ZISCO INSTALLS WORLD'S LONGEST TROUGHED BELT 15.6 KM HORIZONTALLY CURVED OVERLAND CONVEYOR

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1.0   SUMMARY

A 15.6 km horizontally curved overland belt conveyor was commissioned in December 1996 for Zimbabwe Iron & Steel Co. (ZISCO). It is the world's longest single flite troughed belt conveyor. This paper describes the philosophy and strategy of this unique and successful design. Key issues in the design are:

1.   Belt Strength, Width, Speed and Belt Safety Factor Selection
2.   Power Analysis - Selecting an Efficient Rolling Resistant Compound
3.   Drive Size, Placement, and Control Stability
4.   Take-up Design, Placement and Control
5.   Idler Selection, Spacing, Power Influence, Belt Protection, and Vibration Control
6.   Shock Wave Dynamics and Control Analysis Methods
7.   Horizontal Curve Control & Belt Turnovers

2.0   INTRODUCTION

Troughed conveyor belt technology is extended with the successful commissioning of the Zimbabwe Iron & Steel Co. (ZISCO) 15.6 km overland in 1996.

This paper describes the mechanical and control logic design of the world's longest single flite troughed belt conveyor. ZISCO selected Bateman Materials Handling (BMH - South Africa) and Conveyor Dynamics, Inc. (CDI - USA) to design the horizontally curved overland. BMH and CDI were awarded the contract, in 1992, against stiff international competition.

The conveyor transports secondary crushed iron ore (-31.5 mm) from ZISCO's new Ripple Creek Mine stockpile to their modernized steel making plant near the town of Redcliff, Zimbabwe. Unique design and component features yield improvements in capital and operating cost, together with high reliability and low noise emissions.

Figures 1 and 2 show the beginning trek of the overland traversing the first 2.5 km negotiating complex vertical and horizontal curvatures.

Figure 1 - Looking Toward Direction of Flow
at 2300m chainage

Figure 2 - Looking at Take-up Tower
(Opposite Figure 1 Vantage)

3.0   DESIGN METHODOLOGY

Selection of equipment type, sizes, location and performance criteria challenged our understanding. The conveyor is complex due to its world record breaking length, associated control sensitivity, tracking requirements of the horizontally curved section, and demand for low noise emissions. Routing the conveyor near many small farms raised the issue of designing for noise control. Figure 3 illustrates the conveyor elevation profile.
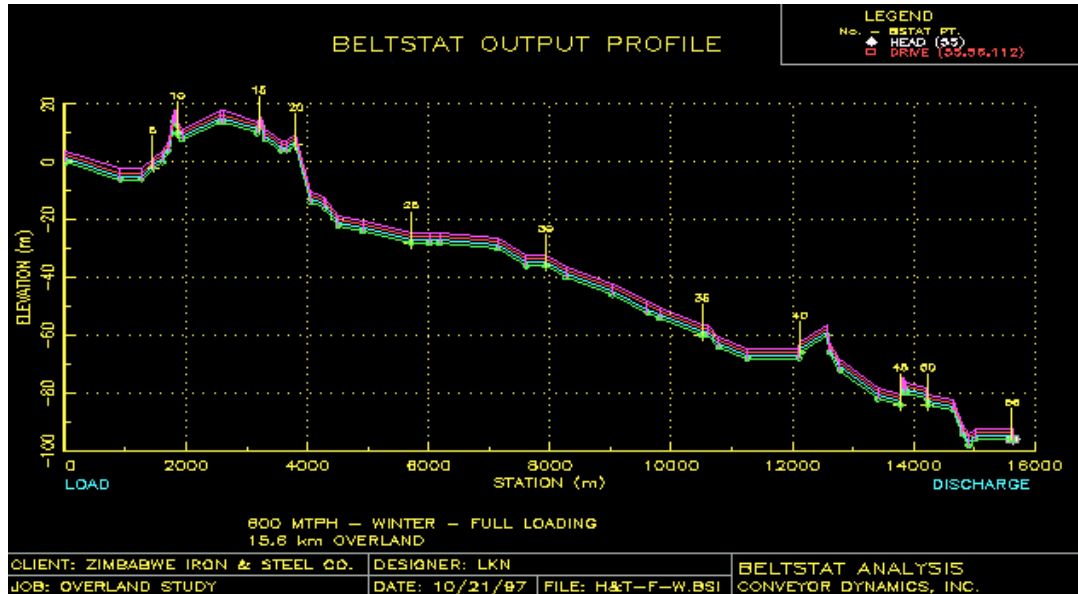

