A State of the Art Belt Conveyor
Review of installation, commissioning and start-up

Walter Küng
MAN TAKRAF

SUMMARY

The Henderson Coarse Ore Conveying System is a three-flight belt conveyor system transporting primary crushed molybdenum ore 24 km underneath the North American Continental Divide. One of the conveyors is 16.825 km in length with 8,200 kW of installed power. The overland conveyor contains 11 vertical curves and 9 horizontal curves, some with radii as small as 1500 m. The system began full operation in December 1999 and is one of the longest and most complicated belt conveyor systems yet constructed. The design of this conveyor system utilized many innovative engineering concepts including low energy belt covers, low resistance idlers, torque controlled tail drives, combined gravity and active winch take-up systems, small radius horizontal curves and belt turnovers. This paper reviews the start-up of the system, discusses the success of the design and addresses the lessons that can be learned from the five years of operation of this unique conveyor system.

1 INTRODUCTION

The Henderson 2000 Project replaced an existing 24 km rail haulage system that had been in operation since 1976, with an underground crusher and underground/surface conveyor system consisting of 3 flights to cover the 24 km. The project commenced basic engineering in September 1996 and began full production in December 1999. The project expended 1.35 million man-hours, finished US$14 million under budget and 3.5 months ahead of schedule. The site is located in Colorado USA, approximately 150 km west of Denver, at an elevation of 3000 m. Figure 1 shows the overall layout of the project

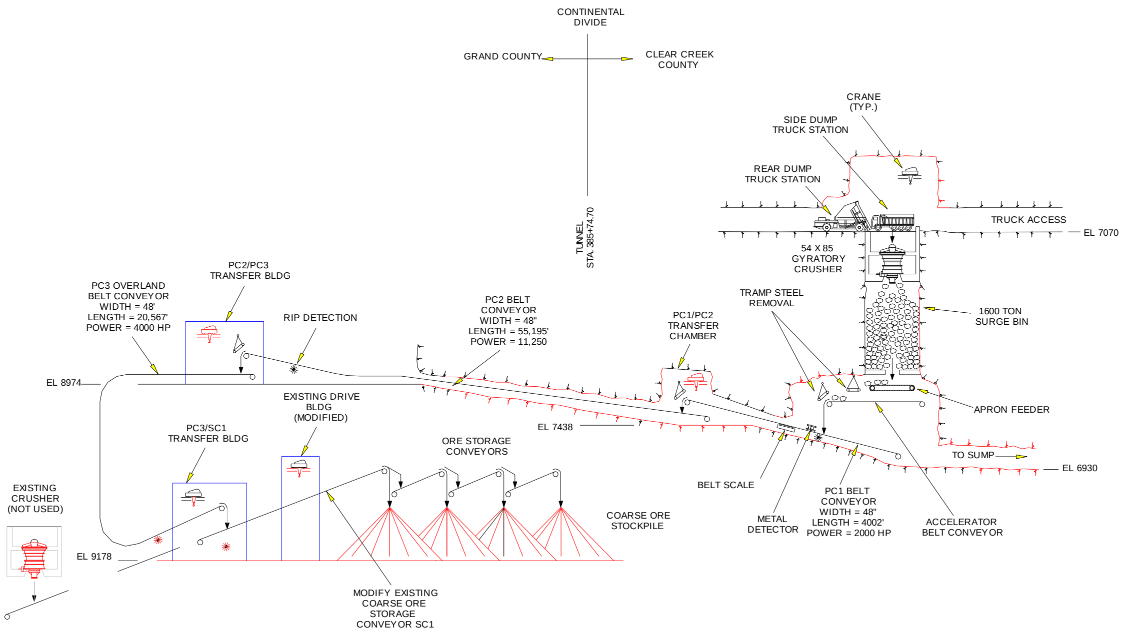

Figure 1
The Henderson 2000 Project

The focus of this paper is on the three conveyors in the system. The first is known as conveyor PC1 (Production Conveyor No. 1). Figure 2 shows the general layout of this conveyor.

