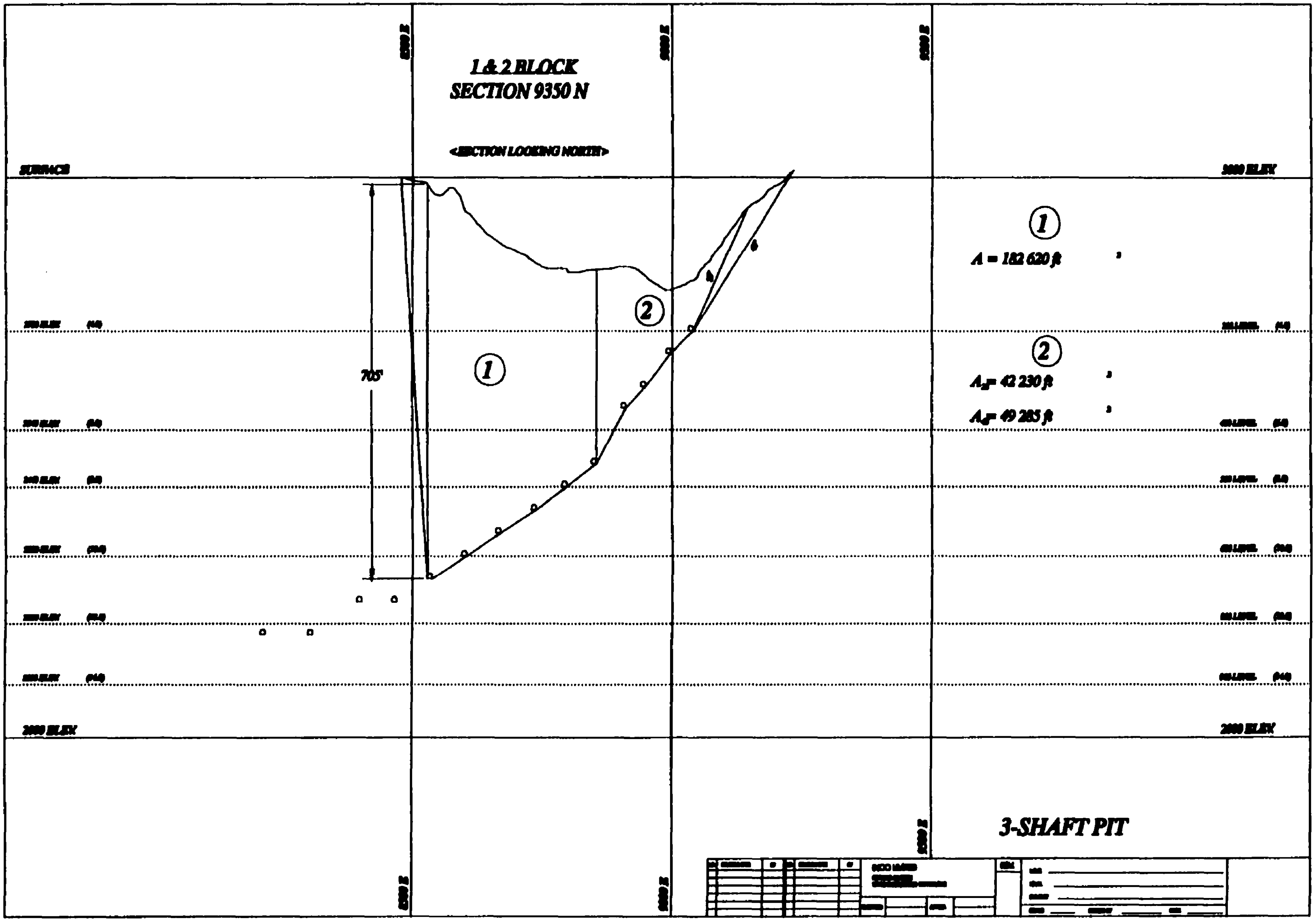

1 BLOCK
SECTION 9100 N
3000 ELEV
2000 ELEV
440
A = 93 094 ft²
A = 99 594 ft²
3-SHAFT PIT

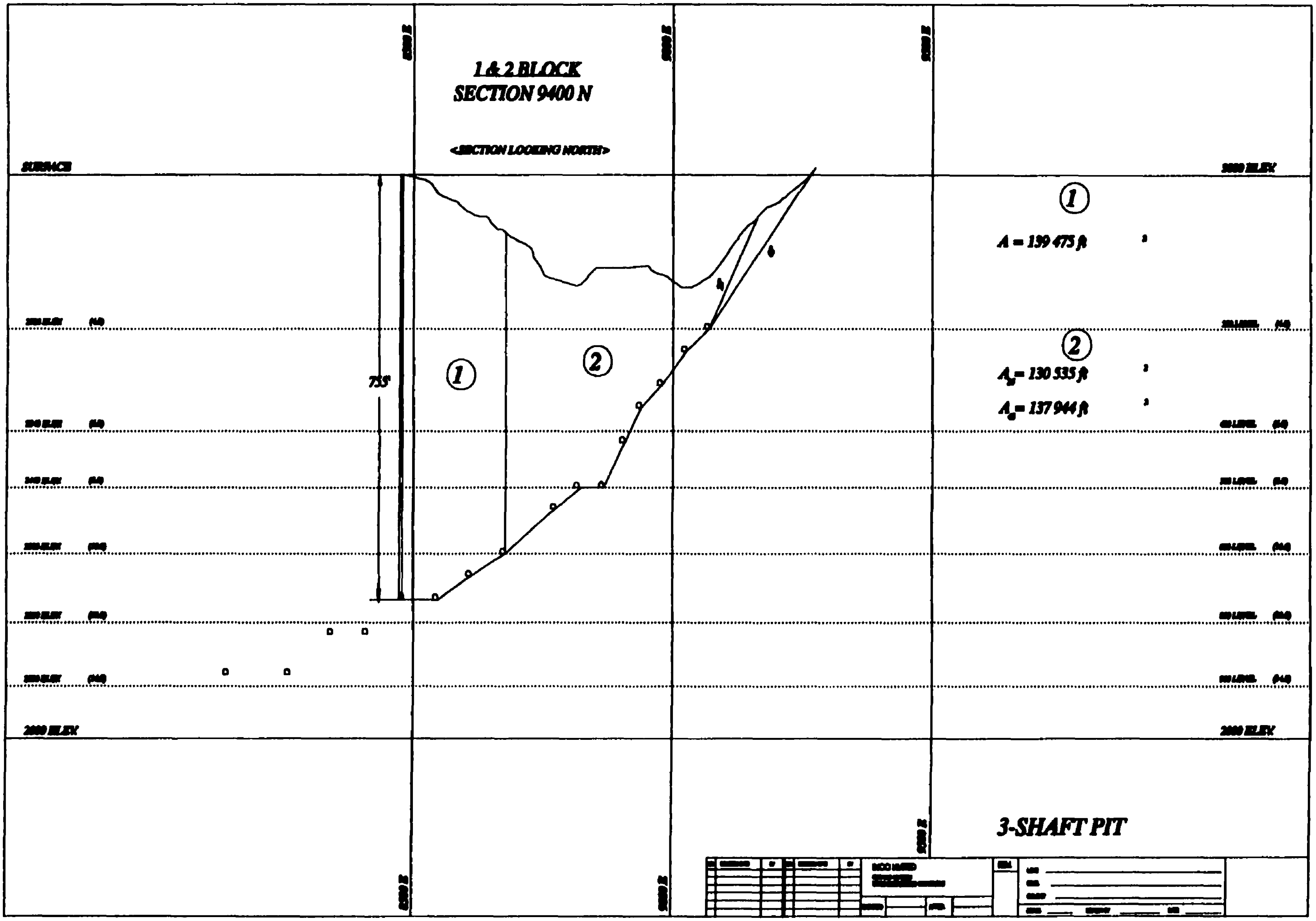

1 BLOCK
SECTION 9150 N
<SECTION LOOKING NORTH>
SURFACE
495'
A_s = 118 095 ft²
A_a = 122 757 ft²
3-SHAFT PIT


1 BLOCK
SECTION 9200 N
SURFACE
2000 ELEV
A_m= 133 434 ft³
A_c= 136 731 ft³
3-SHAFT PIT

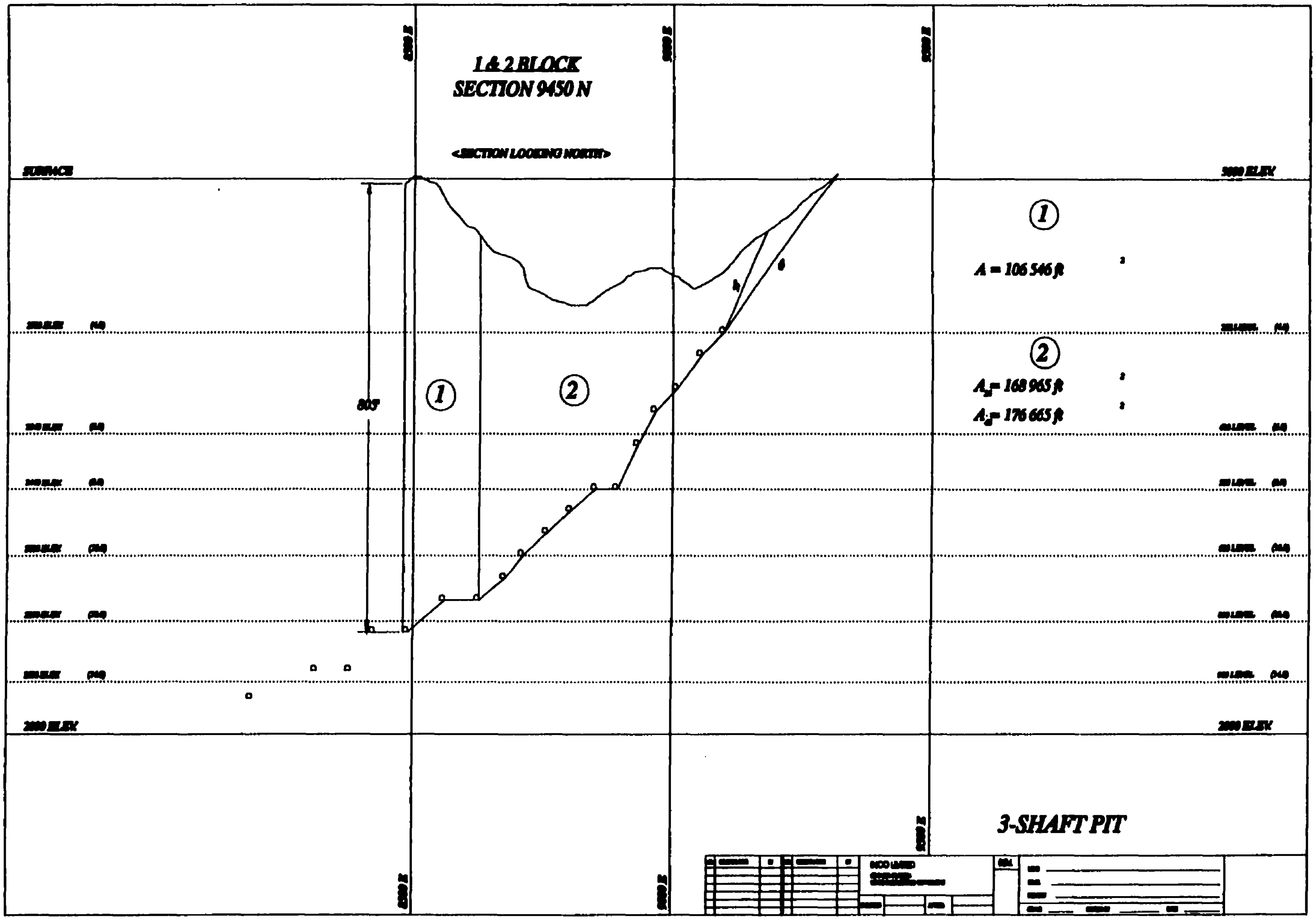

1 BLOCK
SECTION 9250 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
2000 ELEV
540
A_s= 154 541 ft^2
A_d= 155 458 ft^2
3-SHAFT PIT


1 BLOCK
SECTION 9300 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
710'
A_d= 194 852 ft^3
A_d= 200 002 ft^3
2000 ELEV
2000 ELEV
3-SHAFT PIT

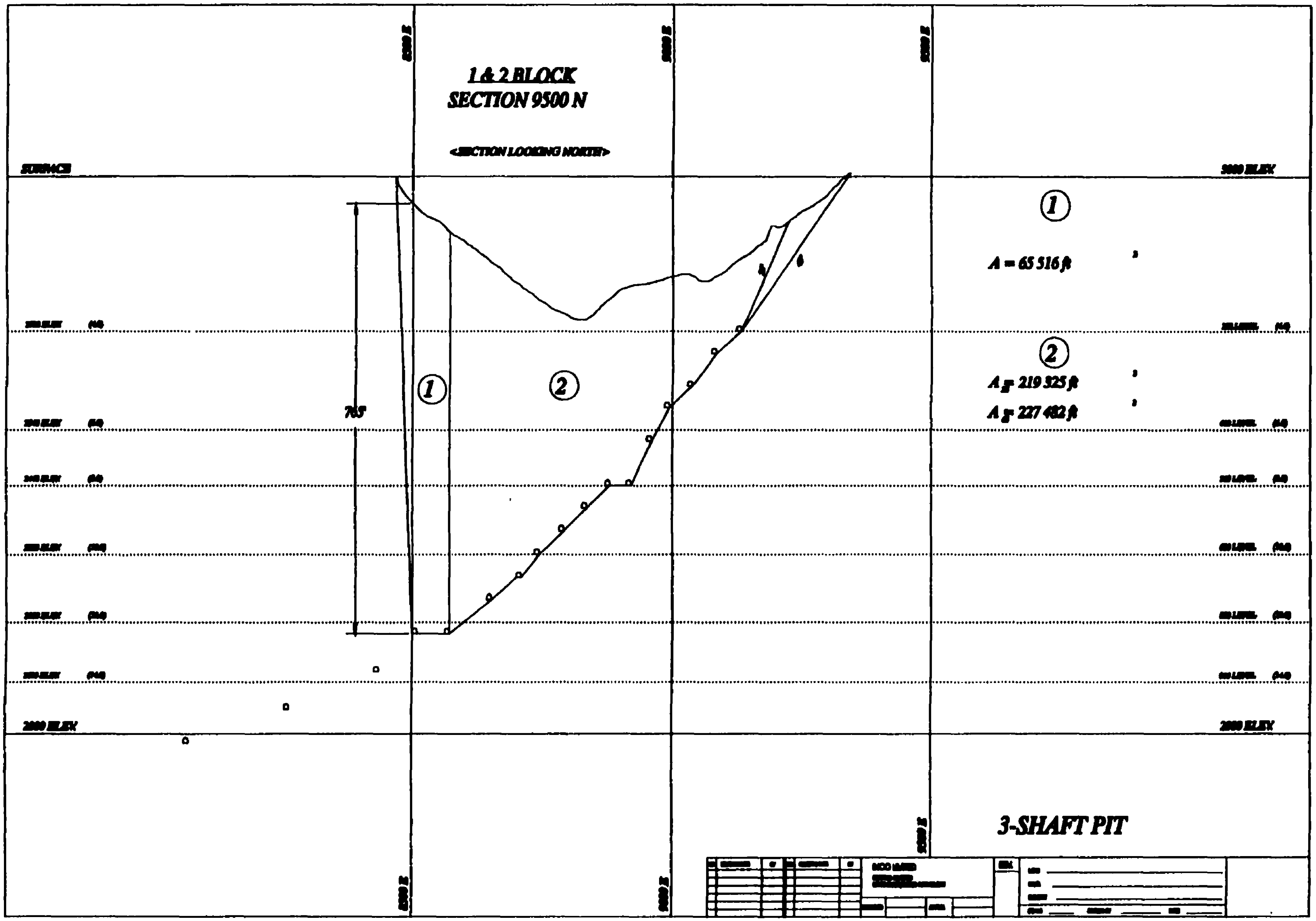

1 & 2 BLOCK
SECTION 9350 N
<SECTION LOOKING NORTH>
SURFACE
705'
1
2
A = 182 620 ft³
A₂= 42 230 ft³
A₃= 49 285 ft³
2000 ELEV
3-SHAFT PIT

