ABSTRACT

In May of 1996, Twentymile Coal Co. began Longwall mining in the Foidel Creek Mine’s East Mine District. These Longwall panels are some of the longest in the world (5.48 km / 18,000 ft) with very difficult and challenging geometry. The conveyance system is 1800 mm (72 in) wide, has a 4500 mtph (5000 stph) capacity and employs multiple tripper drives. The trippers include traditional powering designs as well as unique regeneration and braking designs. Several components were developed specifically for this application including, a mobile tail brake. Control algorithms were carefully developed and tested via computer modeling. A world production record was achieved in the 3rd month of operation and over 1 million tons were mined in June 1997.

INTRODUCTION

Twentymile Coal Company (TCC) has one active underground operation, the Foidel Creek Mine. This operation is located within the Twentymile Park Basin in Routt County, approximately 39 kilometers (24 miles) southwest of Steamboat Springs, Colorado (Figure 1).

Two seams occur on the property, the Wolf Creek and the Wadge Seam. All are in the lower part of the Upper Cretaceous William’s Fork Formation of the Mesa Verde Group. The Foidel Creek Mine is producing from the Wadge Seam, which is classified as a low sulfur, high BTU group C bituminous coal. The in-place thickness of the seam ranges from 2.6 m (8.5 ft) to 3.0 m (10 ft). TCC began mining at the property in 1982 employing
continuous mining equipment with the first Longwall equipment installed in 1989. The thickness mined with the Longwall equipment may reach a minimum thickness of 2.1 m (7 ft) due to localized areas where the ash content in the bottom 1 foot of coal can be excessive.

As for the profile of the coal seam, the mine is located on the eastern flank of the Tow Creek and Oak Creek Syncline. TCC initially began Longwall mining in the West Mine District in 1989 and completed those panels in 1994. The panels in the Southwest Mine District were mined in 1994 through 1996. The past and present mine plan is shown in Figure 2. TCC’s annual production gradually increased from 2.45 million tons (2.7 million short tons) in 1986 to 5.3 million tons (5.84 million short tons) in 1996.

**Conveyor Evolution**

The conveyors in these mine districts ranged in length from 3,048 m (10,000 ft) to 3,658 m (12,000 ft). All conveyors were installed on an inclining grade from tail pulley to head pulley with the overall grade varying from 6% to 8%. The width, speed, and the design tonnage of these conveyor systems increased from a 1,200 mm (48 in) conveyor operating at 3.3 mps (650 fpm) and 2,200 mtph (2,400 stph) in the 6 Left Panel to a 1,400 mm (54 in) conveyor operating at 3.7 mps (730 fpm) and 2,900 mtph (3,200 stph) in the 3 Southwest Panel.

Due to the long panel lengths and large lift requirements, the initial 6 Left Panel conveyor utilized what was then relatively new intermediate drive technology. TCC and suppliers, Continental Conveyor & Equipment Co. (CC&E) and Dodge/Rockwell, worked jointly through an evolution from the original linear belt-on-belt design into the eventual multiple tripper design utilizing a target tension control algorithm. The profile and layout of the 3 Southwest Conveyor using this tripper technology is shown in Figure 3. TCC and CC&E had led the way in the development and application of these tripper drive control techniques that were the foundation for the East Mine District conveyor designs.

**EAST MINE DISTRICT**

When TCC began Longwall mining in the Southwest Mine District in 1994, planning for the East Mine District (EMD) also began. One of the most difficult areas associated with this new district involved the design of the panel conveyance system (Figure 4). Since the panels were designed to be laid out across the syncline structure in a southwest to northeast direction, the head pulley and tail pulley location at the start of Longwall mining in each panel would be at an elevation above the lowest point on the conveyor. The difficulty of this conveyor system was further increased due to the fact that the panels were initially designed to be 6.7 km (22,000 ft), which were to be some of the longest in the world. The final installed length was eventually decreased to 5.48 km (18,000 ft).