Conveyor PC1:

Length:	1220 m
Lift:	160 m
Belt Width:	1.2 m
Speed:	4.5 m/s
Design Capacity:	2270 mtph
Installed Power:	1500 kW

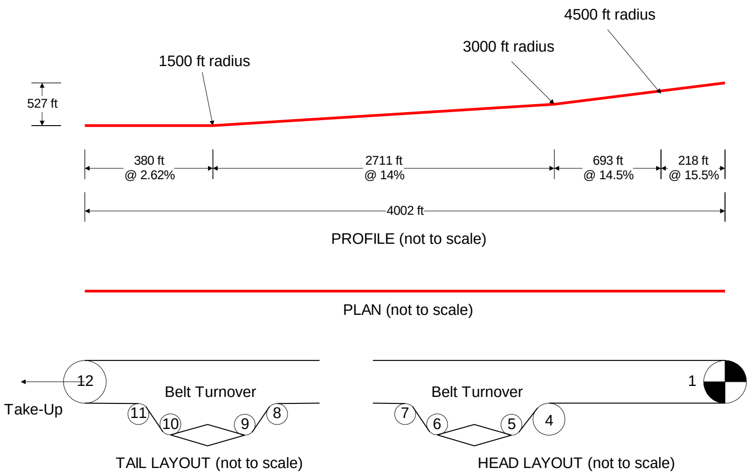

Figure 2
Conveyor PC1

Conveyor PC2 with a length of 16.825 km is one of the longest single flight conveyors yet constructed. The general information on conveyor PC2 is shown in Figure 3.

Conveyor PC2:

Length:	16,825 m
Lift:	475 m
Belt Width:	1.2 m
Speed:	0 - 6.1 m/s
Design Capacity:	2270 mtph
Installed Power:	8200 kW

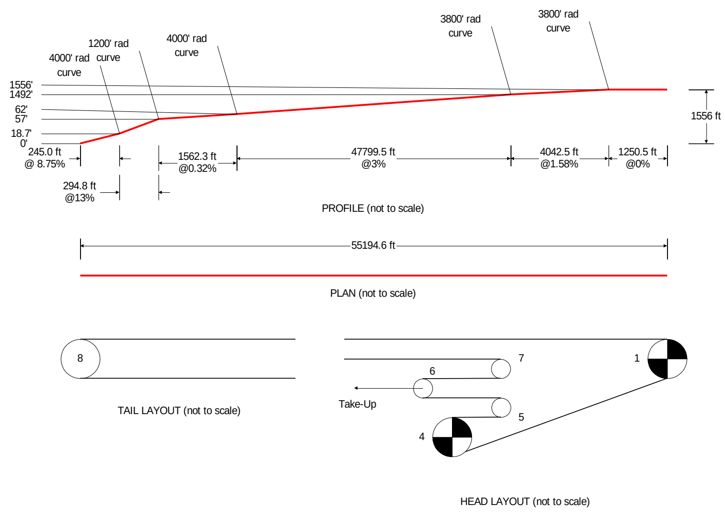

Figure 3
Conveyor PC2

Conveyor PC3 runs overland through the mountains for a distance of 6.4 km. The system contains 11 vertical curves and 9 horizontal curves, belt turnovers and a tail drive. From a design standpoint it is one of the most complicated conveyor systems yet constructed.