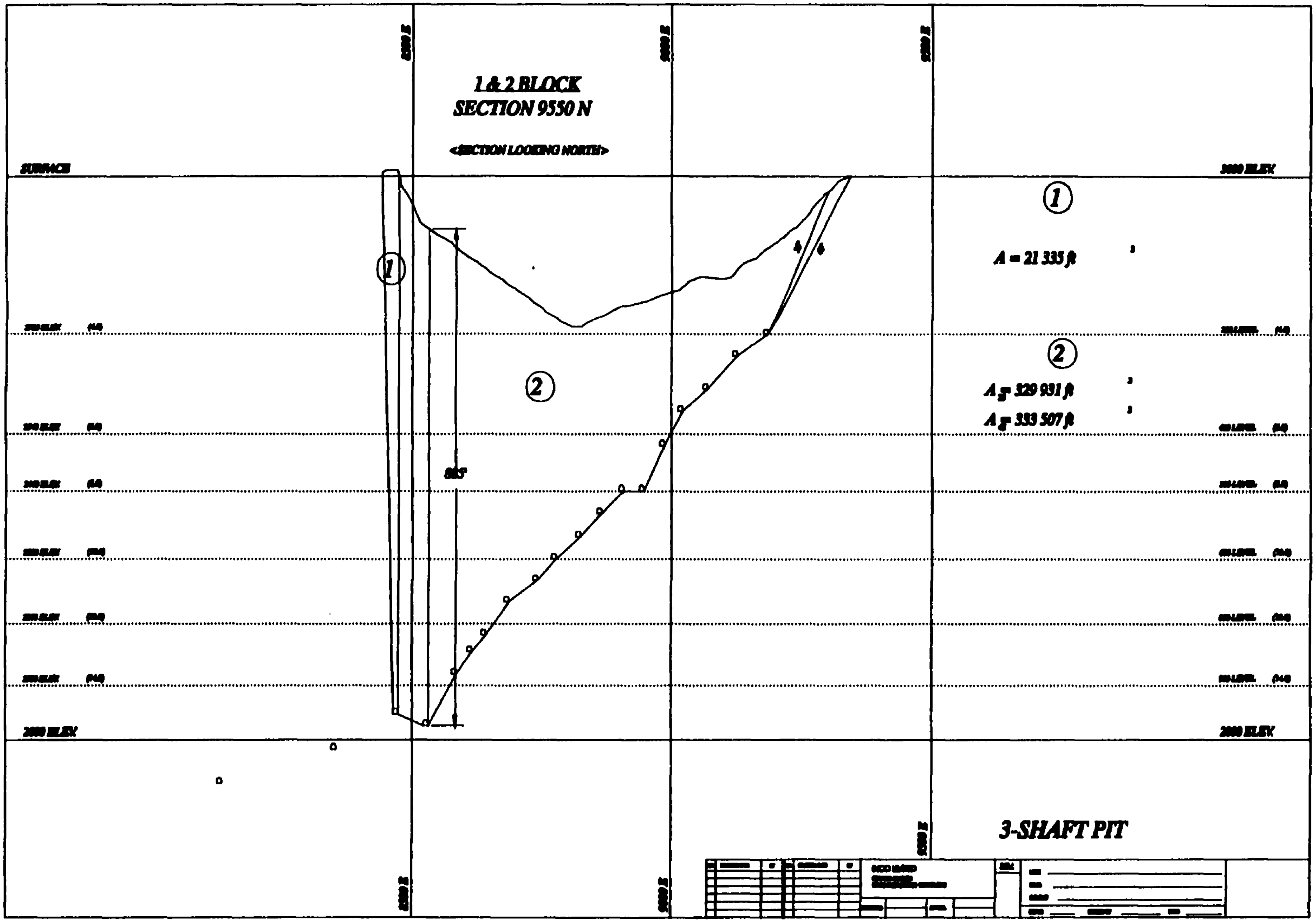


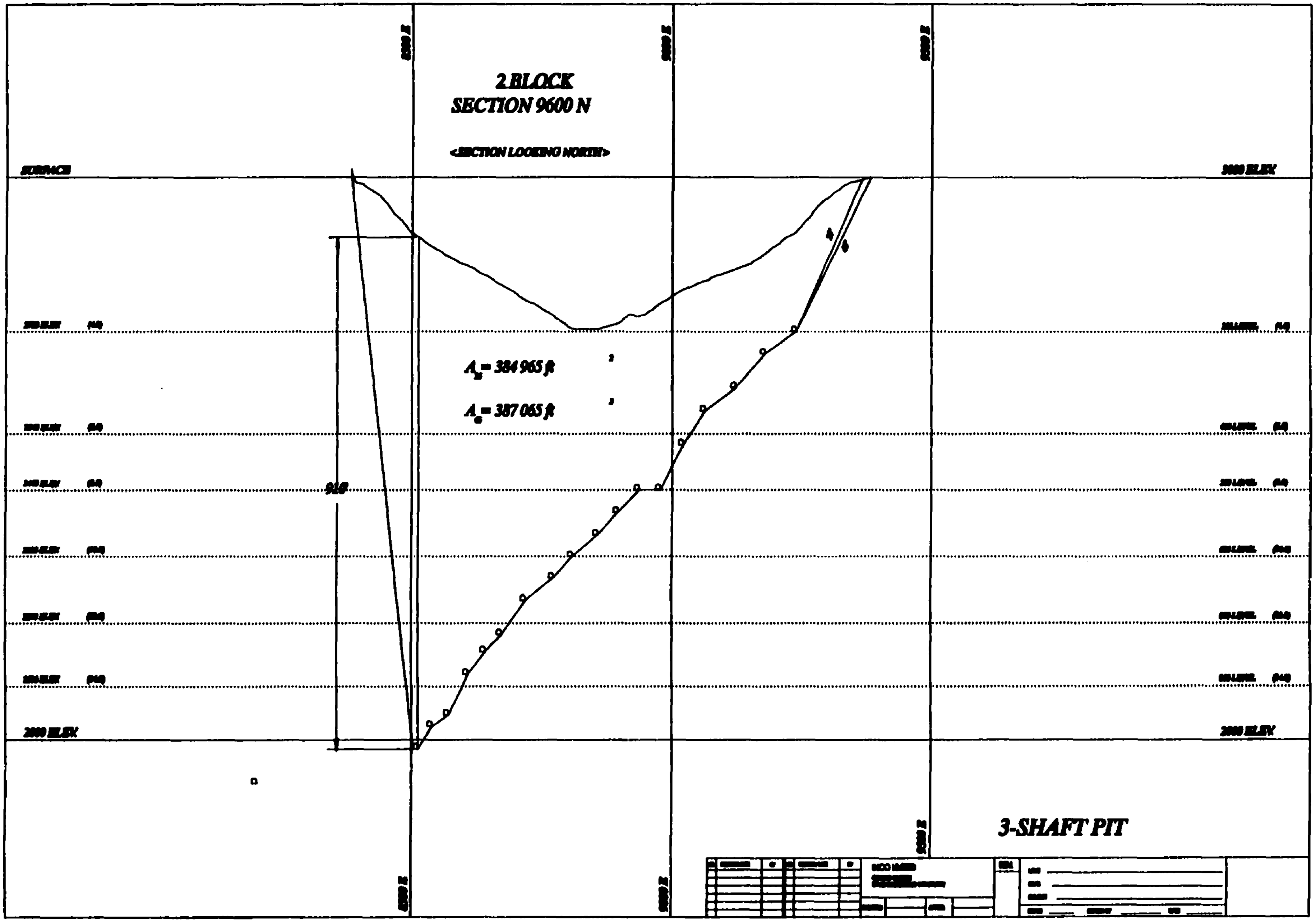
1 & 2 BLOCK
SECTION 9400 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
735'
1
2
A = 139 475 ft²
A = 130 535 ft²
A = 137 944 ft²
3-SHAFT PIT



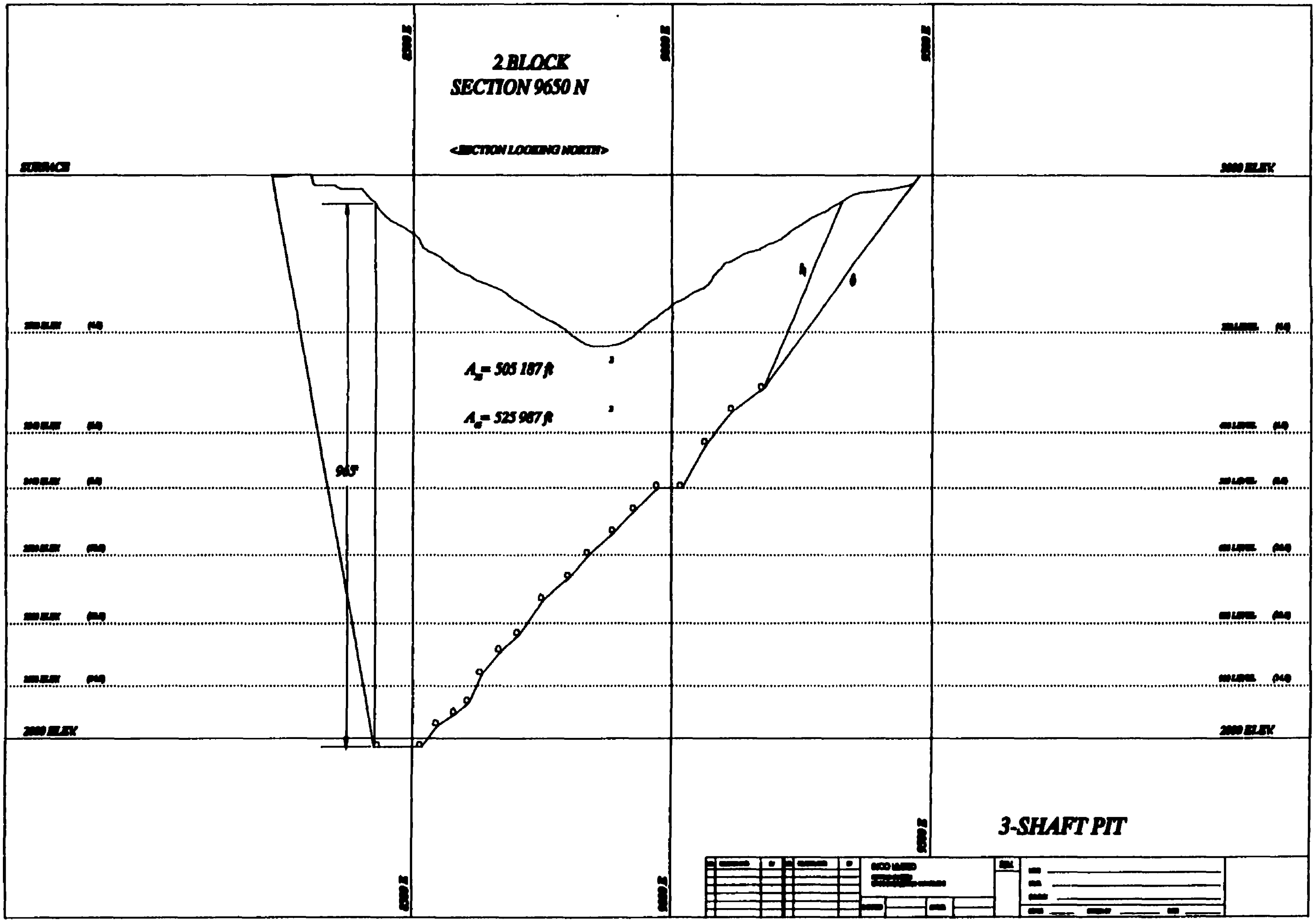
1 & 2 BLOCK
SECTION 9450 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV.
2000 ELEV.
805'
1
2
A = 106 546 ft 2
A = 168 965 ft 2
A = 176 665 ft 2
3-SHAFT PIT
INCO LIMITED

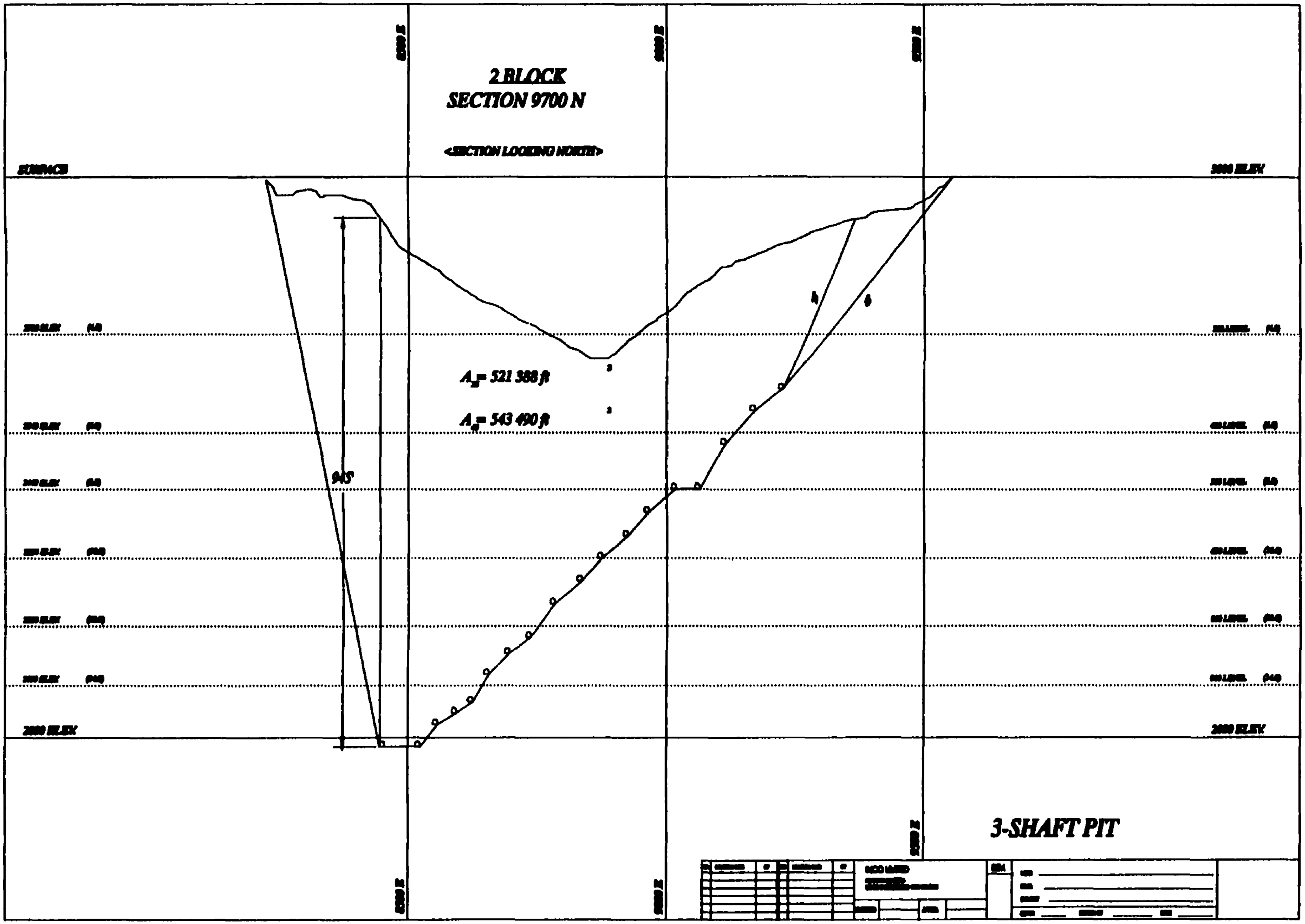

1 & 2 BLOCK
SECTION 9500 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
765'
1
2
A = 65 516 ft²
A = 219 325 ft²
A = 227 482 ft²
3-SHAFT PIT

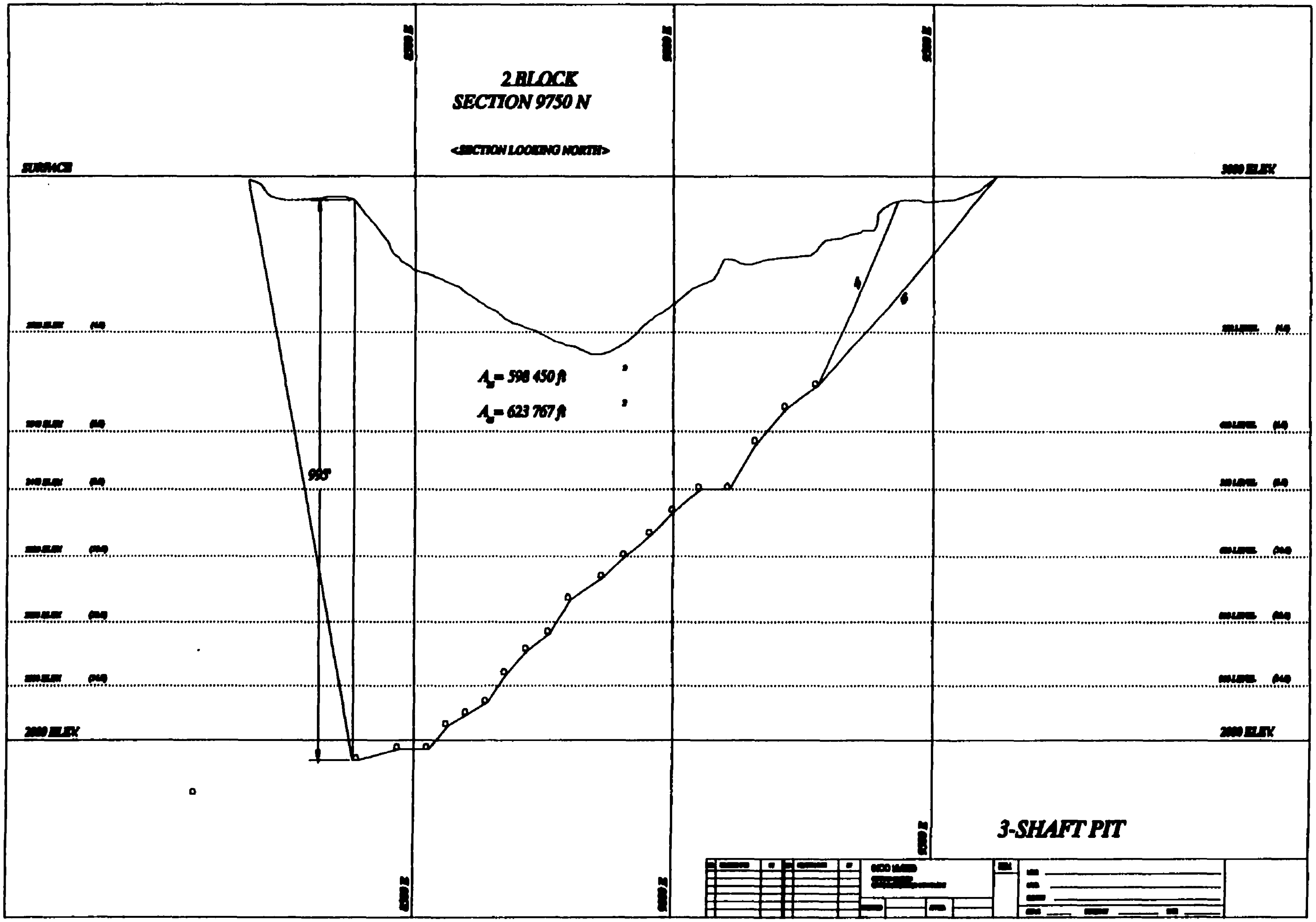

1 & 2 BLOCK
SECTION 9550 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
1
2
$A = 21\,335\ ft^3$
$A = 329\,931\ ft^3$
$A = 333\,507\ ft^3$
3-SHAFT PIT