**Conveyor Requirements**

While designing the conveyor for the first panel (9 Right), all future panel conveyors had to be evaluated to insure the design and equipment selected for first panel would be adequate throughout the EMD. During the initial design stage, TCC established the following objectives in designing the EMD conveyance systems:

- Install a system with an initial capacity of 4,500 mtph (5,000 stph) in the 9 Right Panel with the capability of increasing the capacity of the system to over 7,000 mtph (8,500 stph) by increasing belt speeds and installing additional trippers in future panels.
- Install the system as one conveyor to minimize the capital expenditures and installation costs associated with multiple conveyors such as storage units/take-ups, discharge assemblies, chute work, drives, etc. A one conveyor system would also minimize Longwall production delays that would result due to conveyor equipment removal during Longwall retreat.
- Minimize the number of trippers required or installed on the conveyors. TCC’s history with tripper technology had shown that trippers reduce conveyor

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**Figure 3 - 3 Southwest Panel**

**Figure 4 - East Mine District Panel Profile**
system availability by approximately 0.2% to 0.5% per tripper on a panel belt installation. Therefore, while tripplers were necessary to insure adequate power for the conveyor and keep belt tensions low, one of the major focus was to minimize tripplers.

9 RIGHT PRE-DESIGN DECISIONS

After an initial look at the application, it became apparent the 9 Right conveyor design was going to be unique and several difficulties were identified. More than ever, it was important than the design be sound from the beginning of the project to insure its success. Therefore, two years prior to equipment purchase, TCC and CC&E began working on the design. A program was established to study the existing conveyance system to verify existing design methodology, equations and friction factors, as well as existing equipment performance and control algorithms. This was accomplished by collecting extensive data on the 3 Southwest Panel conveyor (3 tripplers) and the South Mains conveyor (regenerative). It was important to verify the past before designing for the future. Specifically, six items were identified as major concerns that needed to be resolved before the system design could continue; (1) proper power equations, (2) belt width, (3) belt type and strength, (4) drive type, (5) brake type and (6) the analysis of the dynamic conditions of stopping and starting.

Power Equations

Because of the unique profile, it was important to define accurate friction factors (Kx and Ky as defined by CEMA) to be sure the design was not too low or too high. Once completed, a design friction was established, as well as a friction range, to provide a worst case and best case scenario for component selection.

Belt Width

Although a 1,500 mm (60 in) wide belt was adequate for the initial requirements of 4,500 tph (5,000 stph), the conveyor would have to be designed to operate at a belt speed of 4.67 mps (920 fpm) (Figure 5). This speed was quite high by historic US underground experience and any future increases in production could only be accomplished by further speed increases. It was also realized that the higher speeds would only make braking the declined portion more difficult. Therefore, TCC and CC&E decided that a 1,800 mm (72 in) belt width should be used. This size allowed the number of tripplers to be minimized (based on the same belt rating as the 1,500 mm option), allowed for a lesser belt speed than the 1,500 mm option, and provided future capacity increases at speed ranges comfortable to CC&E and TCC.

Belt Type and Strength

Since a Longwall Panel conveyor application requires the constant lengthening and shortening of the belt, belt splicing techniques are extremely important. The use of mechanical fasteners greatly reduces the time and expense of adding and deleting belt rolls. However, higher strength belts will reduce the number of tripplers drives required. Therefore the design goal became the maximization of belt strength while still being able to use mechanical fasteners. This eliminated steel cord reinforced belts. TCC had considerable experience with 4 ply 210 kn/m (1200 PIW) beltling and was aware that 260 kn/m (1500 PIW) belts were available. Because a goal of 315 kn/m (1800 PIW) was set (to minimize tripper drives), a separate project was initiated to select a belt. The result was the selection of a Scandura Mineflex 2000 with a 350 kn/m (2000 PIW) belt rating (See Mullen, Sollars, Hock, Chopra, Frank for more details).