Conveyor PC3:	
Length:	6400 m
Lift:	76 m
Belt Width:	1.2 m
Speed:	4.5 m/s
Design Capacity:	2270 mtph
Installed Power:	3000 kW

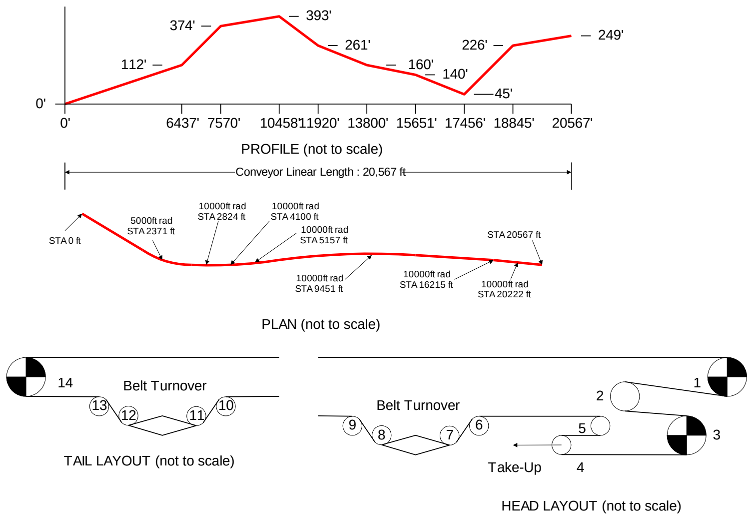

Figure 4
Conveyor PC3

2 DESIGN CONCEPTS

During the design phase of the project a paper was written which outlined some of the innovative design concepts employed in the system [1]. This section is devoted to a review of those concepts and a discussion of whether they met expectations. The intent of the review is to make an evaluation of whether these types of design innovations were of practical benefit to the final construction and operation of the system.

2.1 The Running Resistance of the PC2 Belt

The PC2 conveyor contains close to 34,000 m of ST5400 N/mm steel cable belting. The belting alone weighs over 2300 tonnes and has some 2400 km of steel cord. An important fact about this system is that 50% of the operating power is consumed turning an empty belt. Any reduction in the running resistance of the system not only reduced consumed power; it also reduced tension rating, which reduced weight, which further reduced running resistance, etc. The following design principles were implemented on the PC2 conveyor with the purpose of reducing the running resistance of the system.

- Low Resistance Idlers
- Oversized Idler Centre Roll
- Low Power Loss Belt Cover Compound

Design friction factors. One of the most important factors in the calculation of a conveyor system's operation is the "equivalent friction factor". This factor is used to estimate the force required to move the belt and, depending on the standard used, takes into account idler losses, rubber indentation and material flexure. To compare design, expected and measured values for power consumption and running resistance the basic DIN calculation will be used. The DIN standard 22101 allows for a range of equivalent friction factors between 0.01 and 0.04 with 0.02 being a "typical" value. Three important factors listed in the standard that result in friction factors at the higher end of the scale are speed above 6.0 m/s, carry idler spacing over 1.5 m and return idler spacing over 3.5 m. The PC2 conveyor has all of these factors. Despite this, experience with long conveyor systems led the design team to select the following values in the design stage for the PC2 conveyor

Equivalent DIN friction factor for design	0.013
Equivalent DIN friction factor expected for operation	0.011

Clarifying the use of these two numbers; the design number was used for selection of equipment, in particular the belting, drives and pulleys. The expected value represents what the design team realistically thought we could achieve with successful implementation of the design concepts mentioned above.

Measured power consumption. After commissioning was complete the mine operated at approximately 1800 mtph for a number of months. This production rate will be used for evaluation of the conveyor operating resistance. Figure 5 shows the measured results for operating conveyor PC2 at 1800 mtph with a speed of 4.6 m/s (the system is capable of operating at any speed between 0 to 6.1 m/s).