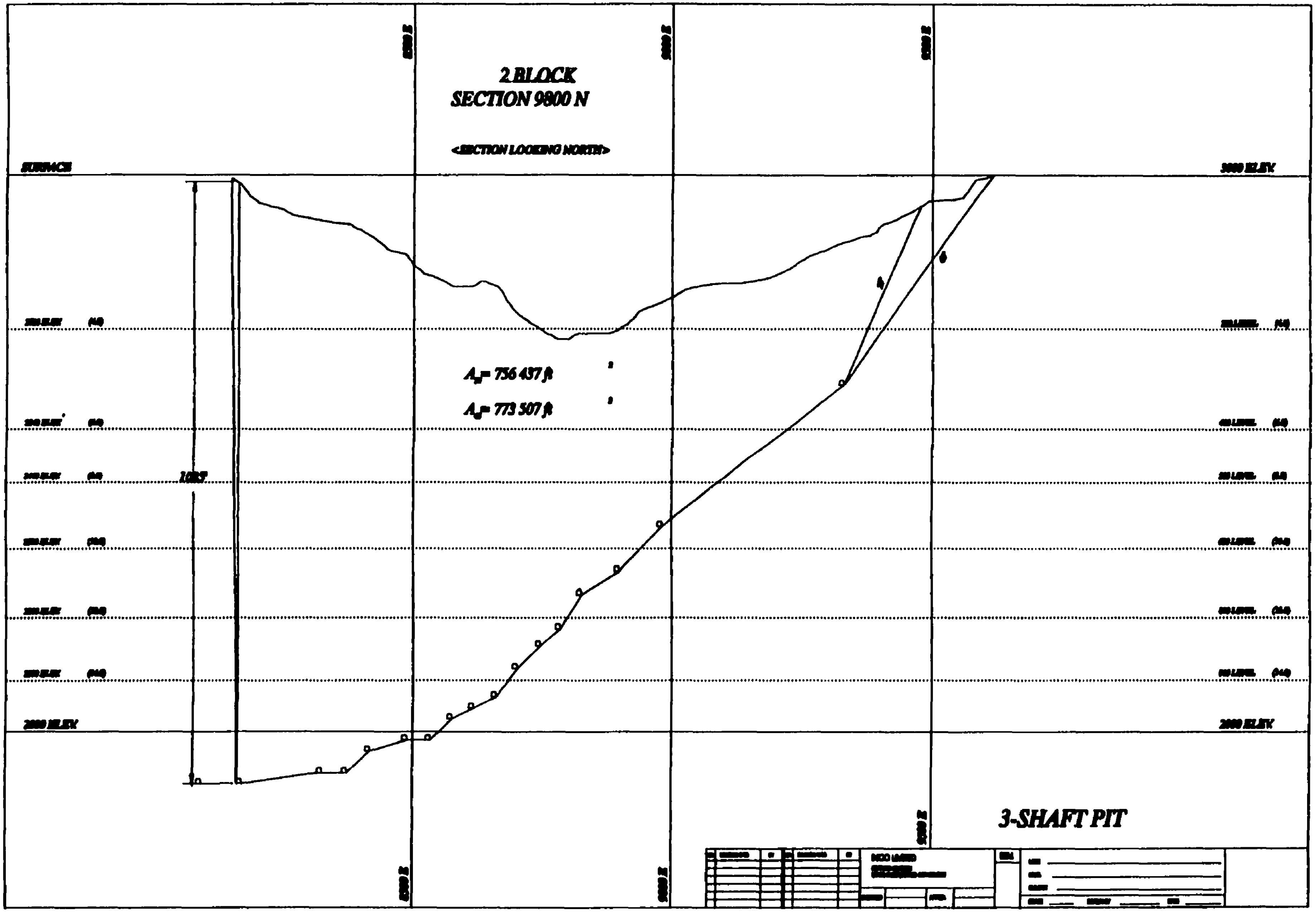


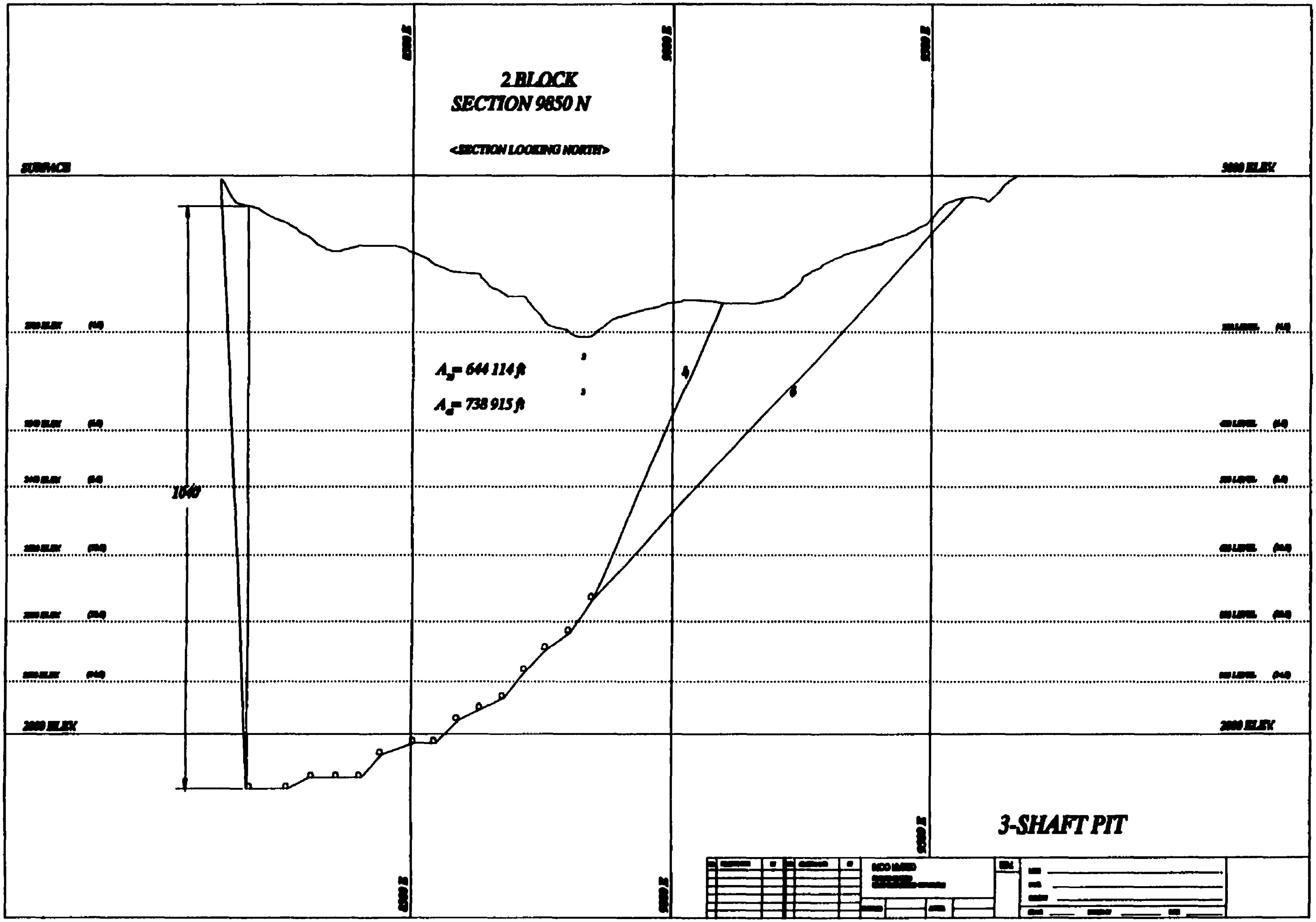
2 BLOCK
SECTION 9600 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
A_s = 384 965 ft²
A_e = 387 065 ft²
3-SHAFT PIT

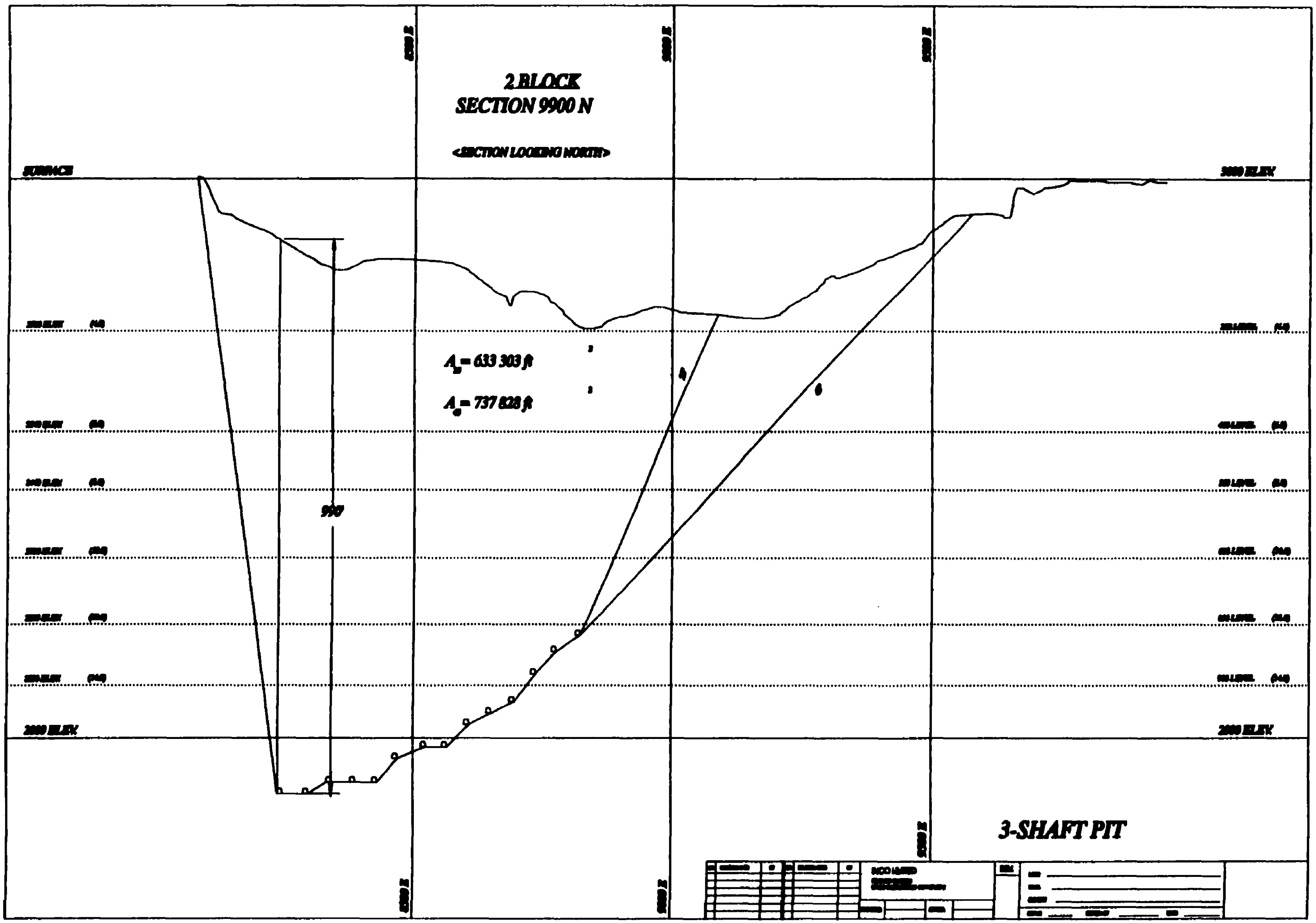

2 BLOCK
SECTION 9650 N
SURFACE
2000 ELEV.
A_{s} = 505 187 ft²
A_{e} = 525 987 ft²
965'
3-SHAFT PIT


2 BLOCK
SECTION 9700 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
2000 ELEV
945'
A_{sr}= 521 388 ft²
A_{d}= 543 490 ft²
3-SHAFT PIT

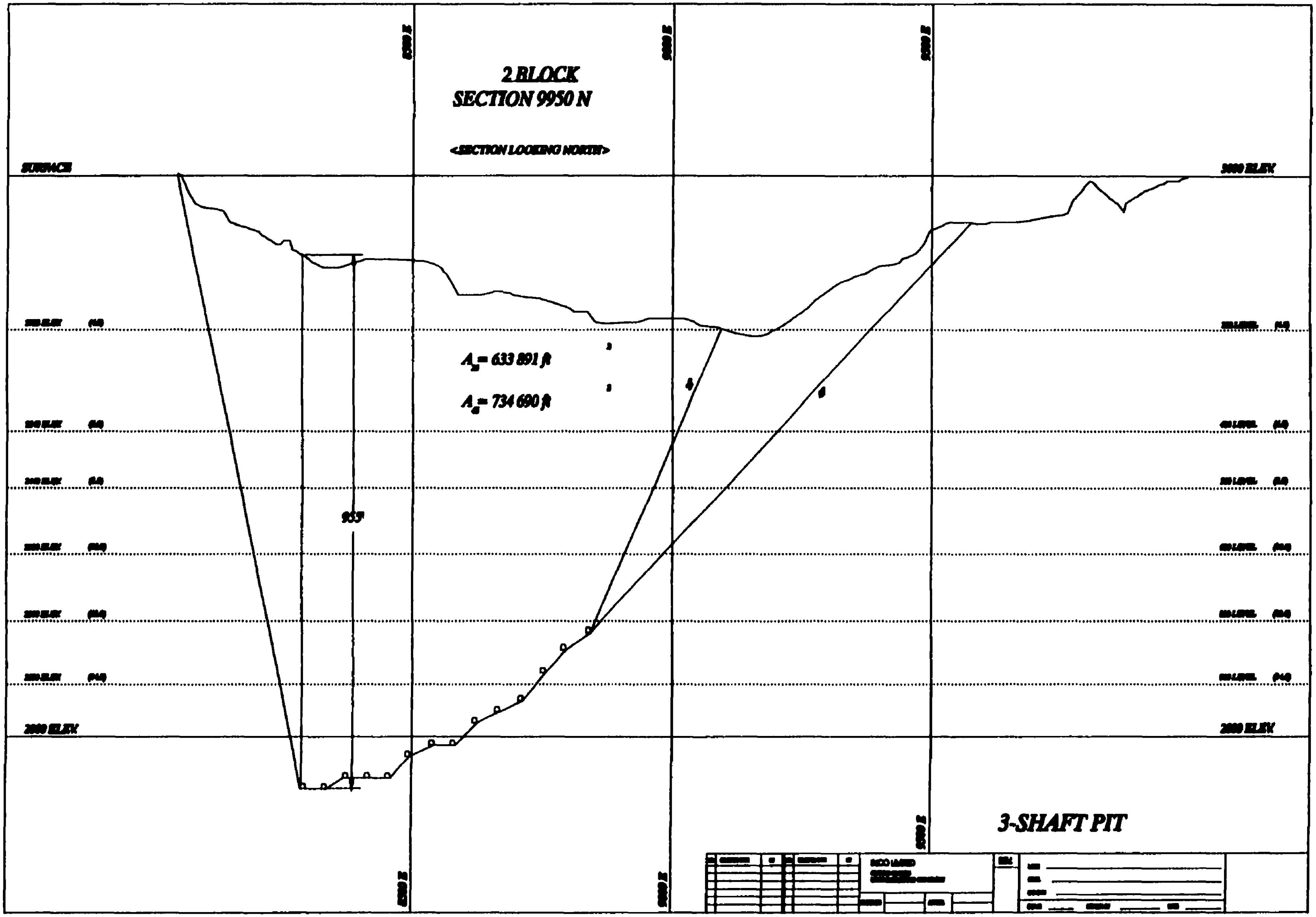

2 BLOCK
SECTION 9750 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
2000 ELEV
A_s = 598 450 ft²
A_s = 623 767 ft²
995'
3-SHAFT PIT

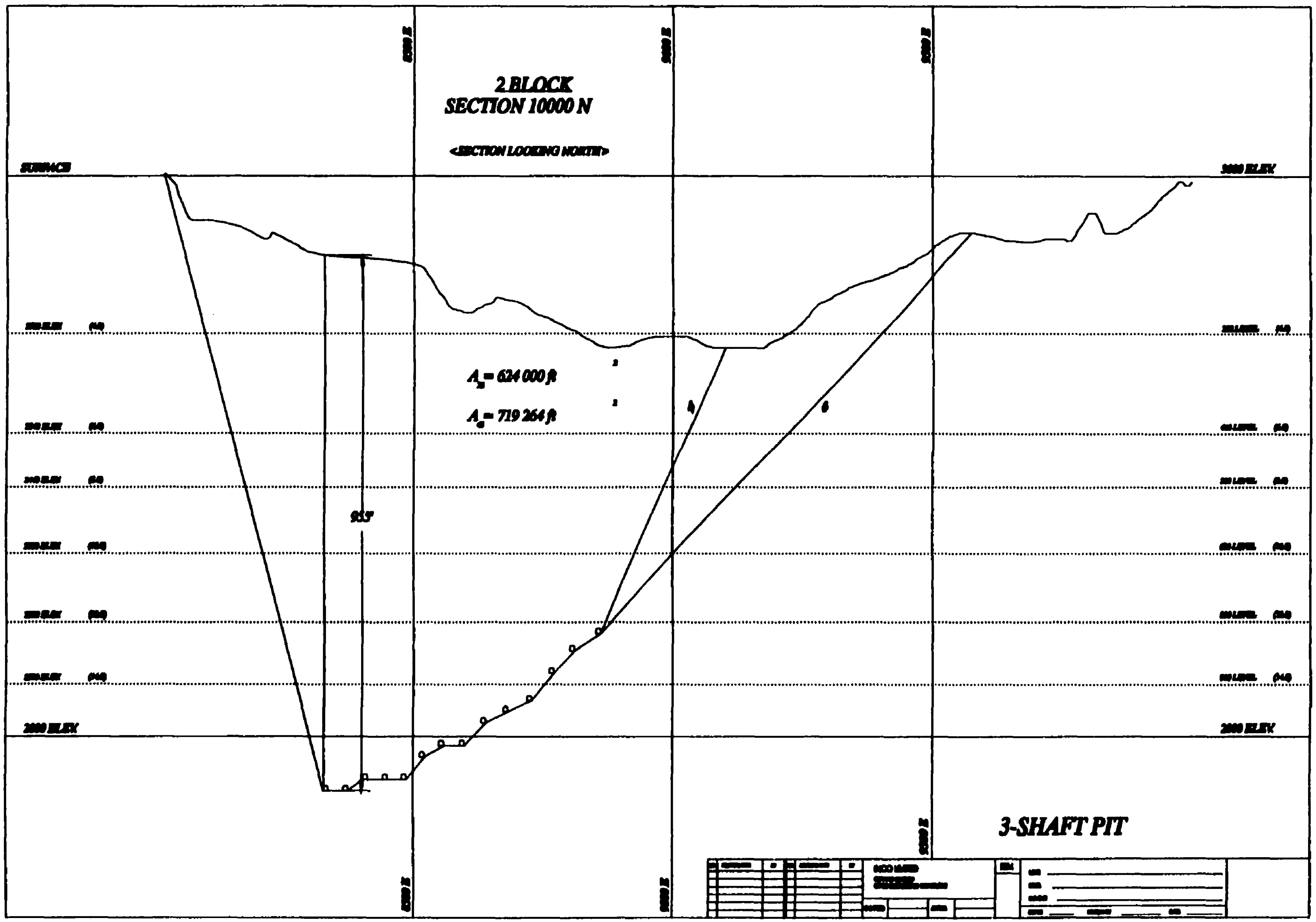

2 BLOCK
SECTION 9800 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV.
2000 ELEV.
$A = 756\,437\ ft^2$
$A = 773\,507\ ft^2$
3-SHAFT PIT