Drive Type

TCC had a long and successful history with AC motors and Dodge CST’s (Controller Start Transmission). CST’s employee a low speed hydrosiscous cluch to control output torque. Specifically, the controllability of the CST had greatly contributed to the development of control algorithms during the evolution of the currently used tripper drive technology. Because of the unique requirements of tripper drives, only drive types which allowed fine, active torque control at or near design speeds were considered. This limited the choices to CST, DC and VF. In the end, it was decided the CST could provide the required performance. Moreover as CST’s were very well known and accepted by the mine operating and maintenance personnel, CST’s were selected.

Brake Type

When the 9 Right discussion began, TCC did not have any experience with conveyor brakes or conveyor regeneration therefore this issue attracted considerable attention. The traditional method of braking underground conveyors was dry caliber disk brakes and this technology was initially presented. However, this was not appealing to the project engineers and thus a search for something new was started. Dodge presented a solution using a modification of its CST technology called a CSB. During the mining of the Southwest Panels, the South Mains conveyor became regenerative and required brakes. Consequently, an opportunity was presented to test this new technology. The CSB proved very successful. (See Maloney, Shadow, Sollars for more details).
Dynamic Analysis

Due to the mechanical complexity of the conveyor, the stopping and starting control algorithms would also have to be more complex than anything done previously. All the past conveyors had been designed with traditional rigid body mechanics. However, it was agreed early by all team members that an analysis as a flexible system would be required.

MAJOR DESIGN CONSTRAINTS

Belt

A 350 kn/m (2000 PIW) belt strength was the highest available. Since this belt strength was higher than anything every installed before and its mechanical fastener life was unknown, it was imperative to keep the belt tensions as low as possible and to minimize transient tensions during stopping and starting.

Belly

Due to the extreme elevation changes (Figure 6), a low tension point existed (belly) 1/3 of the way between the head and tail. This minimum belt tension point would be the critical design location for the sizing of the take-up and brakes, as it was imperative spillage did not occur under any condition.

Tail Pulley

Since there was a vertical drop of 159 m (525 ft) from the tail pulley to the belly, the resulting belt pull on the tail pulley would be very large. However in this case, the tail pulley is mounted on a mobile tailpiece that is attached to the Longwall stage loader. The Longwall stage loader manufacturer specified a maximum force of 80 tons at the tail pulley during any running or stopping condition. This constraint on tail pulley belt tension became a major factor in the overall conveyor design.

Brakes

Also due to the 159 meter (525 ft) drop from the tail pulley to the belly, it was possible for the conveyor to be regenerative in some load conditions. This would require braking in order to stop safely. Since the profile also included a vertical lift of 185 m (610 ft) in the remaining 1,899 m (6,230 ft) to the head pulley, the brakes needed to be adequate to stop the decline section as quickly as the incline section would stop due to gravity to prevent material spillage in the belly. Traditional conveyor engineering dictates the best location to place this braking is at the tail. However as stated above, the tail pulley was limited to 80 tons which greatly reduced the braking torque possibility.

CONCEPTUAL DESIGN

Considering the above constraints, it was determined the only way to successfully design 9 Right as a single flight was to utilize tripper drive technology in a unique manner. It would require a main drive, traditional powering tripper drive on the inclined section, as well as regenerative and/or braking trippers on the decline section. Although using a tripper as a regenerative drive was new, the concept was thoroughly studied and it was concluded the existing control logic should work. However, a more difficult question to be answered was whether a conveyor could be controlled with remote drives powering and regenerating at the same time. This concept had never (to the authors’ knowledge) been attempted before and had to be studied and modeled in depth.