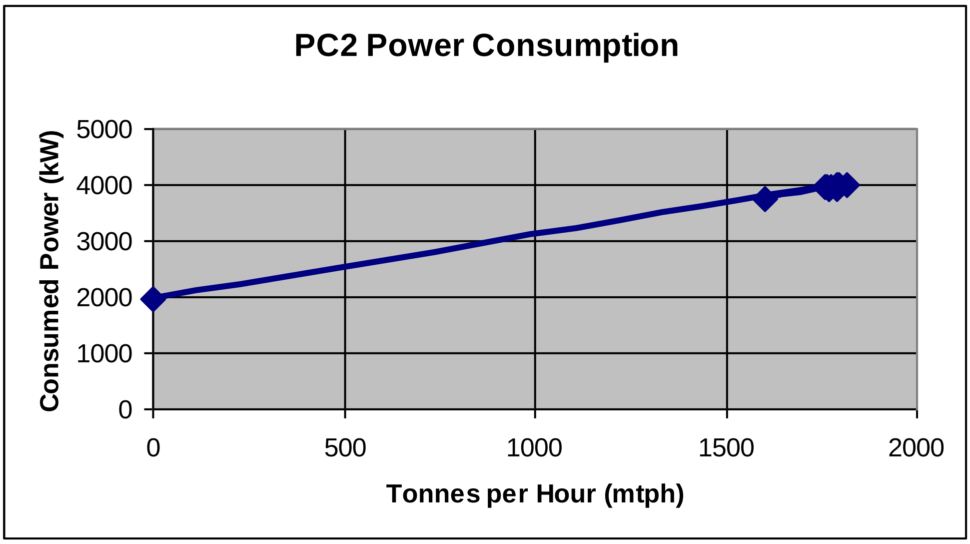

Figure 5
Measured Power Consumption for Conveyor PC2

Average power consumption based on this tonnage over 6 operating shifts is as follows:

Belt Speed	4.6 m/s (75%)
Average Tonnage	1783 mtph
Average Power Consumption	3962 kW
Equivalent DIN friction factor (based on above)	0.0075

This represents a running resistance for the conveyor 16% below the realistically expected value of 0.011 and 23% below the design value of 0.013. The next stage of the evaluation is to determine the contribution of each the design concepts to this reduction.

Idler resistance. The idlers used for the Henderson conveyors were tested at The University of Hanover for bearing and seal resistance, indentation resistance and the effect of load, temperature and various rubber compounds. The idlers used for the system are low resistance with oversize bearings and non-contacting seals. The laboratory tests indicated the reduction in running resistance for the idlers over a more conventional configuration is approximately 2 N per idler roll. There are 9,600 carry sets and 4,800 return sets in the PC2 system, this equates to approximately 58 kN reduction in operating tension

Large diameter centre roll. Testing conducted by MAN TAKRAF indicated that a reduction in running resistance of approximately 0.75 N per idler set was possible by using an 219 mm diameter centre roll rather than a 178 mm diameter roll. This equates to an overall savings of 7 kN in running resistance. Figure 6 shows the effect on indentation resistance of a large diameter centre roll.