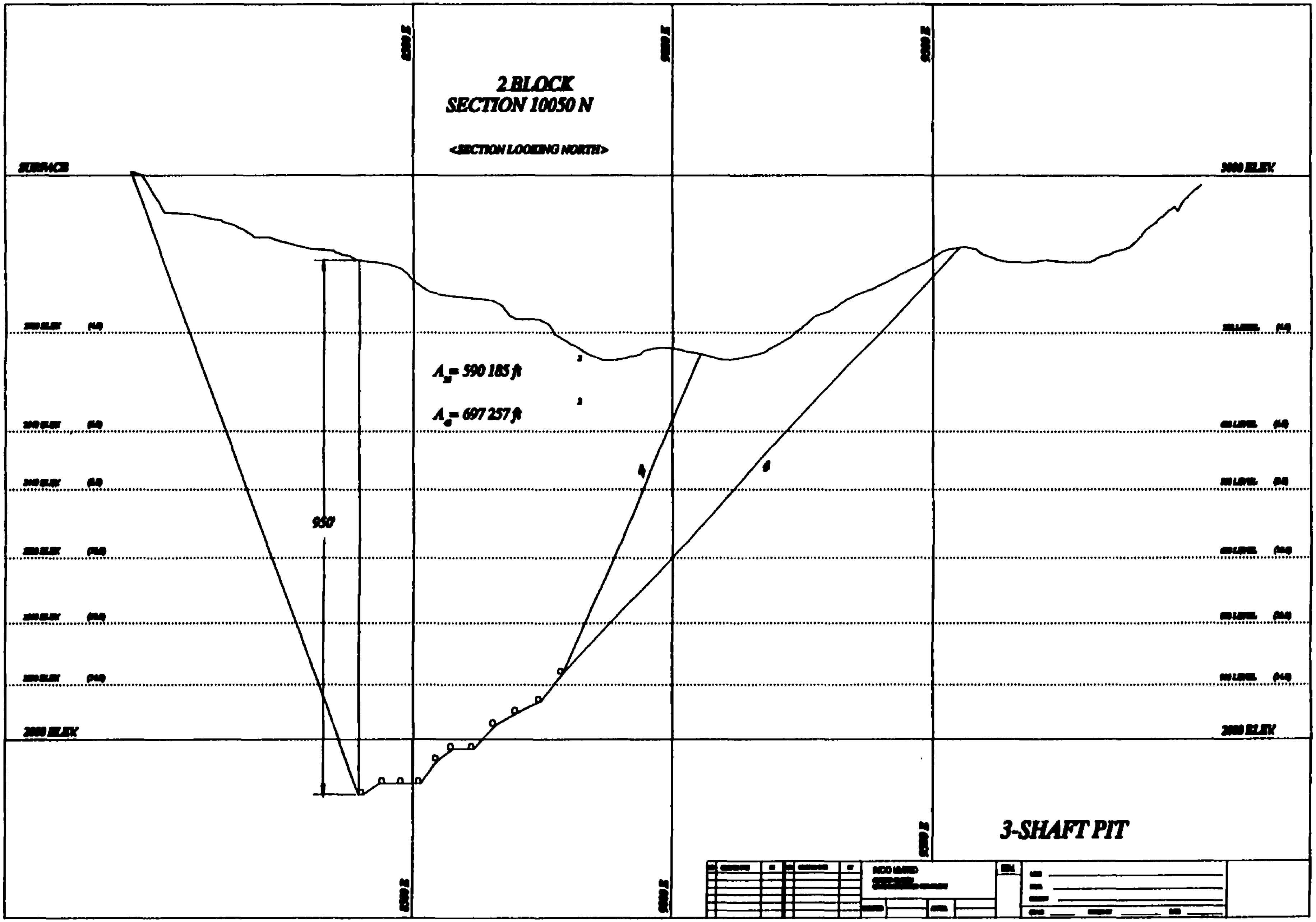

2 BLOCK
SECTION 9850 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
2000 ELEV
A_g = 644 114 ft
A_d = 738 915 ft
1040'
3-SHAFT PIT

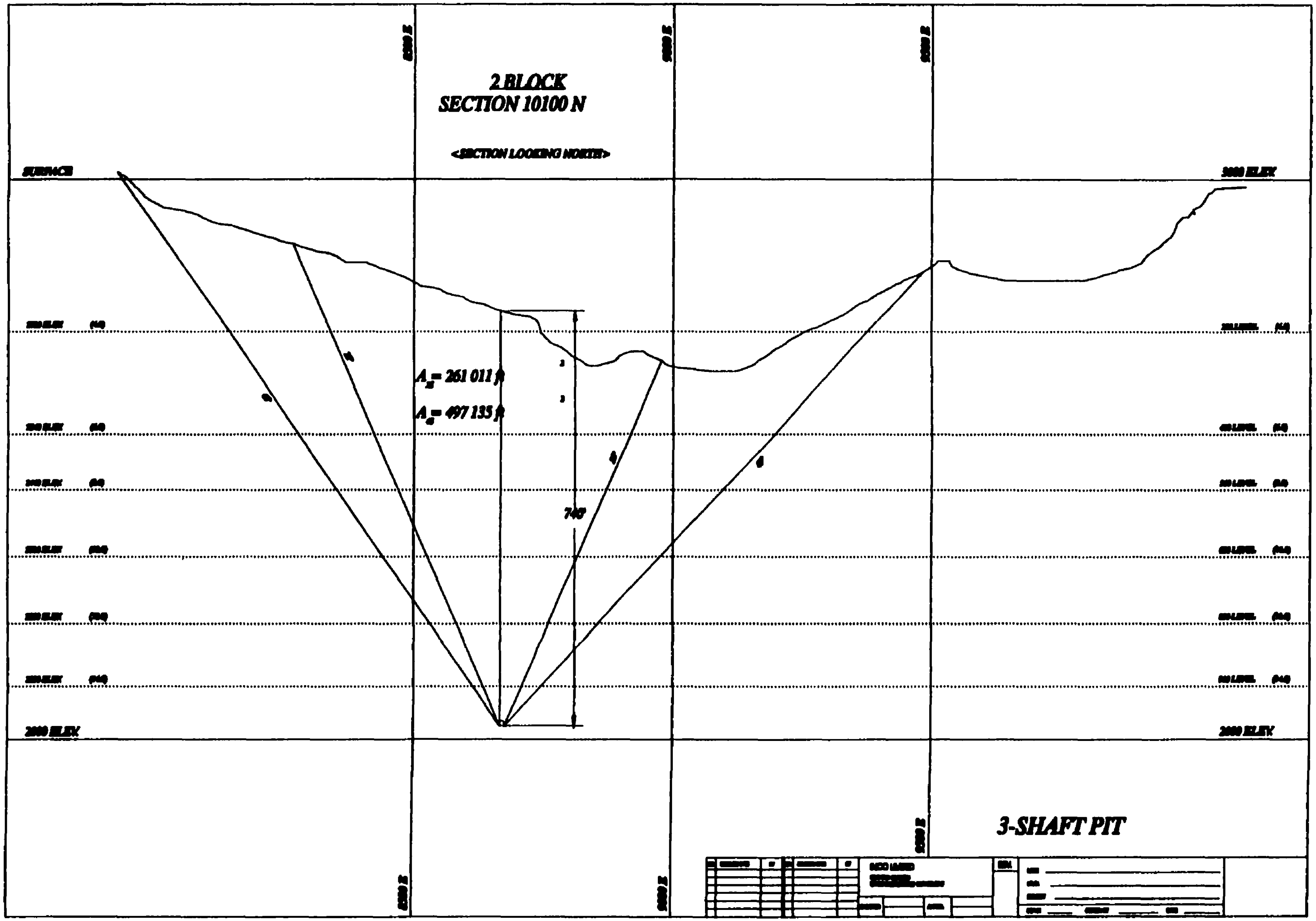


2 BLOCK
SECTION 9900 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV.
2000 ELEV.
A_p = 633 303 ft²
A_a = 737 828 ft²
990'
3-SHAFT PIT

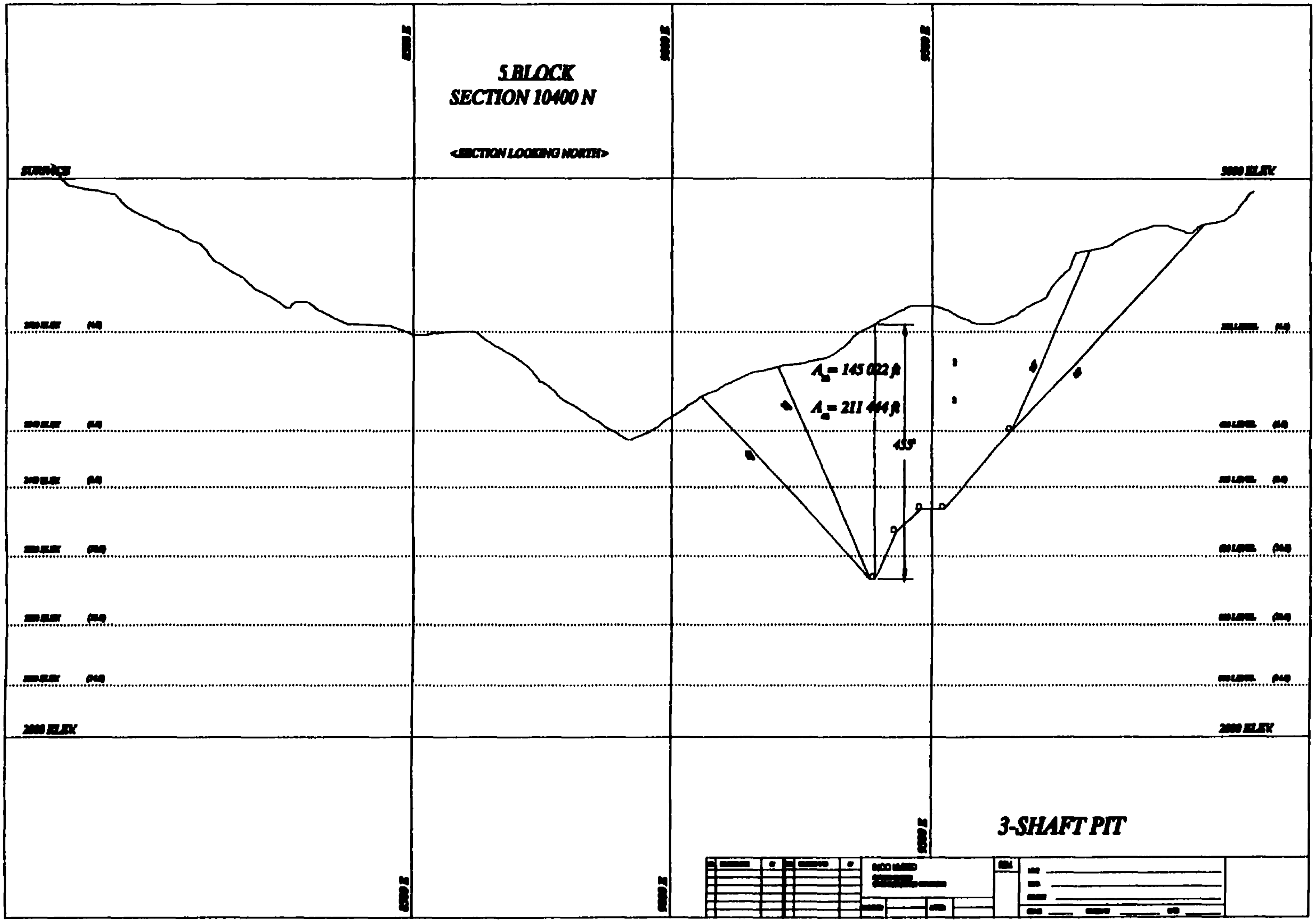

2 BLOCK
SECTION 9950 N
<SECTION LOOKING NORTH>
SURFACE
A_s = 633 891 ft²
A_a = 734 690 ft²
95.5°
3000 ELEV
2000 ELEV
3-SHAFT PIT
INCO LIMITED

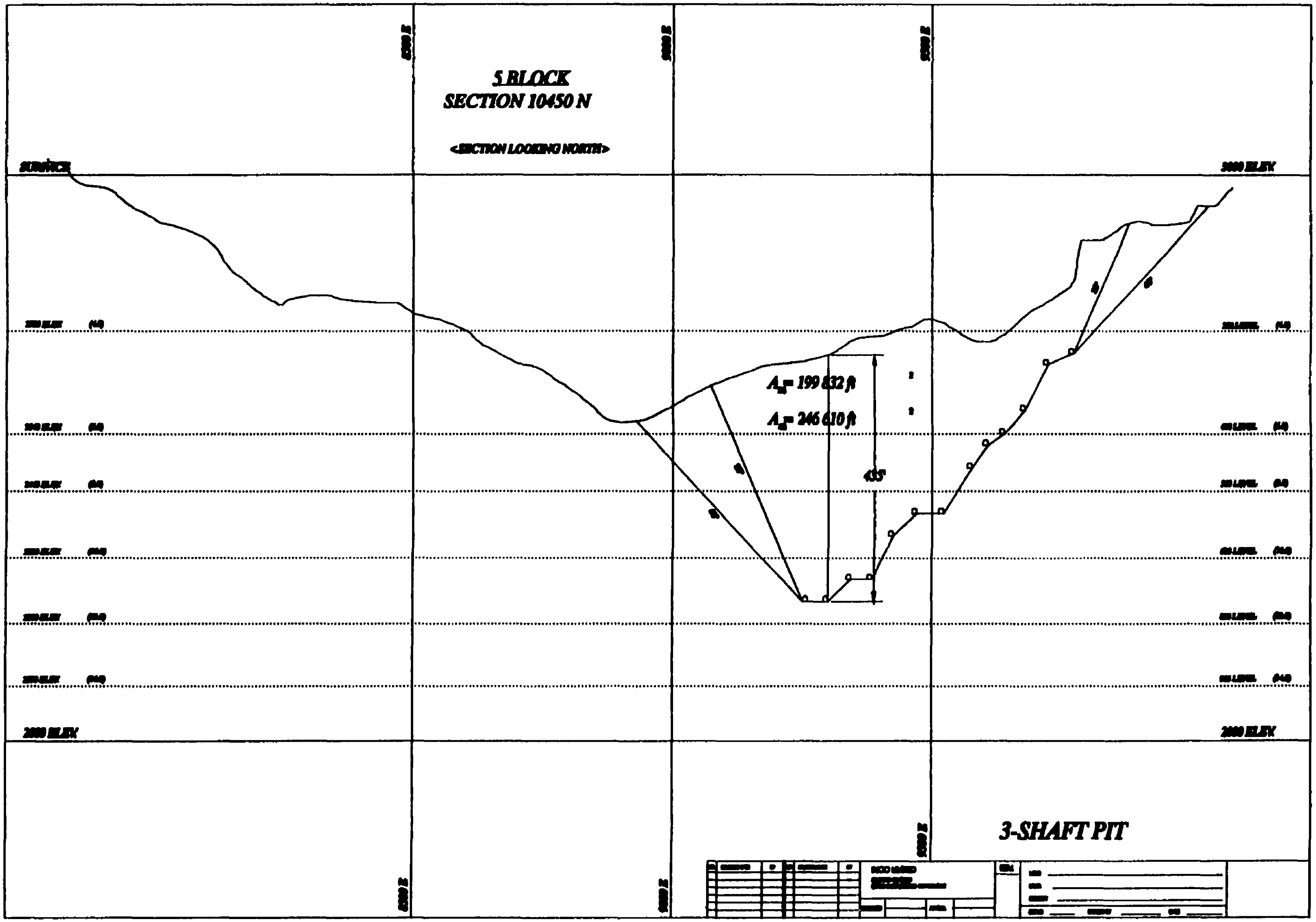

2 BLOCK
SECTION 10000 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
2000 ELEV
$A_{..} = 624\,000$ ft²
$A_{..} = 719\,264$ ft²
93.5°
3-SHAFT PIT