Running

The static running tension diagram of the fully loaded conveyor is shown in Figure 7. Point “A” represents the location of the main drive which is dual 970 kw (1300 hp). Point “B” represents the location of the powering tripper (Tripper 1) which also is dual 970 kw (1300 hp). Point “C” represents the location of the regenerative tripper (Tripper 3) which is 375 kw (500 hp). You can see from the diagram, Tripper 3 is holding the belt back to maintain tension above the sag limit (lower dotted line) in the belly. The upper dotted line represents the rated belt tension of 350 kn/m (2000 PIW).

Stopping
The static stopping tension diagram of the fully loaded conveyor is shown in Figure 8. Point “A” is the main drive. Large flywheels have been added to the drive inertia to keep the uphill loaded section moving as long as possible. Point “B” is Tripper 1. Large flywheels have also been added to the drive inertia here as well. Point “C” is the braking tripper (Tripper 2). Brakes have been placed on both pulleys with a maximum torque of 70,500 n-m (52,000 ft-lbs) per pulley. Point “D” is (the regen/braking) Tripper 3. The same brakes have been placed on both pulleys. Point “E” is tail pulley. A maximum braking torque of 108,000 n-m (80,000 ft-lbs) has been applied. It should be noted this rigid body approximation was used to develop general stopping requirements. A dynamic stopping analysis using finite element methods was utilized later in the design process.

Starting

Again, a rigid body approximation was used to evaluate acceleration conditions at this time (Figure 9). Since the CST’s are capable of long start times and each drive/brake/regen station would have complete controllability with multiple feedback options, it was determined that conveyor starting would be the lesser of the concerns.

DETAIL COMPONENT DESIGN

Main Drive (Figure 10)

The main drive of the conveyor involved a four pulley drive placed on the return side of the conveyor approximately 91m (300 ft) from the head or discharge pulley. This design insured that both the primary and secondary drive pulleys contacted the clean cover of the belt. The design involved two power units that incorporated Dodge 1120K CSTs, flywheels and Reliance 970 kw (1300 hp) motors. Due to the size of these power units, only one unit could be installed in a cross-cut. Therefore, offset cross-cuts had to be mined when the continuous miner sections developed this area (Figure 10). The pulleys used were manufactured by Dodge and had an 2057 mm (81 in) face width. The high tension bend pulley and secondary drive pulley both had an effective diameter of 1270 mm (50 in). The primary drive pulley had an effective diameter of 1219 mm (48 in) and the low tension bend was 890 mm (35 in). The weight of the large pulleys was approximately 11,800 Kg (26,000 lbs), including the 318 mm (12.5 in) bearings. The two 1120K CSTs employed at this drive had a gear reduction ratio of 28.62 to 1. The difference in pulley diameters provided the slip differential between the primary and secondary drives (approximately 4%). The flywheels employed at the drives were installed on the high speed shaft on the CSTs. These flywheels weighed 2,500 pounds each and were supplied by Dodge as part of the design. Dodge performed a complete finite element analysis on these flywheels to evaluate the clamping system employed. The 1300 hp Reliance motors employed the G9500 frame and had a full load speed of 1,796 rpm. The motor, combined with the flywheels and 1120K CST provided a WK2 of 10,584 n-M (25,600 lb-Ft) at the low speed shaft of the CST. As mentioned earlier, this large inertia capacity was critical to prevent low tension situations in the “belly” of the conveyor during the stopping condition when the conveyor was fully loaded. The total combined weight of an 1120K CST, flywheel, 970 kw (1300 hp) motor and welded steel power base exceeded 27,216 Kg (60,000 lbs).
After the 9 Right Panel, the design of the main drive was changed so that the drive was on the carrying side of the conveyor as a tripper design versus the return side of the conveyor. As with the original main drive, the large power units required offset cross-cuts. This installation reduced the belt tension at the discharge pulley from the maximum conveyor tension point to a tension approximately equal to the storage unit/take-up. This resulted in a 890 mm (35 in) diameter head/discharge pulley versus the 1270 mm (50 in) diameter pulley originally installed in 9 Right.