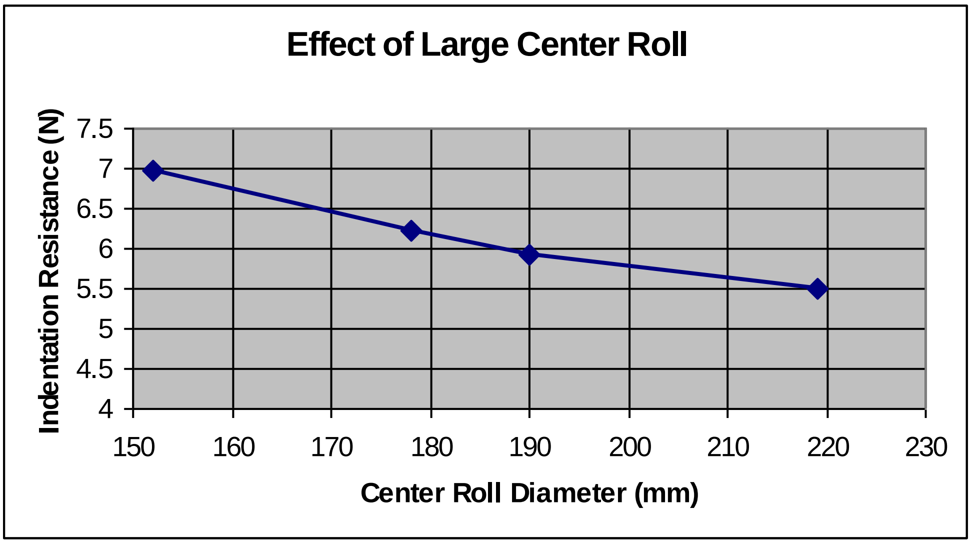

Figure 6
The Effect of a Large Diameter Centre Roll

Low energy loss compound. Testing by the belting supplier Contitech indicated a reduction in indentation resistance of approximately 11 N per carry idler set was possible by using the special rubber compound for the pulley cover of the belt. Multiplying this value by the number of idlers on the carry side of the system adds up to a saving of 106 kN. The return side is not included in this calculation as the low vertical loading of the empty belt would results in minimal savings in indentation resistance and because the "low energy compound" was only used on the bottom cover of the belt.

Summary. Based on test results the expected reduction in running resistance through the implementation of the PC2 design was 171 kN. The breakdown for each item is shown in Table 1. The measured power, shown in Figure 5, equates to a reduction in running resistance of 169 kN from the running resistance based on the expected friction factor of 0.011.

Design Concept Implemented	Theoretical Reduction in Running Resistance
Low Friction Idler Bearings and Seals	58 kN
Large Diameter Centre Roll	7 kN
Low Energy Loss Belt Cover	106 kN
TOTAL	**171 kN**

Table 1
Theoretical Reduction in Running Resistance for Conveyor PC2

The predicted savings in running resistance were very close to the measured results. This indicates that the laboratory test data provided by the manufacturers was accurate and the promised savings were realized. A reduction in the running resistance of the system of 169 kN equates to 770 kW of power savings. Over a year of operation this is over US$100,000 savings in the cost of electricity alone. It should be noted that the only item, which carried a cost premium for implementation, was the low energy rubber compound.

The alignment accuracy that was implemented using preassembly jigs for the conveyor modules and idler sets and laser equipment once the conveyor tables were installed in the tunnel, had some positive effects. The idler frames were installed perpendicular to the stringers within 1 mm, alignment between conveyor modules once these were installed, was

1mm in the vertical plane and 1 mm in the horizontal plane. It is believed that this meticulous alignment procedure and observation of strict installation tolerances not only contributed to a near perfect tracking of the belt, both in empty and loaded conditions, but also contributed to the reduction in belt running resistance and power saving. This aspect has not been fully evaluated and quantified with regards to its contribution to power saving, but it is believed to be of significance.

2.2 The PC2 Gravity/Winch Take-Up System

The PC2 Take-Up System is a combination of active winch and gravity take-up. The take-up carriage has 50 m of movement. The system is reefed using 1-inch rope such that for each meter of counterweight movement the carriage moves 4 m. The active winch has two 75 kW drives and is controlled by limit switches on the counterweight. The system also has a large hydraulic cylinder designed to lock the counterweight and prevent the take-up from feeding belt into the conveyor system during an emergency stop.

Several options were evaluated for the PC2 take-up system. The main problem during the design phase was the dynamics associated with a system stop with loss of power, i.e. the drives do not ramp down in a controlled manner. The low-tension wave generated by the take-up during this stop resulted in very low tensions at the tail end. The purpose of the hydraulic cylinder is to lock the counterweight in position if power to the conveyor is cut thereby eliminating the low-tension wave from the system. The hydraulic cylinder is 1.2 m in diameter and contains approximately 800 liters of fluid.