Figure 3 - 15.6 km Overland Elevation Profile

Highly complex, state-of-the-art design and efficient, competitive system cost seem to be an oxymoron. ZISCO did not subscribe to the conventions of published standards and norms. Instead their design criteria allowed the application of modern technology to enhance the design and economics of the overland system. The project's success and cost efficient design are a testament to ZISCO vision. Figure 4 illustrates major equipment locations.
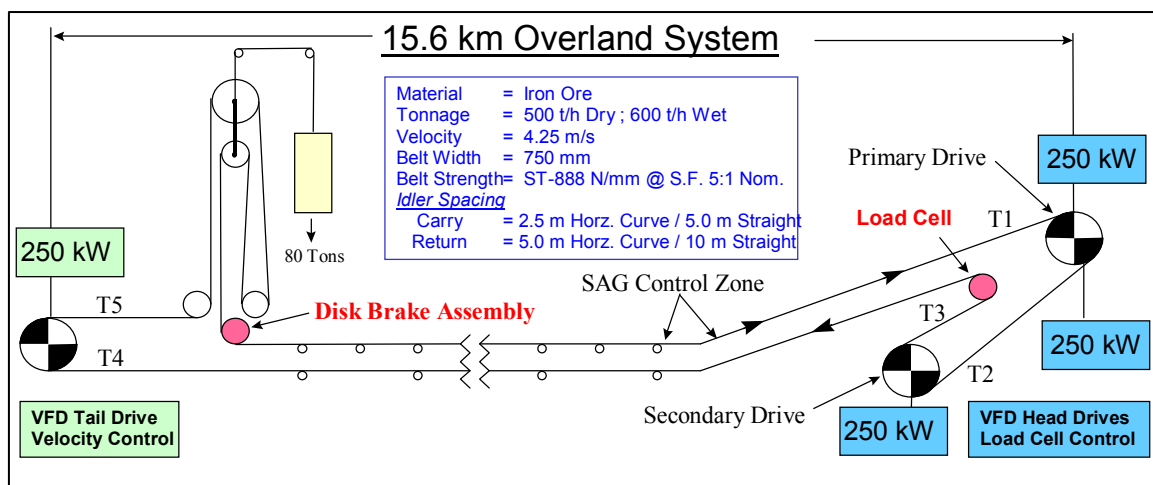

Figure 4 - Arrangement of Major Equipment and Size Specifications

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3.1   Belt Strength, Width, Speed, Safety Factor Selection, and Life

Bridgestone steel cord belt was specified with a strength rating of ST-888 N/mm. The value was governed by the belt safety factor criteria, calculated power consumption, take-up location, and drive motor distribution. Steel cord was selected over a fabric design to control belt stretch, take-up tower size, and belt power consumption. Fabric belt would stretch 7-10 times as much as steel cord, requiring over a 300 m take-up displacement, and require about twice as much power and belt strength to overcome the increased belt flexure and rolling resistance.

Belt width criteria were based on capacity (500 t/h dry), troughability, cross-sectional area, and design speed. The 750 mm elected width provides 44% CEMA (Conveyor Equipment Manufacturers Association, USA) cross-sectional loading and 152 mm edge distance. Additional cross-sectional capacity was specified to accommodate 30% volumetric fluctuating surges from the mine stockpile rake reclaimer, while maintaining a 125 mm edge clearance.