2 BLOCK
SECTION 10050 N
<SECTION LOOKING NORTH>
SURFACE
A_s = 590 185 ft²
A_d = 697 257 ft²
950'
3-SHAFT PIT

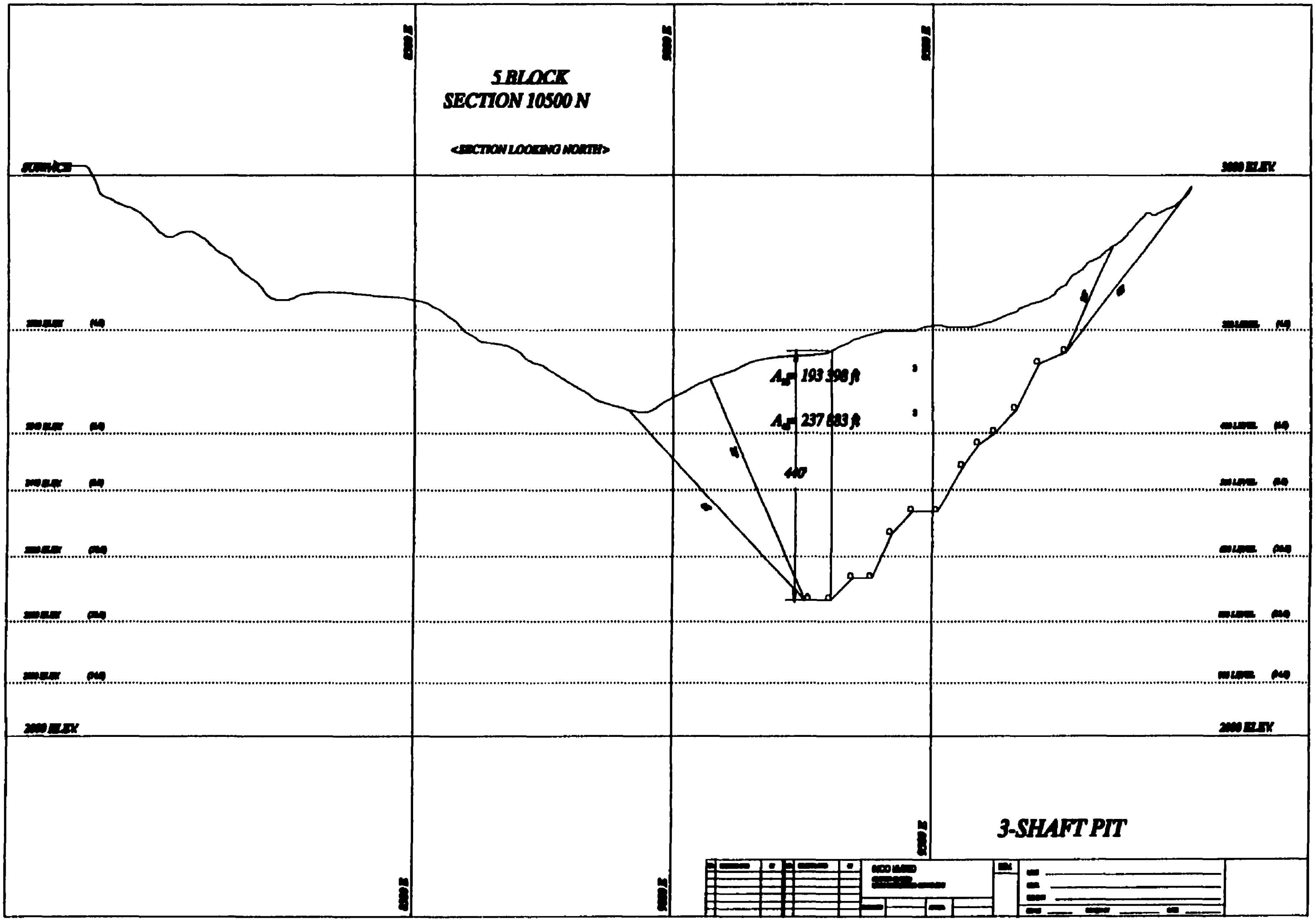


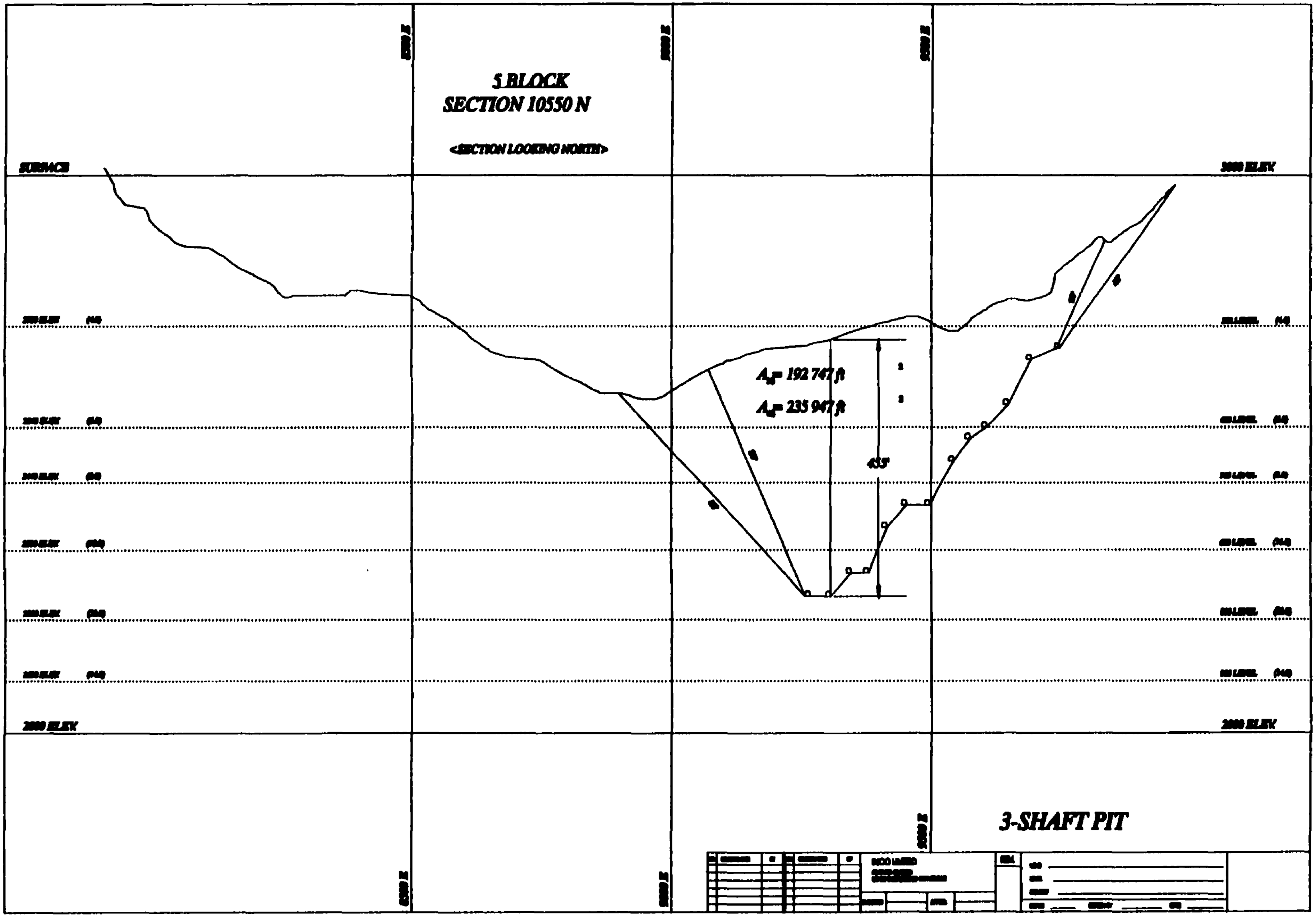
2 BLOCK
SECTION 10100 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
A_s = 261 011 ft
A_c = 497 135 ft
740'
3-SHAFT PIT

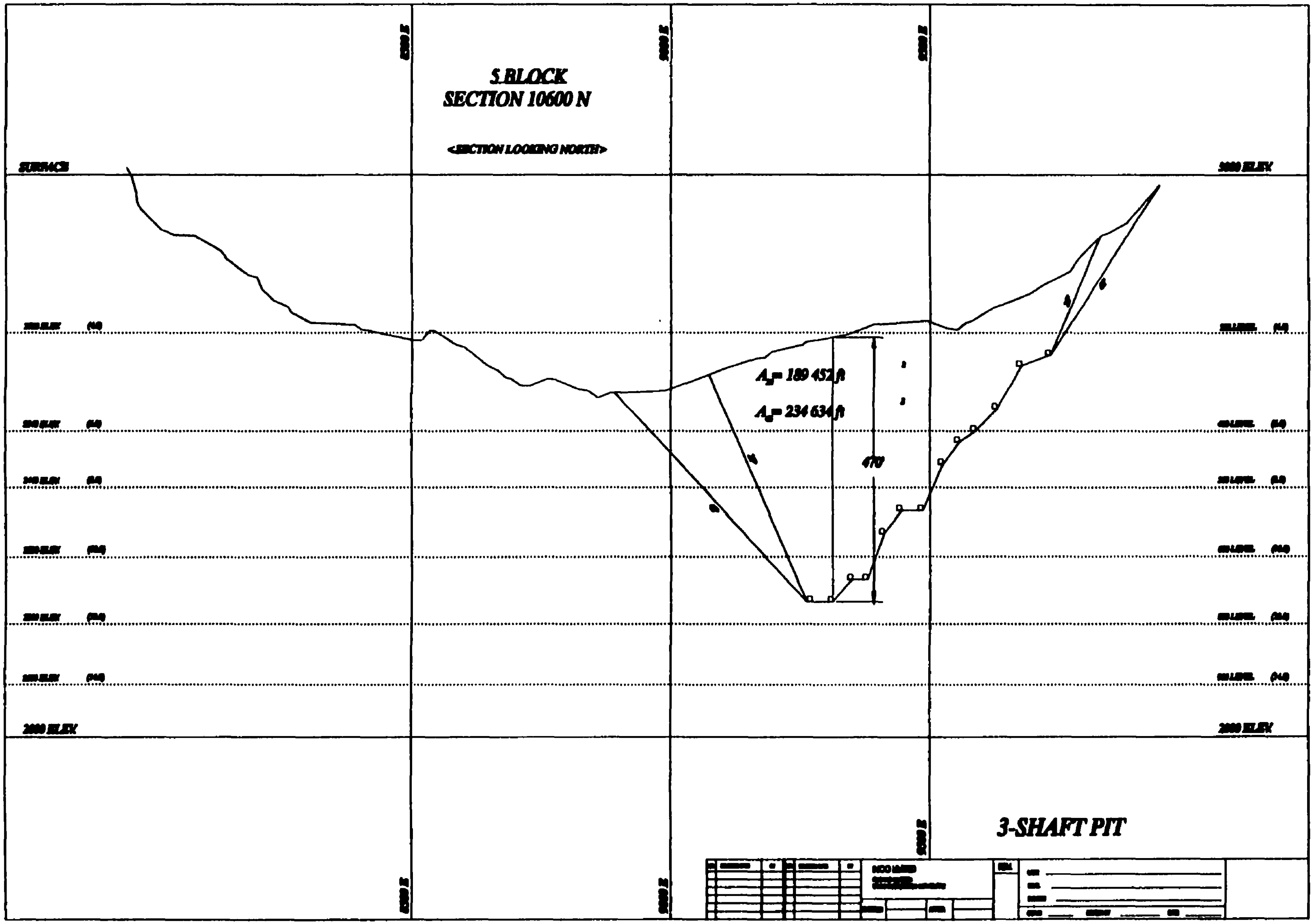

5 BLOCK
SECTION 10400 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV.
2000 ELEV.
2000 ELEV.
3-SHAFT PIT



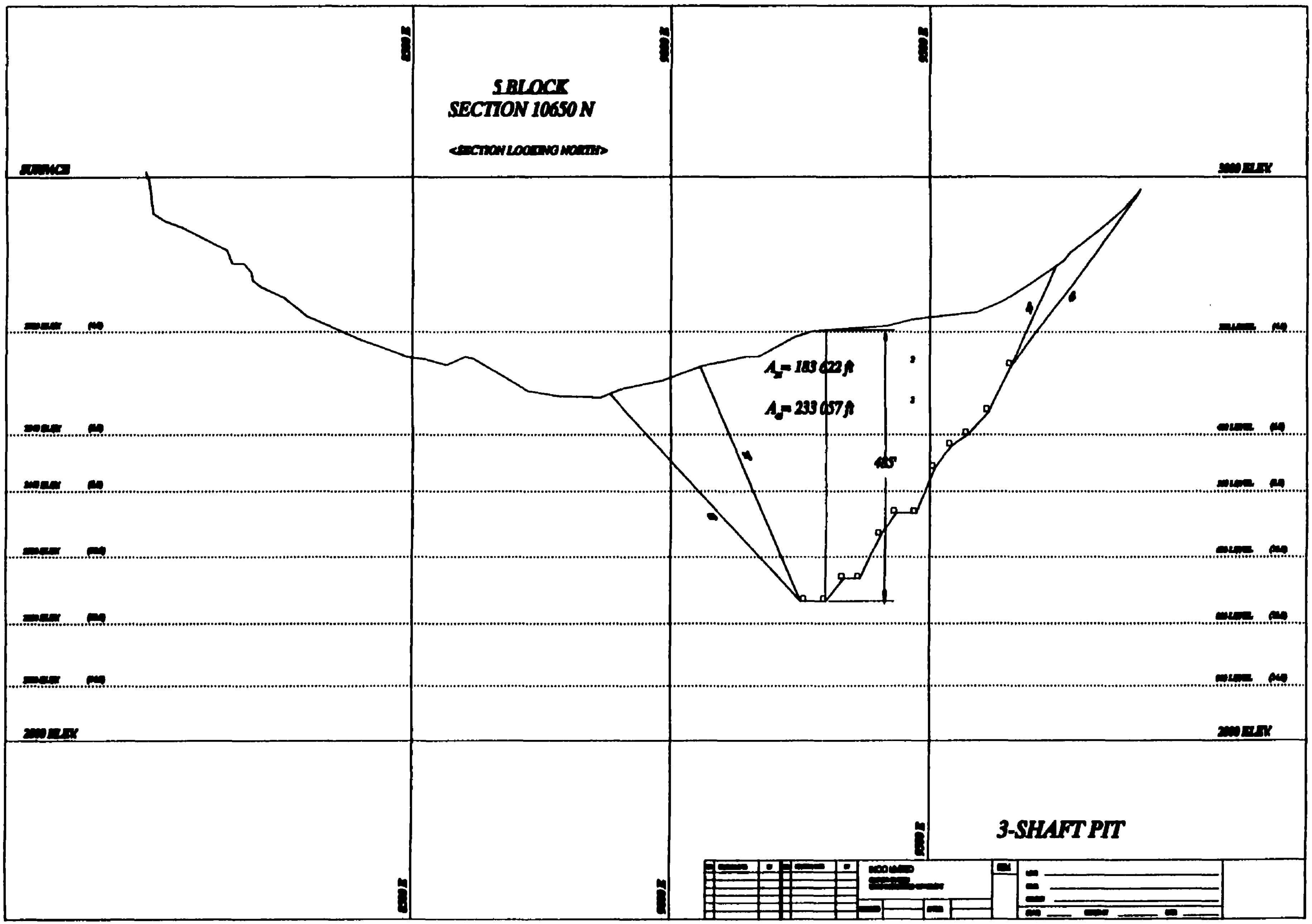
5 BLOCK
SECTION 10450 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
2000 ELEV
A = 199 832 ft
A = 246 610 ft
435'
3-SHAFT PIT


5 BLOCK
SECTION 10500 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV.
2000 ELEV
2000 ELEV.
440'
3-SHAFT PIT

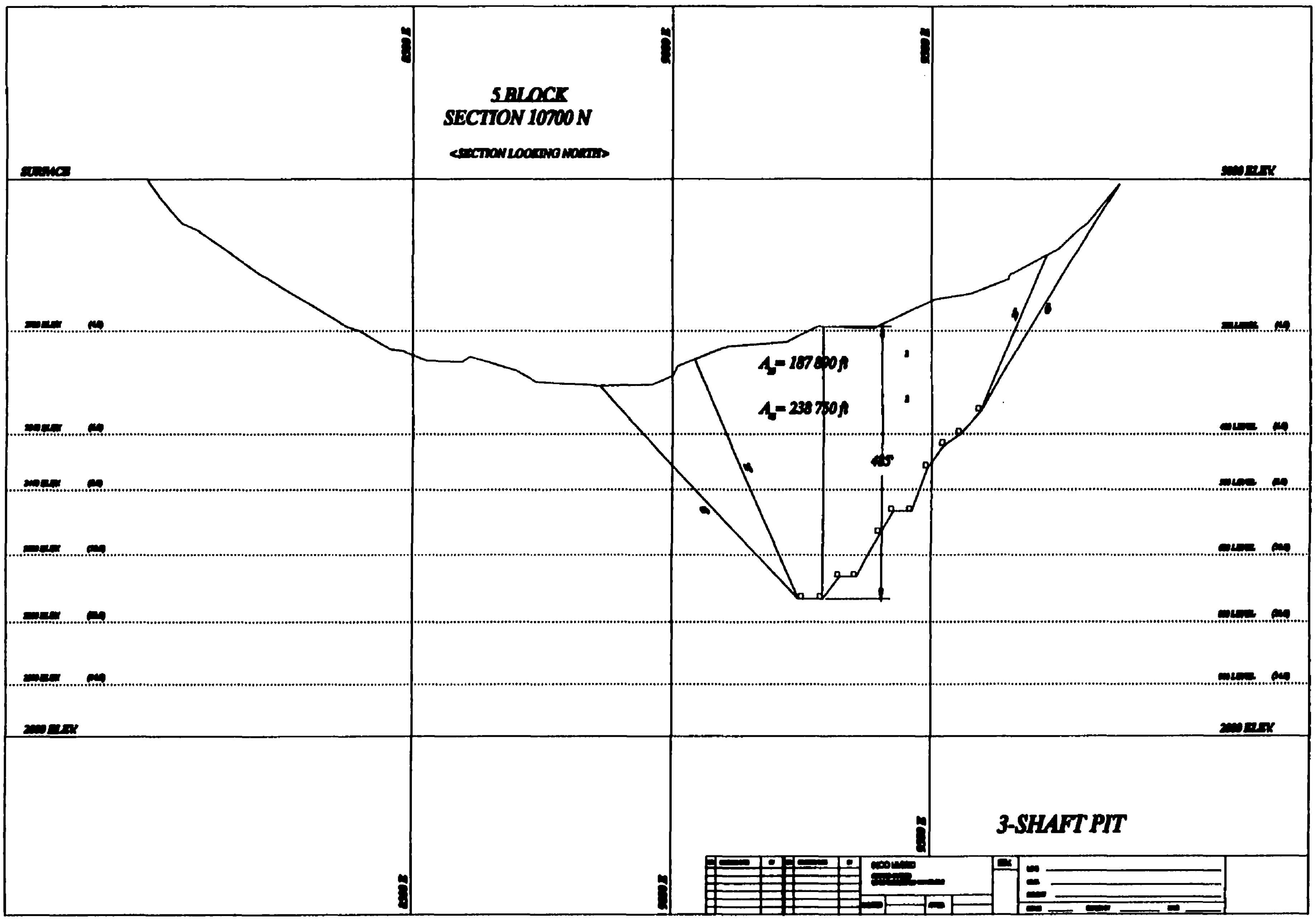

5 BLOCK
SECTION 10550 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
2000 ELEV
A = 192 747 ft
A = 235 947 ft
45.5°
3-SHAFT PIT