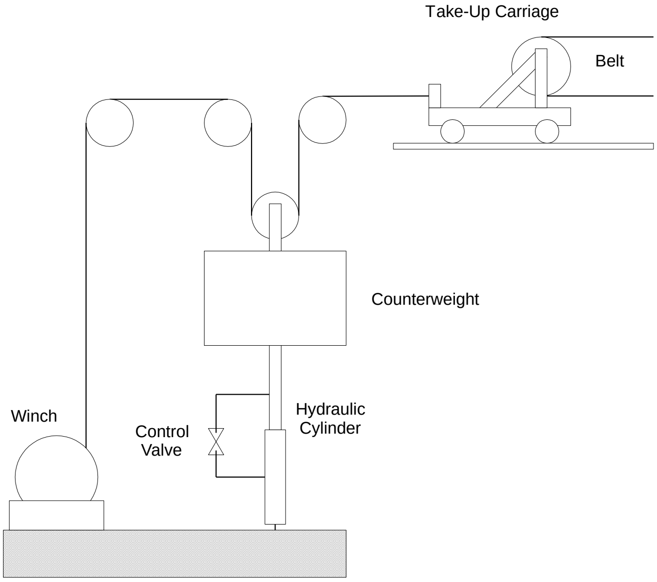

Figure 7
Conveyor PC2 Take-Up System

This take-up system, and its interaction with the drives and brakes on the conveyor, was one of the major problems encountered during the start-up phase of the project. Several failures of the connection between the drive and the drum were experienced on the winch system. Each failure resulted in significant downtime and produced reactions in the belt and take-up that were, at times, rather exciting. The following factors were identified as contributing to the problems with the winch and take-up system.

- The winch drive had little margin in the design, during operation the motor was drawing 110% of rated current.
- There was too much mass in the counter weight box. Despite best efforts during installation it was determined that the counterweight was over design requirements by approximately 10%.

- The connection between the winch drive and drum was poorly designed for this application.
- The winch controls were rudimentary and did not provide good co-ordination between brakes, motor torque and overload protection.
- The winch and gravity take-up could not keep up with the belt being fed into the take-up system by the drives during start-up.

The actions taken to correct these problems were as follows:

- The counterweight was reduced by 19 tonnes to 106 tonnes. A load cell was fitted to one of the counterweight sheaves to measure rope tension, thereby allowing measurement of the counterweight mass.
- The connection between the winch drive and drum was re-designed and modified on-site.
- The starting curve for PC2 was changed to include a 60 second plateau at 10% speed. This was very effective in allowing the winch and gravity take-up to keep pace with the belt stretch (see Figure 8).