Belt speed was nominally set at 4.25 m/s with the possibility of adjustment up or down with the variable frequency drives to minimize belt edge flap excitation of the return strand and its noise consequence. Belt edge flap vibration was analyzed using a CDI special method.

The belt safety factor is SF = 5.8:1. This value will fluctuate between winter and summer, and wet and dry seasons. At zero degrees C, the safety factor falls to SF = 5.5:1. Adding stockpile water saturation, the 500 t/h dry tonnage could increase to 600 t/h. In such a case, the belt safety factor would be reduced to about 4.8:1. Figure 5 illustrates actual site conditions before hood covers were installed, where the water laden "ore stream" becomes a play on words.

Belt life is estimated to exceed twenty years. Belt covers were set at 5 mm x 5 mm to provide more than fifteen years of life, and to allow for possible future rotation (top for bottom cover), and life extension, if the top cover wear exceeds 2 mm. Rotation would be by the Mobius strip concept, requiring one temporary splice and one final splice after the two hour rotation is complete. The 5 x 5 mm cover stock sacrifices power over the alternative 6 x 4 mm covers. The added potential life increase, with rotation, was considered to be of greater benefit. This was a gamble at the bidding stage, because it required greater power, belt strength, and cost over the 6 x 4 mm covers.

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3.2   Power Analysis - Efficient Rolling Resistance Advantages

The greatest single design advantage was derived from the CDI power calculation method and resulting solution of an efficient Bridgestone low rolling resistance compound.

Applying conventional wisdom such as DIN 22101, ISO 5048 or CEMA would require a rolling friction factor of DIN f = 0.017 - 0.020. Furthermore, these factors could not provide rolling resistance values for the proposed wide idler spacing of 2.5 m and 5.0 m carry strand and 5.0 m and 10 m return strand. ZISCO allowed Bateman / CDI to apply modern cover rubber viscoelastic equations to resolve the belt tensions and power consumption. The following carry strand CEMA Ky rolling factors were used in the design:

Idler Pitch	Idler Pressure Compensation	Ky Friction Factor
2.5 m	Horizontal Curve, minimum	0.0066
2.5 m	Horizontal Curve, nominal	0.0080
2.5 m	Horizontal and Convex Curves	0.0141
2.5 m	Horizontal and Concave Curves	0.0066
5.0 m	Minimum at Concave Curve	0.0072
5.0 m	Maximum at Convex Curve	0.0128
5.0 m	Nominal, no Curves	0.0105

These values are 65% of the best CEMA estimates, and for design purposes they have +15% margin of error added to account for manufacturing variances in cover thickness and in viscoelastic properties. The overall DIN (22101) f rolling resistance design factor is f = 0.0135.

The resulting power estimate versus field measured, after extracting the 15% design margin of error factor, are within 5%: 445 kW (calculated) versus 440-484 kW (measured).

The power analysis methods are published in Bulk Solids Handling [1][2][3].

3.3   Drive Size, Placement and Control

Figure 4 illustrates the drive locations. The conveyor is driven by four (4) 250 kW Siemens variable frequency drive and motor assemblies. Variable frequency drives (VFD) were selected to provide accurate 500 second "S" style acceleration ramp, good load sharing regulation, and accurate speed adjustment to fine tune the conveyor to the best operating speed, minimizing belt edge flap. Figure 6 illustrates the "S" acceleration curve.
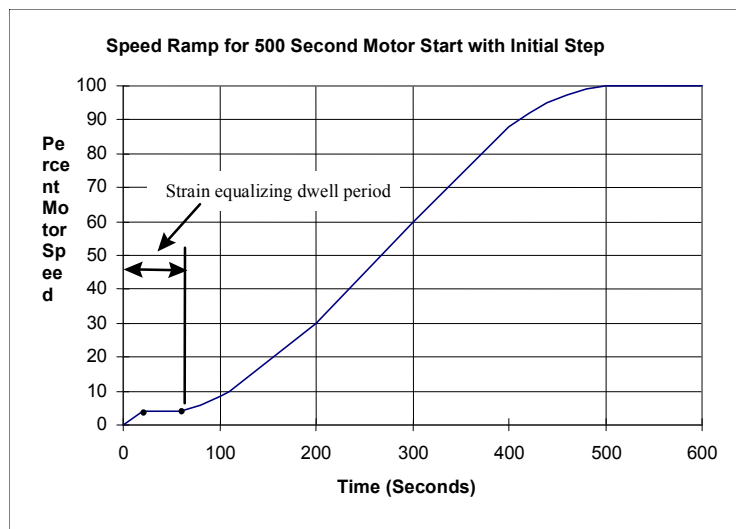