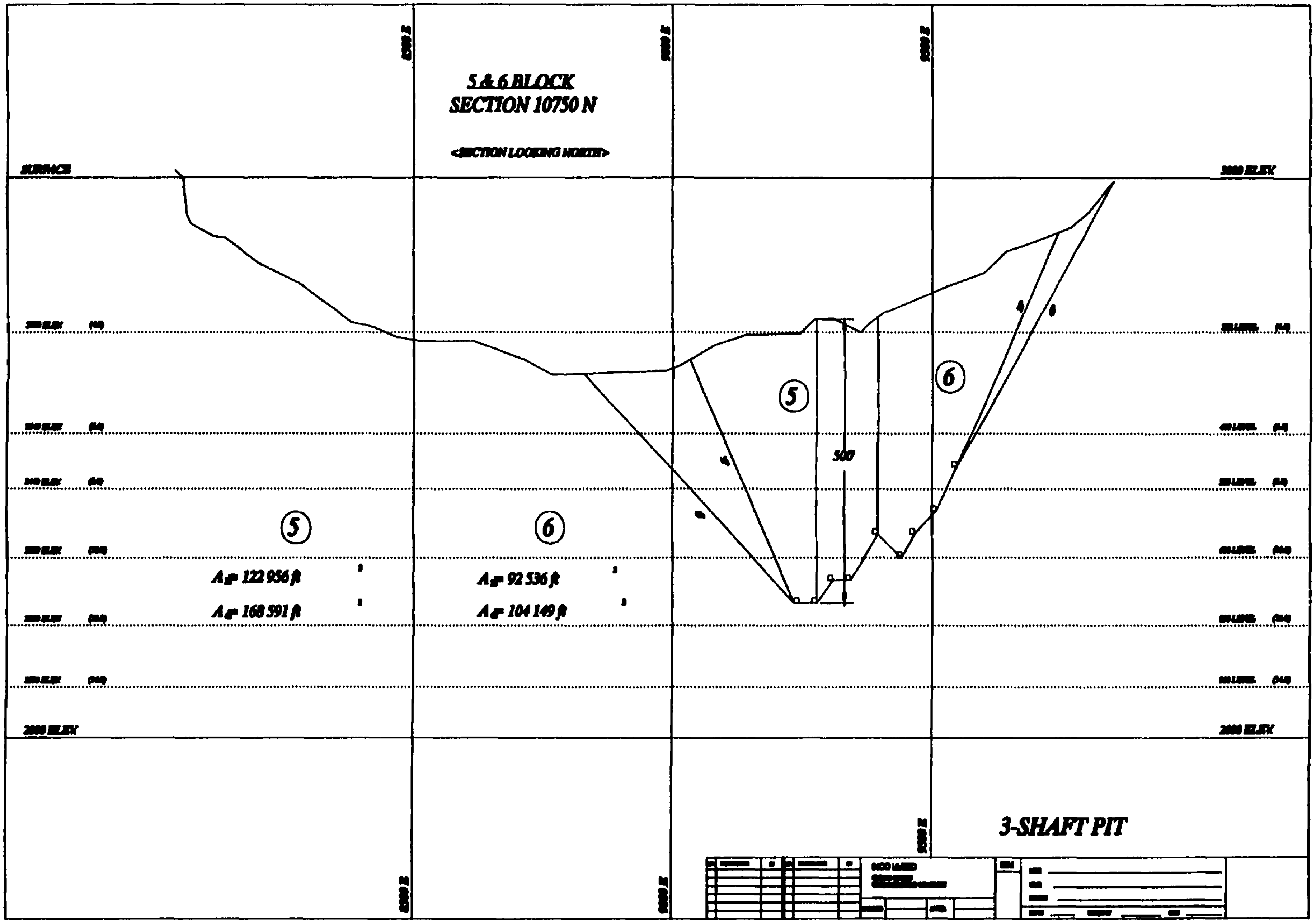


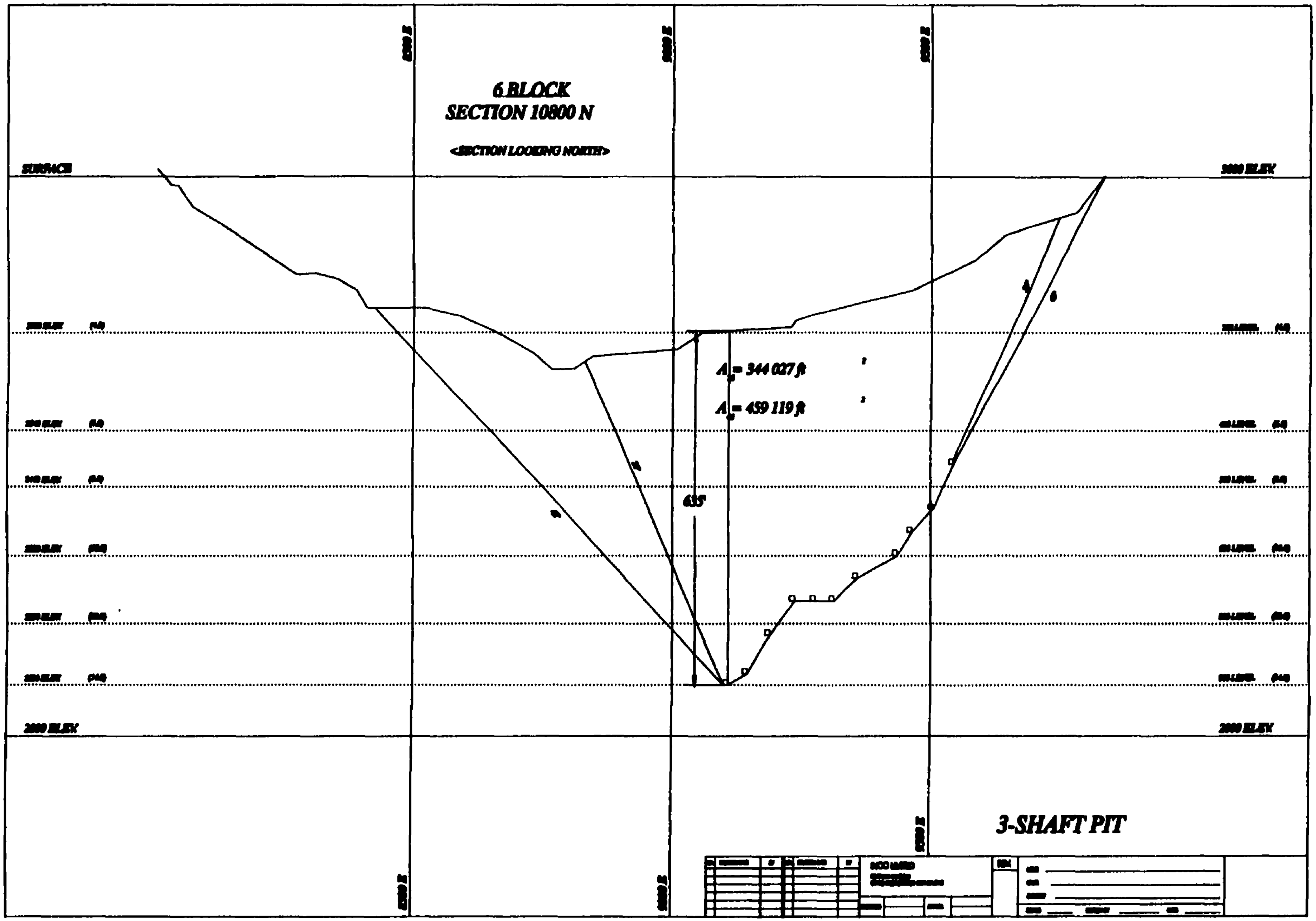
5 BLOCK
SECTION 10600 N
3-SHAFT PIT
2000 ELEV
470'

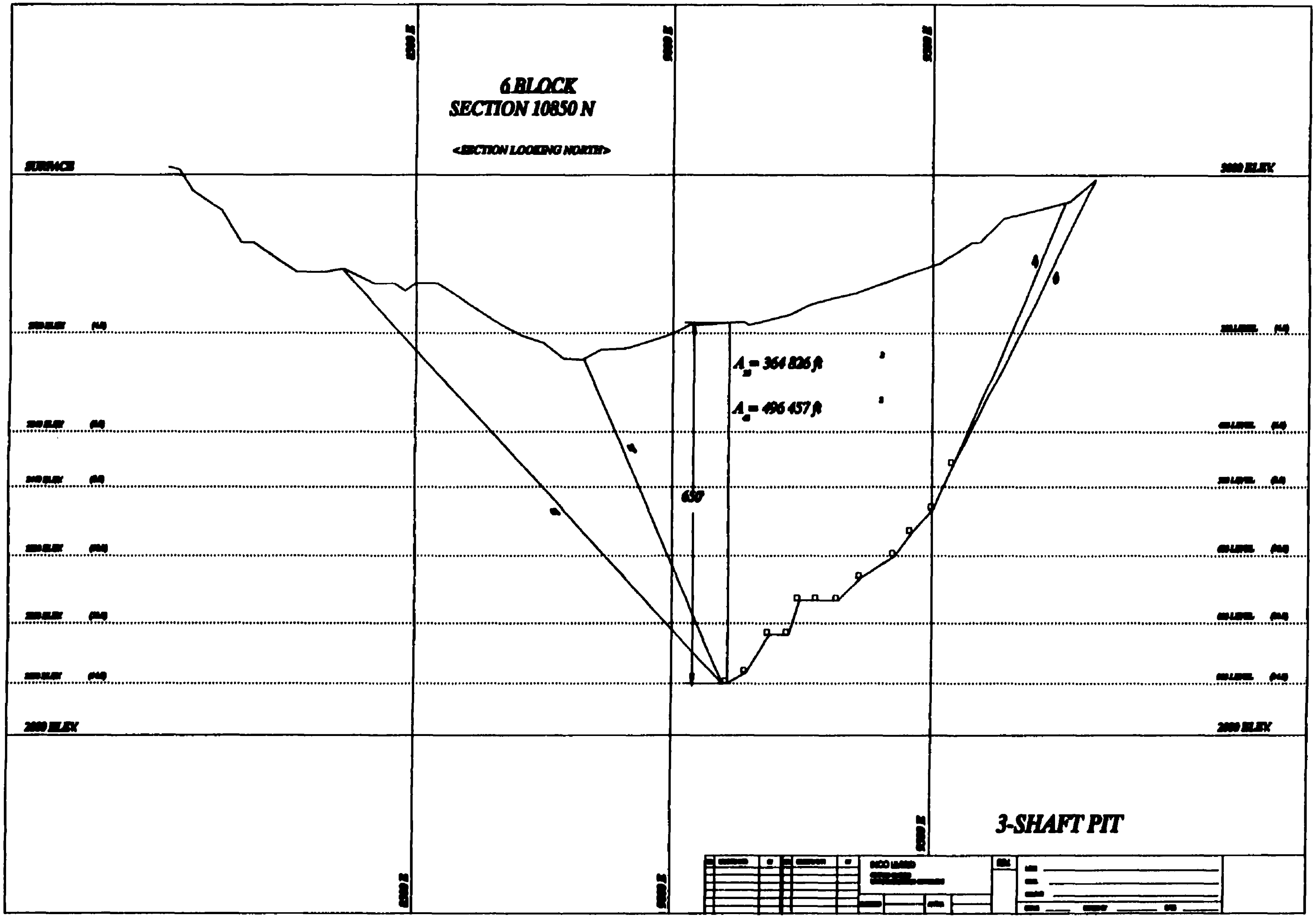


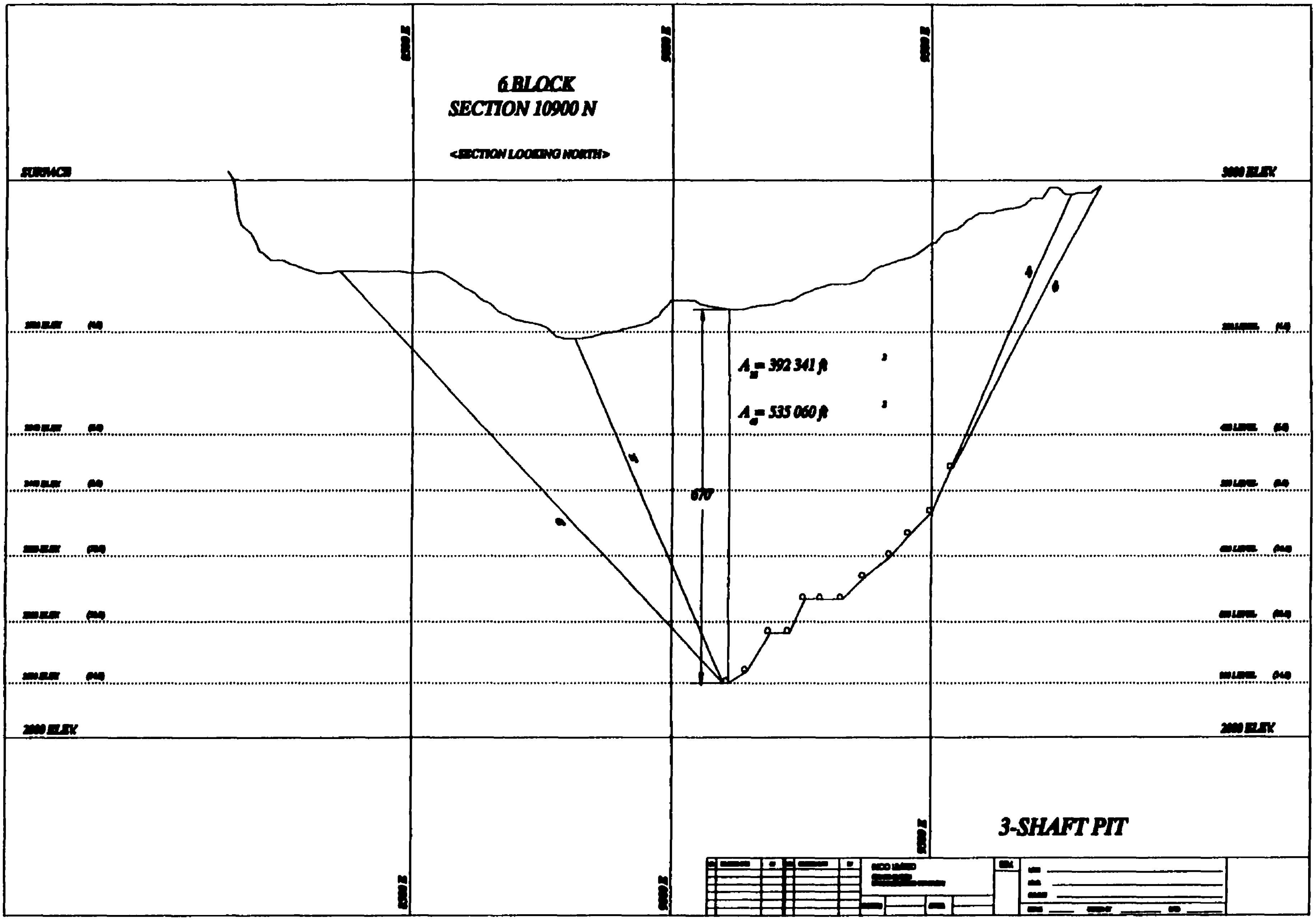
5 BLOCK
SECTION 10650 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
A_{st}= 183 622 ft
A_{sl}= 233 057 ft
485'
2000 ELEV
3-SHAFT PIT