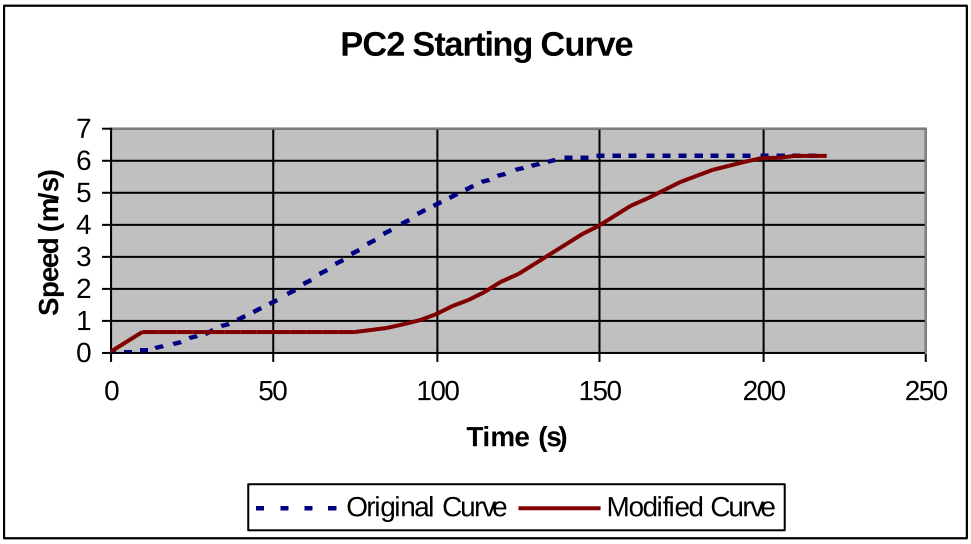

Figure 8
PC2 Starting Curve

Lessons learnt. A method of verifying the mass of the counterweight should have been included in the system. When one is dealing with about 120 tonnes of mass, it is a challenge to obtain the equipment to weigh this on-site. The cost of the load cell was small in comparison to the consequences of incorrect counterweight mass. More attention was needed to the design and selection of the winch and winch controls for the take-up. A lot of effort was put into belt, idler, drive, and pulley selection. It is easy to let the little things slip by.

On the positive side, although the starting ramp needed to be changed, the variable frequency drives installed on the conveyor allowed this to be accomplished easily.

It should be noted here that the overland conveyor, PC3, has an almost identical take-up system to the one on PC2. However, this system has had very few problems and nothing like those experienced on conveyor PC2. One of the main reasons for this is the lower belt tension on the PC3 conveyor resulting in a system, which was more conventional in size when compared to what equipment designers and suppliers had built before.

Other PC2 items – Brake/Drive coordination. The PC2 conveyor system was designed with disk brakes fitted to each of the 4 drive units, no holdback device was installed. Although it was not expected that the conveyor would roll backwards the original design called for using the brakes to hold the conveyor if this did occur. The advantage of a brake system is that some slip is allowed which will relieve very high tensions that may result from an aborted start and ensure load sharing between the 4 brakes. Holdback devices do not allow slip and therefore ensuring load sharing between multiple devices is possible but difficult. The success of reducing the running resistance of the system resulted in a conveyor that did run backwards with a loaded belt. While the brakes were able to deal with this, the co-ordination between the brakes and drives during starting and stopping, and the subsequent effect on the take-up system, proved to be a real challenge.

The first number of loaded starts went something like this. The PLC sent a start signal to the drives and released the brakes. The drives would apply very little torque at this early stage in the start curve and consequently the belt would start to run backwards. The drives, sensing reverse motion of the belt, would shut down. The take-up, which had already started winding in to try and keep up with the expected belt stretch now found itself with the system pulling belt out of the take-up. Although the winch shut down when the counter-weight hit the high-high limit switch the momentum of the system pulled the counterweight up against the mechanical stops at the top of the tower. We now had high tensions locked into the take-up system, which had to relieve, and the counterweight returned to a normal position before the PLC would allow a re-start. These actions at the head end also resulted in low tensions at the tail.

The main problem was that the control system for the brakes was not sophisticated enough to provide control for different scenarios. During loaded starts the brakes had to hold the belt until the drives reached the appropriate torque to accelerate the belt. During stopping the brakes had to bring PC2 to a stop before PC3 stopped in order to ensure we didn't bury the transfer point. The main solution to this was to install a low speed holdback. The holdback device was installed on a low-tension pulley, one holdback on each side and each one rated to hold a fully loaded belt. By using a low-tension bend pulley if very high tension did get locked into the system on an aborted start would allow some slip at the pulley rather than causing high stress in the drives or holdbacks.

The installation of the holdbacks, combined with the new starting curve, eliminated most of the problems. It allowed the focus of the brake operation to be power off stops of the conveyor. The only remaining crucial item was the requirement that the application of the brakes be delayed at least 3 seconds after the power was cut to the drives to prevent the dynamics of the system generating low tensions at the tail. This was predicted by the dynamic modeling but also proven on several occasions during start-up when the delay in the brakes was not set correctly.