Figure 6 - "S" Acceleration Curve

The motors were originally specified at 200 kW. Siemens provided 250 kW drives at no cost increase due to their standard frame size ratings. The 200 kW rating was selected above nominal estimated power draw (132 kW) to account for the long acceleration time and peak torque rate, buildup of moisture in the iron ore material, variations in engineering values of idlers and belt, and possible deviations in belt installation conditions.

The single tail drive regulates starting, controlling initial conveyor acceleration by linearly ramping up to 4% speed at 0.2% per second rate. This takes 20 seconds. At 4% speed, tail drive speed is held constant, for 60 seconds, until all belt strain is near its steady-state tension. During shutdown of a loaded belt, portions of the carry strand, near the head end, will have low tensions and increased belt sag. The selected starting ramp minimizes control instability and drive slip caused by the head end low shutdown tensions. After 60 seconds, the "S" ramp is applied. Dynamic analysis demonstrated better horizontal curve displacement control is achieved with this tail drive acceleration control. Head drives are independently regulated by a load cell feedback loop. The load cell is located on the slack tension side (T3) of the head end secondary drive pulley. When the tail drive applies torque, return strand belt tension increases the force on the load cell. As the load increases above a specified setpoint, a P.I.D. loop proportions the torque increase of the secondary drive. The two head drives are slaves to the secondary drive. This system is equivalent to a 16 km "booster drive." See Section 3.6 for further details.

3.4   Take-up Assembly

ZISCO's 50 m high take-up tower is one of the highest building structures in south central Zimbabwe, and is fitted with aviation beacons. It is illustrated in Figure 7.
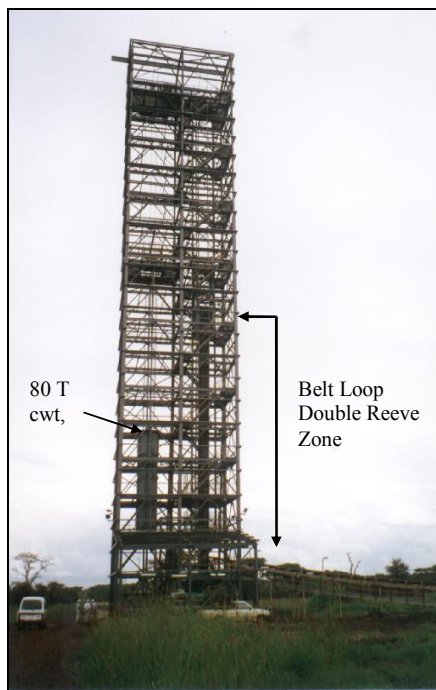

Figure 7 - Take-up Tower

The gravity take-up system is located at the tail loading region after the tail drive, as shown in Figure 4. Its location and preference for a gravity acting system were chosen to maximize control and fluctuations of the horizontal curve belt lateral displacement. Gravity and load cell winch controlled methods were extensively evaluated. Gravity regulation provides superior control of belt dynamic response. Its location is based on its proximity to the horizontal curve. A head station take-up would produce significantly greater fluctuations in belt tensions in the curve region. The tower is novel, accommodating both belt loop and counterweight.