**Tripper 1- Powering (Figure 11)**

This tripper employed two of the same 1120K CSTs, flywheels, 970 kw (1,300 hp) motors and power unit bases as the main drive. However, both the primary and secondary drive pulleys had an effective diameter of 1270 mm (50 in) to provide the approximate 4% slip as compared to the primary pulley at the main drive. Since the conveyor is immediately adjacent to the Longwall Panel, the ability to offset the drives on opposite sides of the belt was impossible. Also, due to MSHA roof control/ventilation requirements, a minimum pillar size was required. Therefore, the primary and secondary drive skids had to be separated by a minimum of 21 m (70 ft). This required special intermediate structure for the three laps of conveyor that exist between the two drive pulleys.

**Tripper 2- Braking Tripper**

The brake station employed two through-shaft 630K Controlled Stop Brakes (CBS) coupled to Eaton Lebow torque sensors which in turn were coupled to the brake pulleys. The Eaton Lebow torque sensors were rated for 135,600 n-m (100,000 ft-lbs) of torque and were employed as the feedback loop for the brake control system. The brake pulleys employed at this station had an effective diameter of 1257 mm (49.5 in). This pulley diameter was selected based on the braking force required; therefore, the primary and secondary drive skids had to be separated by a minimum of 21 m (70 ft). This required special intermediate structure for the three laps of conveyor that exist between the two drive pulleys.

**Tail Brake (Figure 12)**

The tail brake, which was the most critical component to the overall design of the conveyor, had an effective pulley diameter of 915 mm (36 in). This pulley was designed by CC&E and included Fairfield W50 planetary gearboxes installed inside the pulley. These essentially performed as the end discs of the pulley. The reduction ratio of these gearboxes was 24:1. Force Control #20 Positorq high speed brakes (pressure release/spring applied) were employed at this pulley and provided a brake force of 2,825 n-m (2,083 ft-lbs) at the high speed shaft of the Fairfield gearboxes with a total 135,600 n-m (100,000 ft-lbs) on the pulley. These brakes were mounted to the same pedestals that supported the pulley and were connected to the gearboxes by a splined high speed shaft. The pulley was lagged with ceramic lagging to increase the belt/pulley friction factor. Since the pulley was lagged, heat dissipation was a concern. Therefore, the drum was filled with anti-freeze to insure adequate heat removal from the gearboxes. The Positorq high speed brakes included equipment to provide continuous cooling oil flow, accumulators (to supply cooling oil in the event of a power outage stop situation), and a control pressure system to regulate the brake force applied during braking. The equipment required for this included a pump skid (including two cooling pumps, a control pump, and a variable displacement pump to charge the accumulator), a tank supply skid, and two accumulator
skids. All of this equipment was mounted on the rigid portion of the monorail and was retreated as the Longwall retreats.

Storage Unit (Figure 13)

The hydraulic storage unit/take-up for this conveyor was designed to hold a maximum of 365 m (1,200 ft) of belt to provide adequate capacity for the 244 meter (800 ft) belt rolls. This storage unit was designed to provide a maximum belt tension of 245 kn (55,000 lbs). The unit involved a conventional four lap configuration and was designed to provide a mobile carriage speed of 0.3 mps (1.0 FPS). The unit included 890 mm (35 in) pulleys to meet the belt manufacturer's requirements. Because of the high tension requirements, a sheave carriage was required to allow the winch rope to be doubled. As part of the design of the system, CC&E incorporated load cells in the storage units to measure the belt tension. Belt tension was used as a feedback to the Allen/Bradley control system which in turn controlled the pressure of the hydraulic circuit.