2.3 PC3 Design Features: Horizontal Curves, Tail Drive, Belt turnovers

Horizontal curves and belt turnover. The performance of the horizontal curves and belt turnovers was very good. From the first start-up of the belt the tracking in both of these areas was within expected and calculated limits. The banking on the idler frames used on the horizontal curves varied between 2 degrees and 12 degrees. The wing rolls had a forward tilt of 1 degree, which assisted in steering the belt in the horizontal curves. The idler frames are fixed; on purpose there is no field adjustment made available. Banking and tilting the first outgoing pulley control the belt tracking in the turnovers. The calculation for the amount of bank and tilt required equalizes the edge tensions of the belt as it passes through the turnover system. The calculations in this area proved to be accurate.

Tail drive. The PC3 tail drive is required to reduce belt tensions on the carry side to assist in belt tracking in the first horizontal curve. The control is simple constant torque. The original setting called for the drive to add torque equivalent to 100 kN of belt tension. This proved to be slightly too much and the tracking in the first horizontal curve was too low on the idlers. The tail drive torque was reduced to 80 kN of belt tension to improve the tracking characteristics. The co-ordination of the drives during starting consists of a time delay between the start of the head drives and the application of torque at the tail. This is very simple control that proved to be very effective.

3 PROJECT ISSUES

3.1 Partnering – Shared Risk and Shared Rewards

The contractual arrangements for the Henderson Project were based around a shared risk and shared reward concept. Profit for the major partners was performance based. The result was a situation where failure of any of the major partners to achieve project goals severely impacted the profits of the others. While this arrangement carries a degree of risk for the project partners it was very successful in this case. A crucial ingredient of the success was the relationship between the partners during the project. All partners regardless of the scope of work involved undertook major project decisions collectively. This open communication enabled everyone to contribute to the decisions made and promoted a high level of co-operation between the parties involved.

3.2 Minimizing the Duration of the Mine Shutdown

Schedule is always a major project issue but never more so than when it involves closing down an existing operation to complete the work. Careful planning and incorporating this requirement in the basic design of the system resulted in a reduction in schedule of 30% from the original plan. The major items which contributed to this reduction in schedule were:

- Most of the system was complete and commissioned before the shutdown commenced. In particular, the route for the overland conveyor PC3 was selected to allow construction without shutting down the existing train system. This required some additional cost for earthworks and civil construction but proved very beneficial in reducing the duration of the shutdown.
- The design of the structure for conveyor PC2 incorporated the fundamental requirement of fast installation. The existing rail was used as the foundation for the modules. A clamping mechanism was employed to attach the feet of the modules to the rail. Special carriages were constructed to transport the modules into the tunnel using the eastern rail line. This all combined to reduce the required installation time.
- The entire length of conveyor PC2, 34,000m of belting with 69 splices, was pre-spliced and stacked near the tunnel portal in three piles. Each pile was approximately 2m high so care was required with the foundations and stabilizing mechanisms to prevent the piles from collapsing. Once the conveyor tables were installed, the installation of all of this belting was accomplished in less than a week.

4. CONCLUSIONS

The Henderson Coarse Ore Conveying System was a unique opportunity to push the boundaries of conveyor system design and implement some of the technological advances made in this area. Ten to fifteen years ago construction of this system may not have been possible. Advances in splice design, rubber compounds, component manufacture, drive control and conveyor design tools all combined to produce a successful project. In general, the expectations for the system in terms of performance and operating cost have been met or exceeded.

5. ACKNOWLEDGEMENTS

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6. REFERENCES

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7. AUTHOR

Walter Küng holds a diploma in mechanical engineering. He has been employed at MAN TAKRAF since 2001 and holds the position of Vice President Projects & Development. He has previously worked on large material handling projects for major engineering companies in Australia, Indonesia, South Africa, Canada, the US and Chile.

Walter Küng
VP Projects & Development
MAN TAKRAF Chile S.A.
200 Alcantara Piso 6
Las Condes – Santiago
Chile

Tel. + 56-2-369 5606
Fax + 56-2-369 5653
Cell +1-303-898 5952
E-mail: wkung@takraf.com