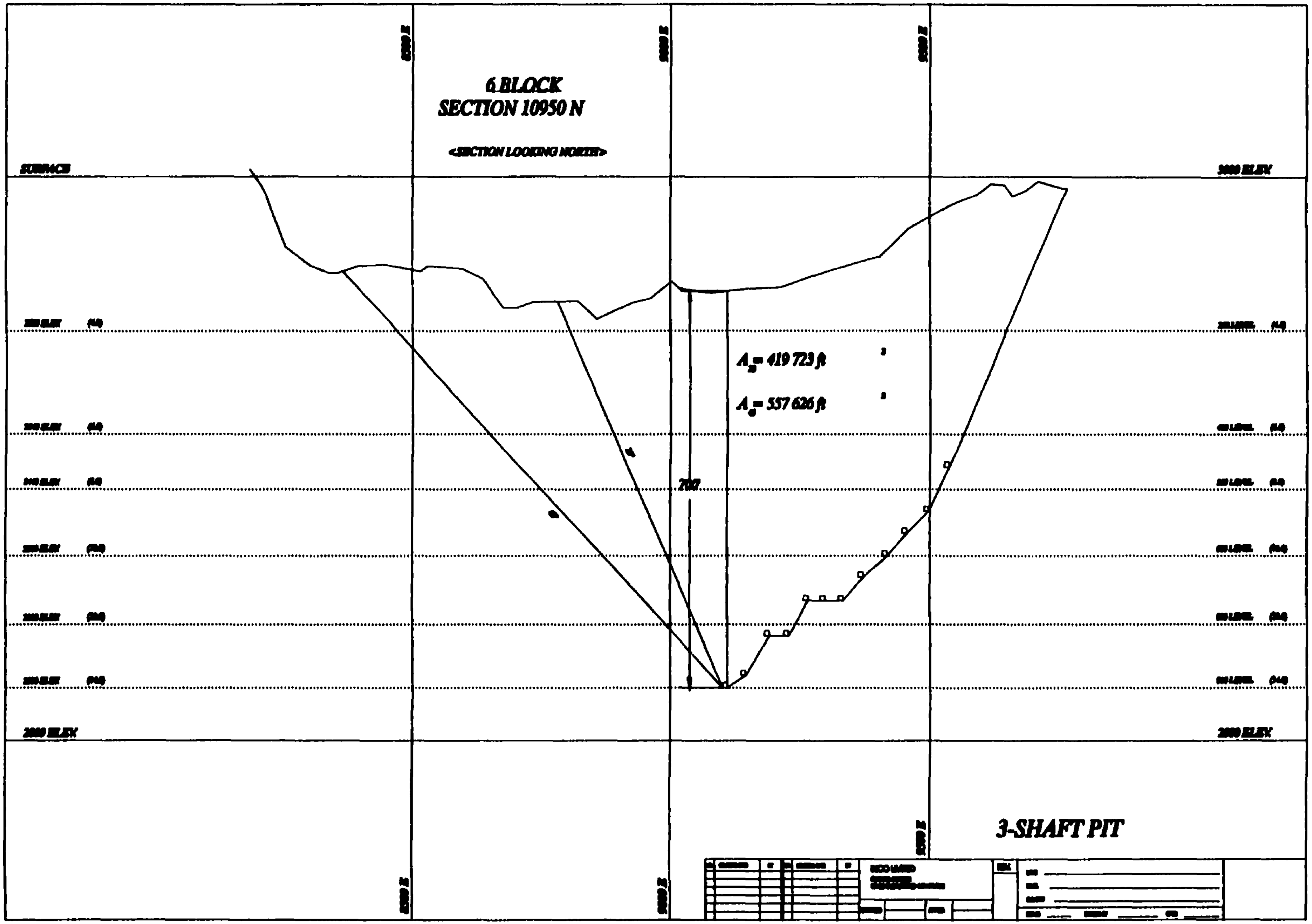

5 BLOCK
SECTION 10700 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV.
2000 ELEV.
3-SHAFT PIT
485'

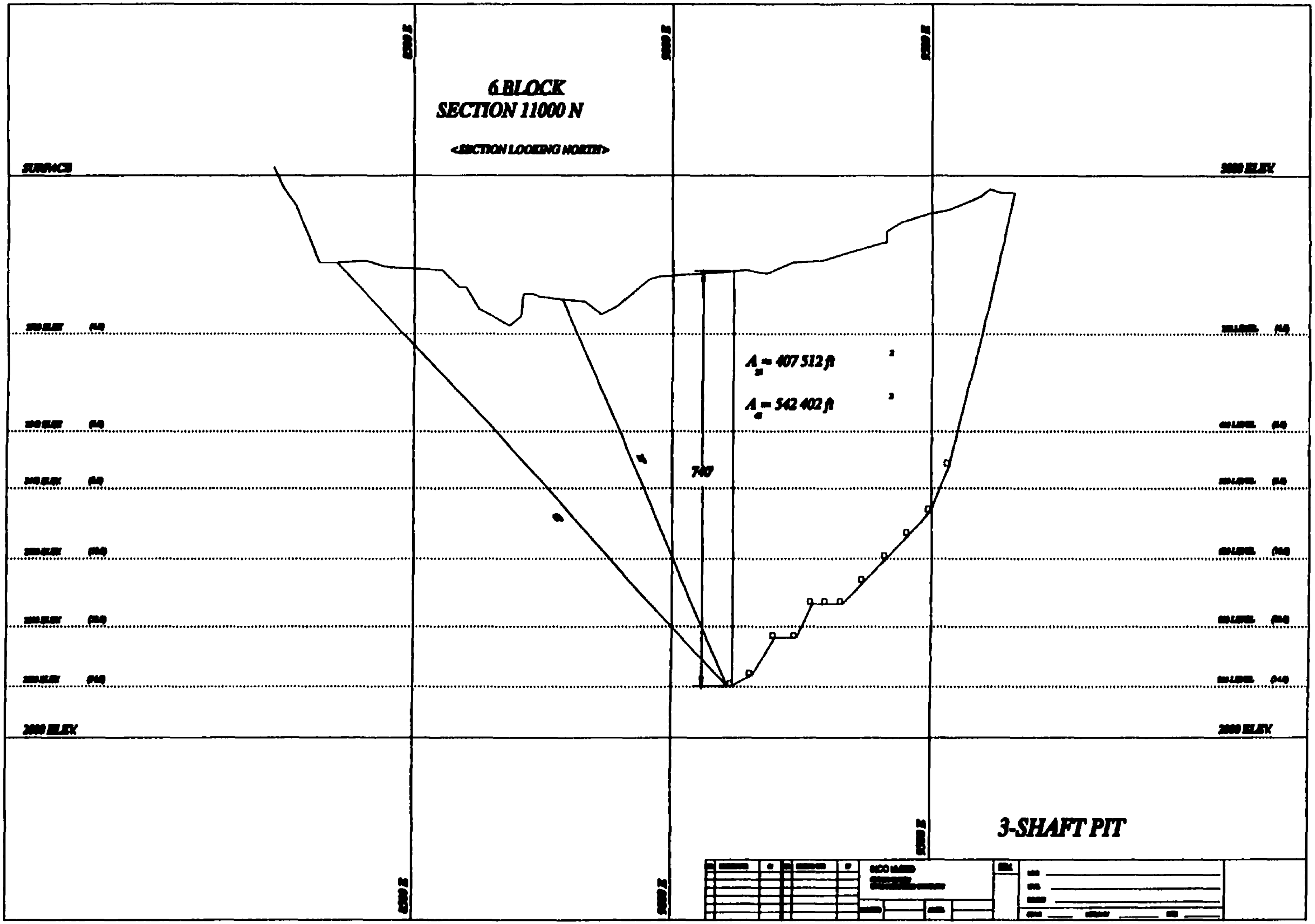

5 & 6 BLOCK
SECTION 10750 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
5
6
500
A_{g}= 122 956 ft^{2}
A_{g}= 168 591 ft^{2}
A_{g}= 92 536 ft^{2}
A_{g}= 104 149 ft^{2}
3-SHAFT PIT

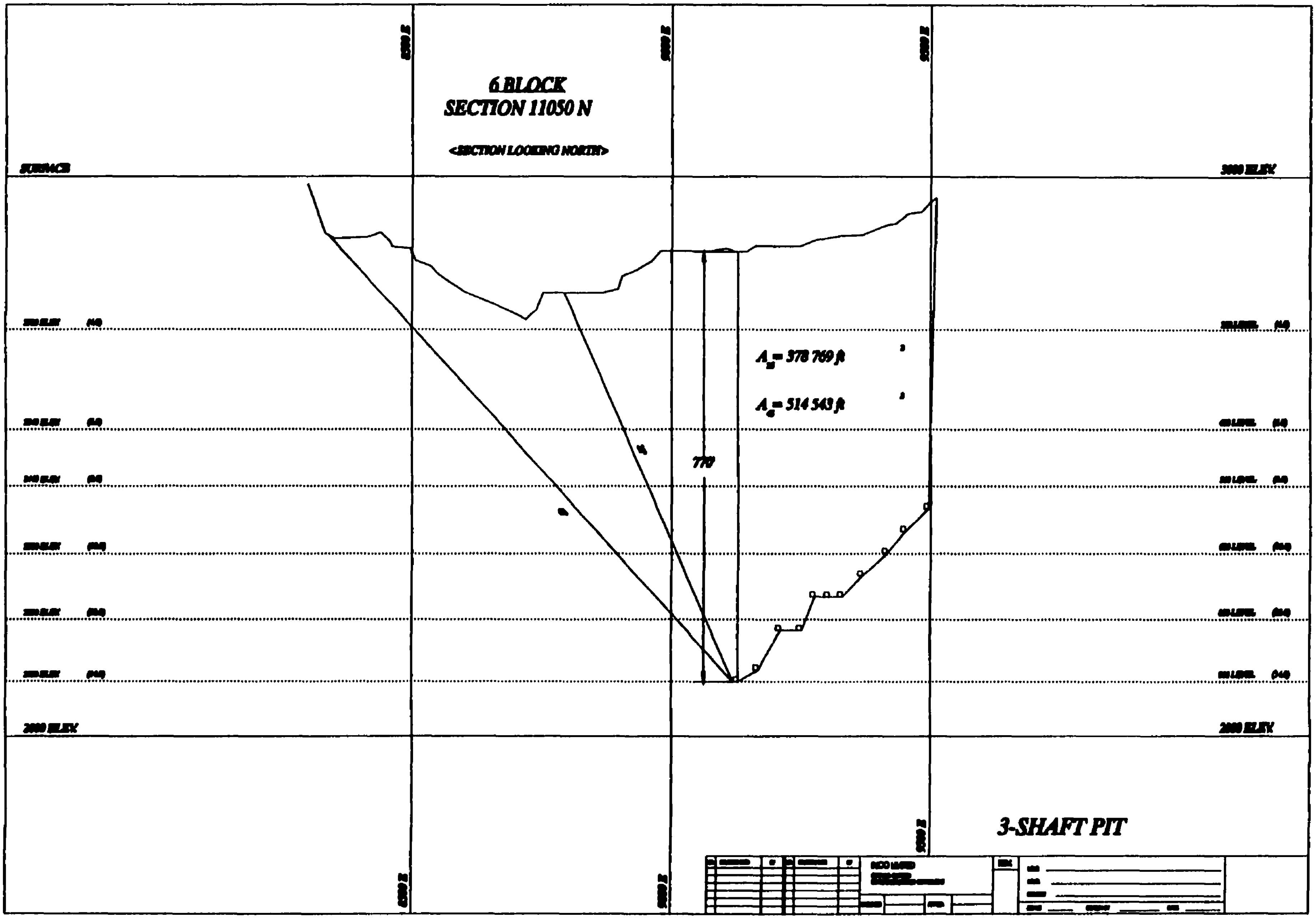

6 BLOCK
SECTION 10800 N
<SECTION LOOKING NORTH>
SURFACE
2800 ELEV
A_u = 344 027 ft^2
A_d = 459 119 ft^2
635'
3-SHAFT PIT

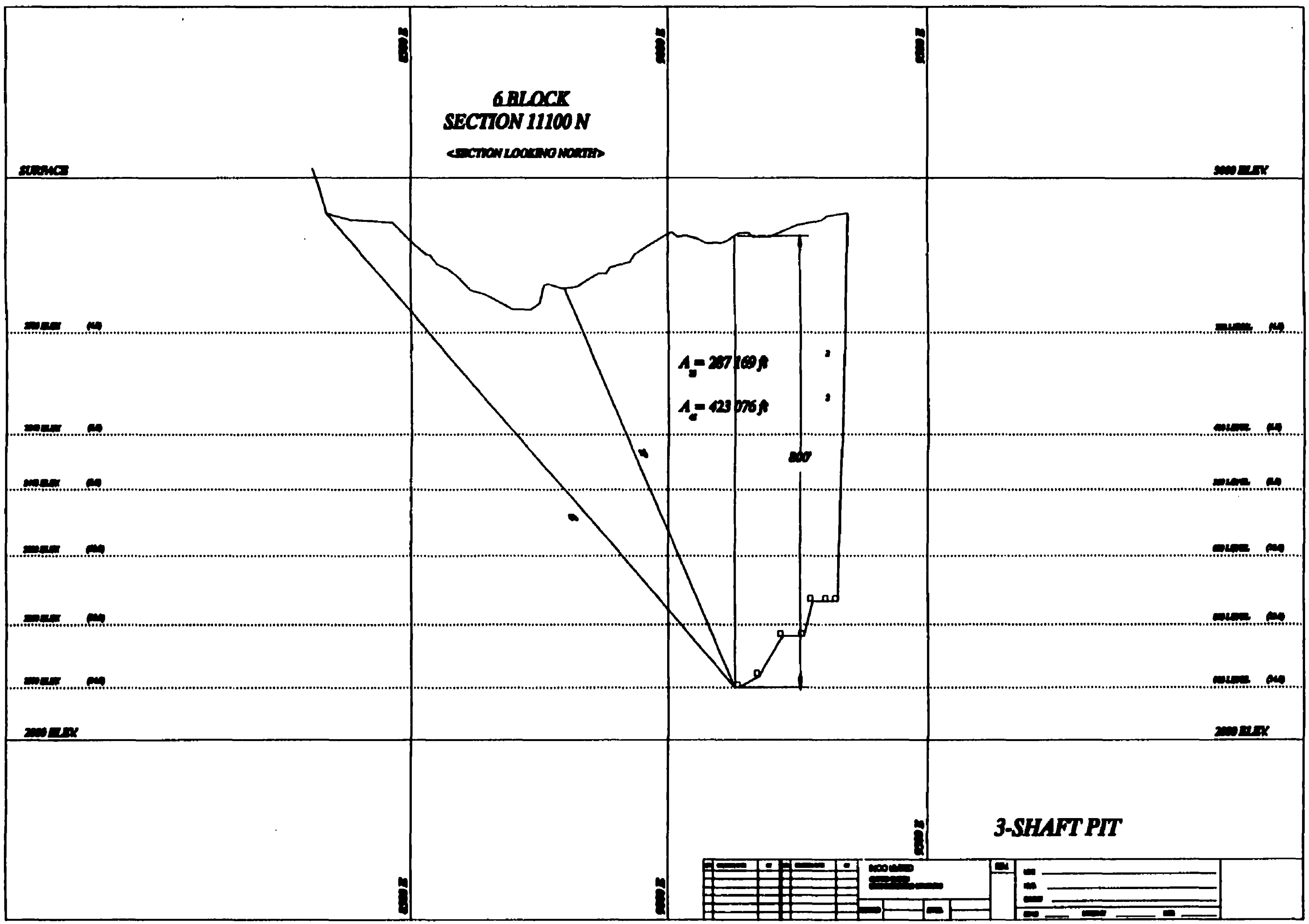

6 BLOCK
SECTION 10850 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV.
2000 ELEV.
A_{st} = 364 826 ft
A_{s} = 496 457 ft
650
3-SHAFT PIT


6 BLOCK
SECTION 10900 N
<SECTION LOOKING NORTH>
SURFACE
3000 ELEV
A_{m} = 392 341 ft ^{2}
A_{c} = 535 060 ft ^{2}
670'
2000 ELEV
2000 ELEV
3-SHAFT PIT


6 BLOCK
SECTION 10950 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
A_{β}= 419 723 ft
A_{ϕ}= 557 626 ft
707
3-SHAFT PIT


6 BLOCK
SECTION 11000 N
<SECTION LOOKING NORTH>
SURFACE
A_s = 407 512 ft²
A_a = 542 402 ft²
740
2000 ELEV.
3-SHAFT PIT


6 BLOCK
SECTION 11050 N
<SECTION LOOKING NORTH>
A_m = 378 769 ft²
A_c = 514 543 ft²
770
3-SHAFT PIT


6 BLOCK
SECTION 11100 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
A_s = 287 169 ft²
A_d = 423 076 ft²
3-SHAFT PIT

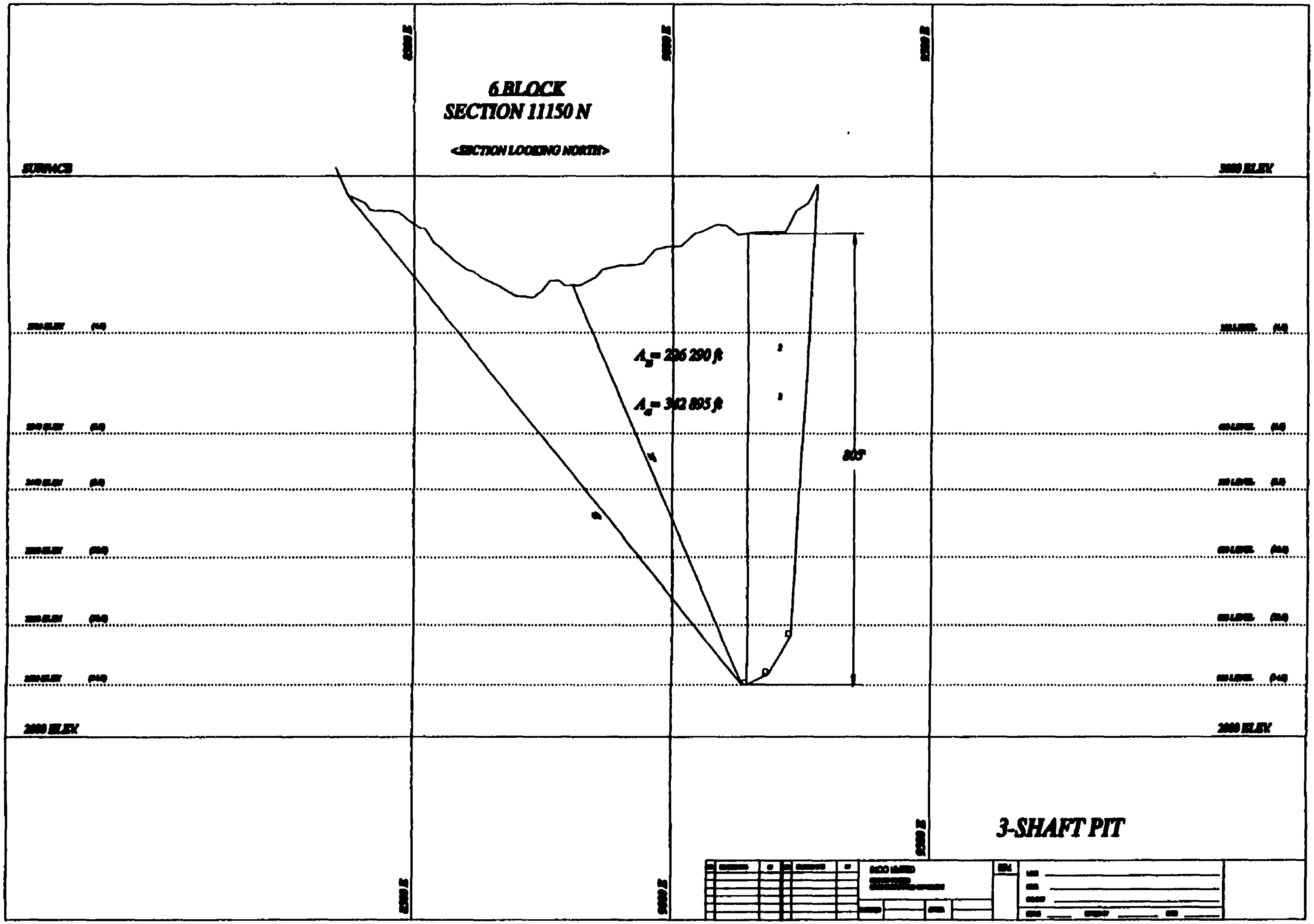

6 BLOCK
SECTION 11150 N
<SECTION LOOKING NORTH>
SURFACE
2000 ELEV
805'
3-SHAFT PIT

Table 4: 3 Shaft Pit
Broken Rock Tonnage Determination
(Angle of Draw = 25 deg)