Belt Handling (Figure 14)

During the design of the conveyor, it was apparent that one of the major problems with the continuously retreating conveyor would be efficiently handling the 1800 mm (72 in) belt as it was installed and removed from the system. This was further compounded by the fact that TCC wanted to maximize the belt roll size in order to minimize the frequency that belt had to be removed. The final roll size selected was 244 m (800 ft). This roll size resulted in a roll diameter of 2.87 m (113 in) and a weight of approximately 12,700 Kg (28,000 lbs). In order to handle this large structure, TCC and CC&E developed unique designs for these idlers. The top idlers included removable side brackets which provided easy removal of not only the side brackets but also the wing rolls. Further, CC&E provided a design that replaced the normal roller shell with an 8 gauge shell and driscoll pipe sleeve. The removable side bracket and wing roll reduced the weight of the frame and center roll to less than 63 Kg (140 lbs). The removable side bracket design not only allowed the weight of the idler to be reduced due to disassembly, it also provided a more compact design for transport into and out of the mine.

As for the return idlers, TCC and CC&E developed a V return frame design that was hinged on one end and employed the conventional underground “J” bolt fastening system. This design allowed the rolls to be removed and thus reduce the overall weight to be reduced to less that 54 Kg (120 lbs). The hinged arrangement and “J” bolt fastening system significantly reduced the installation and removal time requirements. The 8 gauge shell and driscoll sleeve also reduced these rollers by approximately 11 Kg (25 lbs) each.

Scissorveyor

When TCC elected to use 1800 mm (72 in) structure in the EMD, there were major concerns associated with the
Delays that would occur from removing this large structure during a production shift due to Longwall retreat. This concern stemmed not only from the weight and size of the structure but also the belt tension that existed at the tail pulley when the Longwall was on the declining portion of the profile. Therefore, TCC and CC&E developed a conveyor structure system that allowed the system to be retracted automatically during production without causing production delays. During the idle period, the roof hung structure could then be removed and the scissorveyor extended prior to the next Longwall shift. The design of this system originally began in the 3 Southwest Panel and was completed in the 9 Right Panel.

TCC and CC&E also improved the system during the 9 Right Panel by installing a belt lifting/separation device on the outby end of the system to facilitate removal of the roof hung structure. TCC also incorporated a hydraulic system in this lifting device that included a hydraulic winch to mechanize the extension of the system.

DYNAMIC ANALYSIS AND CONTROL ALGORITHMS

Since each major component was unique and had a unique function, the control algorithm for each drive/brake station was also unique. As components were developed, a separate team worked on developing and refining the control algorithms for running, stopping and starting.

A dynamic analysis software program was previously in development as a joint project between Overland Conveyor Co., CC&E, Dr. Tom Rogge, and Dr. Tom Rudolphi of Iowa State University. In January 1995, it was decided this program would be the simulation tool used to model and verify the control algorithms. The schedule on this project was therefore accelerated and the scope expanded so the new program would be capable of simulating complex multiple feedback loops which would be required.

Running

The running algorithms were going to be very similar to previous TCC panel belts. The primary pulley of the main drive is the master and sets the system belt speed by locking the clutch after acceleration. The secondary pulley of the main drive is slightly larger and the clutch slip is controlled in order to load share with the primary.

At Tripper 1, the primary drive clutch slip is controlled to maintain the target tension at the outby load cell. As the belt tension rises above the target, power is increased to bring it back to the desired target. The Tripper 1 secondary pulley clutch slip is controlled to match the primary drive amps.

The Tripper 3 drive clutch is also controlled to maintain a target tension, however here, increasing clutch pressure provides holdback torque or regeneration. As the belt tension tries to fall below the target (as loading increases on the decline) regeneration is increased to bring it back up to the desired target.

Stopping

Initially, three unique stopping conditions were considered: normal stop, emergency stop and power outage. Flowcharts were developed to show how each component (tail brake, Tripper 3, Tripper 2, Tripper 1 and Main Drive) would react during each type of stop. As the complexity of these flowcharts grew to handle all scenarios, it was decided to simplify and create only one stopping mode to handle all conditions. Because the worst case stop is almost always power outage, it was used for all further analysis.