		1 BLOCK			2 BLOCK			3 BLOCK			5 BLOCK	
SECTION	AREA (sqft)	INFLUENCE (ft)	VOLUME (cuft)	AREA (sqft)	INFLUENCE (ft)	VOLUME (cuft)	AREA (sqft)	INFLUENCE (ft)	VOLUME (cuft)	AREA (sqft)	INFLUENCE (ft)	VOLUME (cuft)
9050	45,401	50	2,270,050									
9100	93,094	50	4,654,700									
9150	118,095	50	5,904,750									
9200	133,434	50	6,671,700									
9250	154,541	50	7,727,050									
9300	194,852	50	9,742,600									
9350	182,620	50	9,131,000	42,230	50	2,111,500						
9400	139,475	50	6,973,750	130,535	50	6,526,750						
9450	106,546	50	5,327,300	168,985	50	8,449,250						
9500	65,516	50	3,275,800	219,325	50	10,966,250						
9550	21,335	50	1,066,750	329,931	50	16,496,550						
9600				384,965	50	19,248,250						
9650				505,187	50	25,259,350						
9700				521,368	50	26,068,400						
9750				598,450	50	29,922,500						
9800				756,437	50	37,821,850						
9850				644,114	50	32,205,700						
9900				633,303	50	31,665,150						
9950				633,891	50	31,694,550						
10000				624,000	50	31,200,000						
10050				590,185	50	29,509,250						
10100				261,011	50	13,050,550						
10400							145,022	50	7,251,100			
10450							199,832	50	9,991,600			
10500							193,396	50	9,669,800			
10550							192,747	50	9,637,350			
10600							189,452	50	9,472,600			
10650							183,622	50	9,181,100			
10700							187,880	50	9,394,000			
10750							122,958	50	6,147,900	92,536	50	4,626,800
10800										344,027	50	17,201,350
10850										364,826	50	18,241,300
10900										392,341	50	19,617,050
10950										419,723	50	20,986,150
11000										407,512	50	20,375,600
11050										378,769	50	18,938,450
11100										287,169	50	14,358,450
11150										226,290	50	11,314,500
Total Block Volume (millions cuft)			62.7			352.2			70.7			146.7
Total Block Tonnage (millions tons) (@ 18 cuft/ton)			3.5			19.6			3.9			8.1
Total Pit Tonnage		35.1 million tons										

Table 5: 3 Shaft Pit
Broken Rock Tonnage Determination
(Angle of Draw = 45 deg)

		1 BLOCK			2 BLOCK			3 BLOCK			6 BLOCK	
SECTION	AREA (sqft)	INFLUENCE (ft)	VOLUME (cuft)	AREA (sqft)	INFLUENCE (ft)	VOLUME (cuft)	AREA (sqft)	INFLUENCE (ft)	VOLUME (cuft)	AREA (sqft)	INFLUENCE (ft)	VOLUME (cuft)
9050	52,901	50	2,645,050									
9100	99,594	50	4,979,700									
9150	122,757	50	6,137,850									
9200	136,731	50	6,836,550									
9250	155,458	50	7,772,900									
9300	200,002	50	10,000,100									
9350	182,620	50	9,131,000	49,285	50	2,464,250						
9400	139,475	50	6,973,750	137,944	50	6,897,200						
9450	106,546	50	5,327,300	176,665	50	8,833,250						
9500	65,516	50	3,275,800	227,482	50	11,374,100						
9550	21,335	50	1,066,750	333,507	50	16,675,350						
9600				367,065	50	18,353,250						
9650				525,967	50	26,298,350						
9700				543,490	50	27,174,500						
9750				623,767	50	31,188,350						
9800				773,507	50	38,675,350						
9850				736,915	50	36,845,750						
9900				737,628	50	36,881,400						
9950				734,690	50	36,734,500						
10000				719,264	50	35,963,200						
10050				697,257	50	34,862,850						
10100				497,135	50	24,856,750						
10400							211,444	50	10,572,200			
10450							246,610	50	12,330,500			
10500							237,883	50	11,894,150			
10550							235,947	50	11,797,350			
10600							234,634	50	11,731,700			
10650							233,057	50	11,652,850			
10700							238,750	50	11,937,500			
10750							168,591	50	8,429,550	104,149	50	5,207,450
10800										459,119	50	22,955,950
10850										490,457	50	24,522,850
10900										535,060	50	26,753,000
10950										557,626	50	27,881,300
11000										542,402	50	27,120,100
11050										514,543	50	25,727,150
11100										423,076	50	21,153,800
11150										342,895	50	17,144,750
Total Block Volume (millions cuft)			64.1			396.2			90.3			198.8
Total Block Tonnage (millions tons) (@ 18 cuft/ton)			3.6			22.0			5.0			11.0
Total Pit Tonnage		41.6 million tons										

Table 6: 3 Shaft Pit
Pit Surface Area Determination
(Angle of Draw = 25 deg)

	1 BLOCK			2 BLOCK			3 BLOCK			4 BLOCK		
SECTION	LENGTH (ft)	INFLUENCE (ft)	SURF. AREA (sqft)	LENGTH (ft)	INFLUENCE (ft)	SURF. AREA (sqft)	LENGTH (ft)	INFLUENCE (ft)	SURF. AREA (sqft)	LENGTH (ft)	INFLUENCE (ft)	SURF. AREA (sqft)
9050	310	50	15,500									
9100	450	50	22,500									
9150	530	50	26,500									
9200	640	50	32,000									
9250	720	50	36,000									
9300	650	50	32,500									
9350	420	50	21,000	380	50	19,000						
9400	230	50	11,500	610	50	30,500						
9450	190	50	9,500	650	50	32,500						
9500	150	50	7,500	770	50	38,500						
9550	50	50	2,500	1,030	50	51,500						
9600				1,150	50	57,500						
9650				1,260	50	63,000						
9700				1,360	50	68,000						
9750				1,500	50	75,000						
9800				1,470	50	73,500						
9850				1,260	50	63,000						
9900				1,320	50	66,000						
9950				1,300	50	65,000						
10000				1,350	50	67,500						
10050				1,300	50	65,000						
10100				840	50	42,000						
10400							770	50	38,500			
10450							1,110	50	55,500			
10500							1,000	50	50,000			
10550							850	50	42,500			
10600							830	50	41,500			
10650							750	50	37,500			
10700							760	50	38,000			
10750							400	50	20,000	390	50	19,500
10800										1,010	50	50,500
10850										1,020	50	51,000
10900										1,060	50	53,000
10950										1,050	50	52,500
11000										940	50	47,000
11050										780	50	39,000
11100										610	50	30,500
11150										560	50	28,000
Total Block Surface Area (sqft)			217,000			877,500			324,000			371,000
Total Pit Surface Area		1.8 million sqft										

Table 7: 3 Shaft Pit
Pit Surface Area Determination
(Angle of Draw = 45 deg)

		1 BLOCK			2 BLOCK			5 BLOCK			6 BLOCK	
SECTION	LENGTH (ft)	INFLUENCE (ft)	SURF. AREA (sqft)	LENGTH (ft)	INFLUENCE (ft)	SURF. AREA (sqft)	LENGTH (ft)	INFLUENCE (ft)	SURF. AREA (sqft)	LENGTH (ft)	INFLUENCE (ft)	SURF. AREA (sqft)
9050	390	50	19,500									
9100	530	50	26,500									
9150	580	50	29,500									
9200	680	50	34,000									
9250	790	50	39,500									
9300	790	50	39,500									
9350	420	50	21,000	490	50	24,500						
9400	230	50	11,500	750	50	37,500						
9450	190	50	9,500	830	50	41,500						
9500	150	50	7,500	920	50	46,000						
9550	50	50	2,500	1,080	50	54,000						
9600				1,170	50	58,500						
9650				1,410	50	70,500						
9700				1,560	50	78,000						
9750				1,700	50	85,000						
9800				1,630	50	81,500						
9850				1,800	50	90,000						
9900				1,890	50	94,500						
9950				1,880	50	94,000						
10000				1,910	50	95,500						
10050				1,880	50	94,000						
10100				1,850	50	92,500						
10400							1,200	50	60,000			
10450							1,440	50	72,000			
10500							1,330	50	66,500			
10550							1,210	50	60,500			
10600							1,180	50	59,000			
10650							1,140	50	57,000			
10700							1,150	50	57,500			
10750							610	50	30,500	540	50	27,000
10800										1,580	50	79,000
10850										1,630	50	81,500
10900										1,630	50	81,500
10950										1,500	50	75,000
11000										1,470	50	73,500
11050										1,310	50	65,500
11100										1,150	50	57,500
11150										1,060	50	53,000
Total Block Surface Area (sqft)			340,500			1,137,500			463,000			593,500
Total Pit Surface Area		2.4 million sqft										

APPENDIX 2

Report of
Creighton Heat Load Assessment
As Per 1993 Survey – Inco's Internal Report

The datum horizon used is 800 level; the volume is 880,000 (FM at 0.079), and production 5100 tons per day.

Weighted Return Air Temperature	= 57.5°F W.B.	Σ = 24.66 Btu/Lb.
Fresh Air Temperature	= 35.2°F W.B.	Σ = 13.15 Btu/Lb

Sigma Heat Content Return Air	1,714,000 Btu/Min.
Sigma Heat Content Fresh Air	914,000 Btu/Min.
Heat Removed by the Air	800,000 Btu/Min.
Calculated Heat Loads:	
(a) Fans – Intake)	
- Orebody) 5195 HP	220,000 Btu/Min.
Underground Compressors – 3980 HP	169,000 Btu/Min.
(b) Diesel Equipment – Average Fuel per Day (787 Imp. Gal.) (= 3580 l); (156,000 Btu/Gal.) (Day = 24 Hrs.)	85,000 Btu/Min.
(c) Electrical Load Centres – A Total of 7870 KVA (7870 x 0.07 : 0.746) x 42.4	31,000 Btu/Min.
(d) Body Metabolism – Average of 300 Men at 1000 Btu/Hr.	5,000 Btu/Min.
(e) Cement in Sandfill (Refer to 79 Data)	5,500 Btu/Min.
(f) Pumps (List Levels of Install) 15% (of Average 1372 HP)	8,500 Btu/Min.
(g) Fresh Water – Average 100 Imp. Gal./Min	12,000 Btu/Min.
Net Heat Load Calculated	**512,000 Btu/Min.**
Heat Removed by Air	800,000 Btu/Min.
Net Calculated Heat Load	512,000 Btu/Min.
Therefore – Heal of Wall Rock	288,000 Btu/Min

Heat Load Formula For Lower Level Operations

A basic formula is being used at Creighton Mine to predict future heat loads, depending on the daily tonnage to be mined, the difference between the virgin rock temperature and the fresh air station wet bulb (both at the mean mining depth), and a factor that is determined from the heat load of present mining operations, Stachulak (1978). The factor is applied, along with the predicted temperatures, to determine future heat loads at greater depths; data is in British units.

$$\text{Heat Load Factor} = \frac{\text{HeatLoad, Bt/Min.}}{\text{Tons/Day(VRT} - \text{Station}^\circ\text{F, W.B.}}$$

Two heat load checks of lower mining operations were made in 1993; one for mining below 6000 level (6200 – 7200), mean mining depth 6900 level, and the other for the whole mine (4600 – 7200), mean mining depth of 6200 level.

Mass flow of dry air was calculated at 38,763 Lb./Min., and 69,353 Lb./Min. respectively.

(a) 6000 Level Datum Horizon (For Mining 6200 – 7200L)

Return Air Temperature = 70.6°F W.B. = (Σ H = 31 Btu/Lb.)
From Below 6000 L (Excludes Compressors)
Sigma Heat Content of Return Air 1,200,000 Btu/Min.

Fresh Air Temperature = 57.7°F W.B. = (ΣH = 22.7 Btu/Lb.)
#8 Shaft + #9 Shaft
Sigma Heat Content of Fresh Air 879,000 Btu/Min.

Heat Removed by Air = 321,000 Btu/Min.

Production (From 6200 – 7200L) = 2,896 Tons/Day

Mean Mining Depth = 6900L, (6200 – 7200L)

VRT at Mean Mining Depth = 107°F

Station W.B. Temp #8 and #9 Shafts at 6900L = 61°F (Actual Measured)

From the Above:

$$\text{Heat Load Factor Estimate} = \frac{321{,}000\text{Btu/Min.}}{2{,}896\text{Tons/Day}(107-61)} = 2.4$$

(For Mean Mining Depth at 6900L)

(b) 6000 Level Datum Horizon (For Mining 4600 – 7200L)

Return Air Temperature = 73.1/78.4° F W.B. = (ΣH = 32.85 Btu/Lb.)
(Includes Heat From Compressors
Sigma Heat of Return Air — 2,278,000 Btu/Min.

Fresh Air Temperature = 57.7° F W.B. = (ΣH = 2.7 Btu/Lb.)
#8 Shaft + #9 Shaft
Sigma Heat of Return Air — 1,582,000 Btu/Min.

Heat Removed by Air = 696,000 Btu/Min.

Mass Flow of Dry Air = 69,353 Lb./Min.

Production Rate (Whole Mine 4600 – 7200L) = 5,100 Tons/Day

Mean Mining Depth = 6200L

VRT at Mean Mining Depth = 100°F

Station W.B. Temp. at 6200 L = 58.3°F

(#8 and #9 Shaft Station)

Average Heat of Compressors = 169,000 Btu/Min.

From the Above:

$$\text{Heat Load Factor Estimate} = \frac{(696{,}000 - 169{,}000)\text{Btu/Min.}}{5{,}100\text{Tons/Day}(100 - 58.3)} = 2.5$$

Note: Further to heat removal assessment on 800 and 6000 levels, an additional heat removal check was made on 5800 and 4200 levels. The heat removal values differ to some degree at all these four locations, namely:

Total Σ Heat Returned (Whole Mine) Btu/Min.	Measuring Method	Level	Fresh Air		Return Air		Mass Flow of Dry Air Lb./Min.	Date and Reference 1993
			W.B. °F	ΣHeat Btu/Lb.	W.B. OF	*Σ Heat Btu/Lb.		
800,000	Indirect	800L	35.2	13.15	57.5	24.66	69,353	03/12.p.12
766,000	Direct	6000L	57.9	23.4	74.5	34.4	69,535	03/17.p.0
761,000	Direct	5800L	57.8		73.6		69,535	03/17.p.0
730,000	Indirect	4200L		20.4		30.9	69,535	03/17.p.1
Avg. 764,000								

* Σ Heat = Sigma Heat Content $(\frac{Btu}{lb})$

Average Heat Removed from the Mine = 764,000 Btu/Min.

Production Rate (Whole Mine) = 5,100 Tons/Day

Mean Mining Depth = 6200 Level

VRT at Mean Mining Depth = 100°F

Station W.B. Temperature at Mean Mining Depth = 58.3°F
(#8 and #9 Shaft Station Temperature)

Average Heat of Compressors = 169,000 Btu/Min.

From the Above:

$$\text{Heat Load Factor Estimate} = \frac{764{,}000 - 169{,}000\text{Btu/Min.}}{5{,}100\text{Tons/Day}(100 - 58.3)} = 2.8$$

Discussion:

The 1993 heat load factor range of 2.4 – 2.8 is some 15 – 30% higher than values obtained during the 1979 survey.

The above figures exclude heat generated by underground compressors, and are based on 75° F W.B. stope exhaust temperature.

Recommendation:

The ventilation planning and airflow allocation below 7000 level is to be based on the above Imperial data, namely CFM/ton per day, and mathematical modeling based on detail production data.

The following tables are based on heat load factors of 2.4 and 2.8.

Predicted Heat Loads at Lower Level Stations
Creighton Mine (Excluding Compressors)
For 2500 T/D, Heat Factor 2.4 (For 6200 – 7200L)

Mining Level	B.P. Inch of Hg	VRT °F	Station Intake Temp. W.B. °F	* Σ H Difference for 75°F W.B. Stope Exhaust W.B. Btu/Lb	Heat Load Btu/Min.	Volume CFM at 0.079 Lb./Ft^3	CFM Ton/Day at 0.079 Lb./Ft^3
5,000	34.8	88	54.8	13.7			
6,000	36	98	58.9	11.0	235,000	270,000	108
7,000	37.3	108	63.0	8.3	270,000	412,000	164
7,200			63.8			465,000	186
7,400			64.6			520,000	208
7,600			65.4			575,000	230
7,800			66.2			630,000	252
8,000	38.6	118	67.1	5.6	305,000	690,000	275
8,200			68.0			850,000	339
8,400			68.7				403
9,000	39.95	128	71.2	2.9	340,000	1,484,000	594
10,000	41.3						

Predicted Heat Loads at Lower Level Stations
Creighton Mine (Excluding Compressors)
For 2500 T/D, Heat Factor 2.8 (For Whole Mine)

Mining Level	B.P. Inch of Hg	VRT °F	Station Intake Temp. W.B. °F	* Σ H Difference for 75°F W.B. Stope Exhaust W.B. Btu/Lb	Heat Load Btu/Min.	Volume CFM at 0.079 Lb./Ft³	CFM Ton/Day at 0.079 Lb./Ft³
5,000	34.8	88	54.8	13.7			
6,000	36.0	98	58.9	11.0	274,000	314,000	125
7,000	37.3	108	63.0	8.3	315,000	480,000	192
7,200	37.6	110	63.8		323,000	545,000	218
7,400	37.8	112	64.6		332,000	610,000	244
7,600	38.0	114	65.4		340,000	675,000	270
7,800	38.4	116	66.2		349,000	740,000	296
8,000	38.6	118	67.1	5.6	356,000	805,000	322
8,200	38.9	120	68.0		364,000	990,000	396
8,400	39.1	122	68.7		373,000		470
9,000	39.9	128	71.2	2.9	398,000	1,735,000	694

Comparison of Calculated Lead Load At Creighton In 1979 and 1993

		1993		1979	
Production	**(Tons/Day)**	**5,100**		**5,240**	
Air Flow	**(CFM)**	**880,000**		**500,000**	
Density	**(Lb./Ft.^{3})**	**0.079**		**0.079**	
Mean Mining Depth	**(Feet)**	**6,200**		**5,800**	
Calculated Heat Loads			**Btu/Min**		**Btu/Min**
(a) Fans	Intake Orebody Exhaust Total	5,195 HP	220,000	3,412 HP	144,700
(b) Diesel Equipment		790 Imp.Gal	85,000	420 Imp.Gal	45,500
(c) Electric Load Centres		7870 KVA	31,000	3400 KVA	10,400
(d) Body Metabolism		* (Refer to 1979 Data)	5,000		6,000
(e) Cement in Sandfill		*	5,500		5,500
(f) Pumps			8,500		3,420
(g) Fresh Water		*	-12,000		-12,000
(h) Underground Compressor		3980 HP	169,000		N/A
Net Calculated Heat Load			512,000		203,500
Heat Removed by Air			800,000		467,000
Heat of Wall Rock by Difference**			288,000		203,000

* 1 Lb. of air removed 11.8 Btu/Min. of heat in 1979, which compares with an almost identical figure (11.5 Btu/Min) in 1993.

** Heat Removed by Air – Net Calculated Heat Load