Four main load conditions were analyzed: fully loaded, loaded only from the tail to the belly, loaded only from the belly to the head, and completely empty. However, since the conveyor length is changing constantly, the same load conditions had to be analyzed when each tripper is removed from the system. An analysis was also done with the tail at the lowest possible point in the belly. In all, 23 different scenarios were dynamically stopped in the computer to test the developed stopping algorithm.

During this analysis, particular attention was given to:

- Belt tensions at the tail pulley to ensure the mobile tail piece would not be pulled from the Longwall stage loader
- \( T_1/T_2 \) ratios at each brake pulley to ensure no loss of contact due to slippage
- Belt tension in the belly to ensure belt sag did not exceed 4% which would allow material spillage
- Backstop torque at the main drive and Tripper 1 to prevent backstop overload

One particularly difficult problem encountered during the algorithm development was how the controller would determine the necessary braking torque at each brake location based on the material load on the conveyor at stopping time. The solution involved a unique routine developed by TCC engineers to keep track of material load along the entire length the belt at all times. The load was derived from the stage loader crusher amps and constantly updated during operation.

In the following paragraphs, a description of the final stopping control algorithm is given along with an example of the dynamic output. This example is for a full length conveyor with a full load (4500 mtph) along the full length.

Tail

When the stop signal is given, the tail brake is engaged to its desired torque in a linear ramp over 6 seconds. The desired torque is based on the material load on the conveyor at that time. At 100% design load, the desired is 100% of the full tail braking torque capability. At zero load, the desired torque is 50% of tail braking torque. Between full load and empty, a linear percentage between 100% and 50% of full torque is calculated and applied.

Since this component was a prototype, it was not exactly known how fast the brake could be engaged in service nor how quickly it could reach full torque. Since it was imperative the tail brake lead the tripper brakes to prevent tripper slippage, this was an important design parameter.
For modeling purposes, it was assumed the brake would not start engaging for 0.5 seconds which was determined to be a worst-case scenario. Considering that, it was determined analytically that the minimum ramp time to full torque should be 6 seconds.

During initial operation, it was verified the brake could be set to drag slightly during running so when braking was required, it would begin producing torque immediately. It was also verified in field testing that the physical distance and hydraulic characteristics would not allow actuation to full torque in less than the required 6 seconds.

Figure 15 shows the tail brake torque ramp, the resulting $T_1$ and $T_2$ belt tensions and the resulting $T_1/T_2$ ratio for a full load stop.

**Tripper 3**

When the stop signal is given, the Tripper 3 brakes are engaged to their desired torque in a linear ramp over 10 seconds. The desired torque is based on the material load on the conveyor at that time. At 100% load, the desired torque is 100% of the full torque capability. When empty, the desired torque is zero. Between full load and empty, a linear percentage between 100% and zero is calculated and applied.

The same discussion as Tripper 3 applies.

Figure 16 shows the Tripper 3 torque ramp, the resulting $T_1$, $T_2$ and $T_3$ (between the two pulleys) belt tensions and the resulting $T_1/T_2$ ratio for a full load stop.

**Tripper 1**

Although there are no brakes on Tripper 1, it still comes into play during stopping. Since the incline portion of the power outage is assumed, the loss of the motor torque is evident until the brake can engage and take hold.

With the addition of the Lebow torque sensors, the algorithm was modified to slightly to drag both brakes during running so brake torque was available immediately when required.

Figure 16 shows the Tripper 3 torque ramp, the resulting $T_1$, $T_2$ and $T_3$ (between the two pulleys) belt tensions and the resulting $T_1/T_2$ ratio for a full load stop.

**Tripper 2**

When the stop signal is given, the Tripper 2 brakes are also engaged to their desired torque in a linear ramp over 10 seconds. The desired torque is based on the material load on the conveyor at that time. At 100% load, the desired torque is 100% of the full torque capability. When empty, the desired torque is zero. Between full load and empty, a linear percentage between 100% and zero is calculated and applied.

Figure 17 shows the Tripper 2 torque ramp, the resulting $T_1$, $T_2$ and $T_3$ (between the two pulleys) belt tensions and the resulting $T_1/T_2$ ratio for a full load stop.
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The conveyor (when fully loaded) will naturally come to a stop quickly due to gravity, the inby brakes must stop the decline sections just as quickly or belt will accumulate in the belly and spillage will occur. With the normal inertia of the 970 kw (1300 hp) motors, the incline section was calculated to come to a stop in about 6 seconds. As stated earlier, the tail brake took 6 seconds to reach full torque, therefore bringing the decline sections to a stop in 6 seconds was not possible. Consequently, inertia in the form of flywheels was added to each 970 kw (1300 hp) drive to increase the stopping time of the incline section.

However, when the incline section of the belt was empty, this large drive inertia would keep the incline belt running longer than desired and create high belt tensions in the concave curve in the belly which could pull the belt up into the roof. Thus, a control decision was added.

When the stop signal is given, the Tripper 1 CST clutches are either held closed to maintain the inertia of the motors and flywheels or released based on the amount of material load between Tripper 1 and the belly. If the primary pulley amps are greater than 40% of full load amps, the primary pulley clutch is held closed. If the primary pulley amps are greater than 70% of full load, the secondary pulley clutch is also held closed. If the primary pulley amps are below 40%, both clutches are released.

Figure 18 shows the Tripper 1 torque, the resulting T1, T2 and T3 (between the two pulleys) belt tensions, and the resulting T1/T2 ratio for a full load stop. The torque curve shows the gradual decrease in torque due to the flywheel inertia and the backstop torque on the primary pulley when the belt stops.

Main Drive

Flywheels are also used on the main drive and are used in the same manner as Tripper 1. The resulting output for the main drive looks very similar to the Tripper 1 results.

Belly Tension

As one of the chief concerns, the belt tension in the belly was the reason for adding flywheels and braking the tail end as quickly as possible. Figure 19 shows that tension for this same fully loaded stop.
Take-up Carriage Displacement and Velocity

Figure 20 shows the resulting take-up movement during this full load stop. This is a four lap belt storage unit, therefore the 3.4 m (11.15 ft) of carriage displacement represents 13.6 m (44.6 ft) of belt length differential from running to stopped.

**SUMMARY**

As of December 1, 1997, the Longwall has completed both 9 Right and 8 Right and is being moved to 7 Right. Many production records have been set and the conveyor project can be deemed successful.

Several companies and many individuals have spent considerable time and effort in order to accomplish this task. Cyprus Amax placed significant emphasis on teamwork between suppliers and their own engineering, operations and maintenance personnel with emphasis on system design and reliability. It was stressed early on that the mine wanted a machine that worked with a very high degree of reliability and availability.

However, in many cases during the design, an established component or known engineering technique was not available to fill a need. In some cases, the demands of this underground mining operation did not allow the use of the most desirable engineering options. As such, unique components and techniques were developed. This combination of new technology, prototype equipment and the desire for high reliability at the onset of operation, created a difficult challenge for the design team.

Although most aspects of the project were accomplished successfully, a few problems still occurred and many lessons were learned. In some cases, communications between component manufacturers created compatibility problems that affected system performance. In some cases, compatibility between hardware and control software has led to less than desirable component life.

Little has been documented here about the specifics of the control software and hardware as this would justify a complete paper of its own. However, control software was undoubtedly the most important single part of this project. In the EMD, the control software without a doubt looks different today than it did in May 1996 as it is continuing to be modified and upgraded. Several conveyor changes have occurred and will continue to occur in the EMD as mine plans change. The ability to modify the control algorithms have allowed most of these changes to occur with little concern. However, as conveyors such as these continue to grow more complex, the ramifications of making control changes can be realized on seemingly unrelated components, therefore a definite procedure for evaluating these changes should be developed and followed.

**REFERENCES**