Belt travel, accounting for all tension conditions, thermal growth, long term creep, and maintenance criteria, is calculated to be about 60 m. With a double loop reeving the total loop travel is 30 m. The counterweight travel requirement is about 44 m, equating to 22 m with the double belt loop (no creep or maintenance travel is included). Minimizing travel hysteresis and alignment of the belt loop carriage, governed the selection of a vertical tower over a horizontal arrangement

A winch positions the counterweight for the proper operating location and for maintenance.

The counterweight can be lowered to ground support when belt, drive, or take-up maintenance is required.

Radio telemetry is beamed and received from atop the tower to the head station for all communications.

Tower cladding was added later.

3.5   Idler Selection & Design Features

Novel idler design features include: a) extended carryside spacing to 5 m; return spacing to 10m, b) double pipe support of idler rolls for belt protection, c) elimination of stringers to minimize megaphoning of noise, d) longitudinal sleeper concept, e) U-frame idler support stands with integrated hood cover support, f) increased carry roll diameter for reduced power draw and noise control, and g) idler vertical alignment for belt tracking by stringer foundation plate bolt arrangement. Many of these features can be observed in Figures 8, 9 and 10.

To the author's knowledge, only two other installations have similar large spacing:

	Carry (m)	Return (m)
ZISCO (1996)	5.0	10.0
Phosboucraa (1972)	4.0	8.0
Selby (1983)	6.0	6.0

Acceptance of the recommended spacing was only possible by demonstrating proper, first principle, analytic procedures in the calculation of belt tensions and power based on the idler's spacing, diameter, and the rubber's viscoelastic influences.

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Belt protection on both carry and return strands is provided by supporting the idler and its brackets as shown in Figure 9. If an idler were to break or dislodge, the belt will ride on the round pipe support thereby providing protection against metal tearing. MELCO (South Africa) supplied the idlers and their pipe supports. Great care was taken to control fabrication of the idlers and their support. Accurate manufacturing was required to guarantee idler life, good belt tracking, low power consumption, low belt flap, and noise generation.

Figure 9 also shows the side guide roll lateral displacement restraint working in the horizontal curve. The restraint rolls are also positioned at the head and tail to centralize the belt.
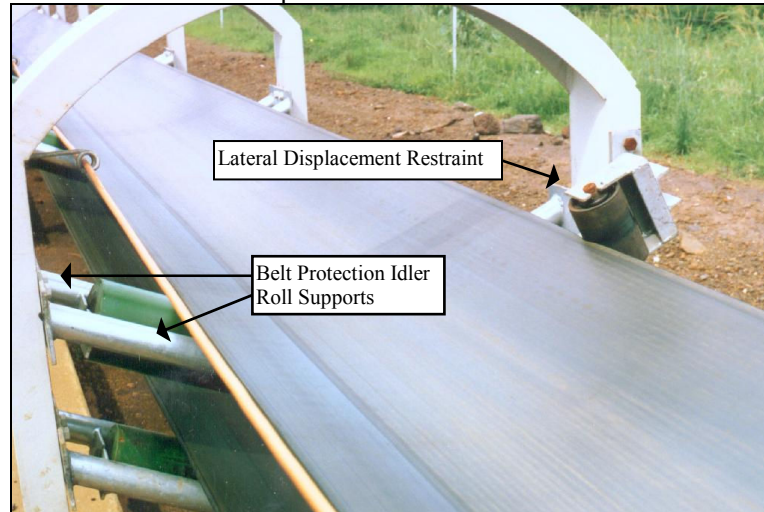

Figure 9 - Horizontal curves with side guide roll in use

Figure 10 illustrates the U-frame design of the idler supports with 5.0 m carry and 10 m return idler spacing. Note the lack of stringers, the steel support plate adjustment bolts at the foundation footing, and convex shaped U-frame integral hood support. Figure 10 shows the installation at commissioning, before the idlers were correctly elevated into contact with the belt, and the hood covers were installed. ZISCO free issued the U-frame steel structures.
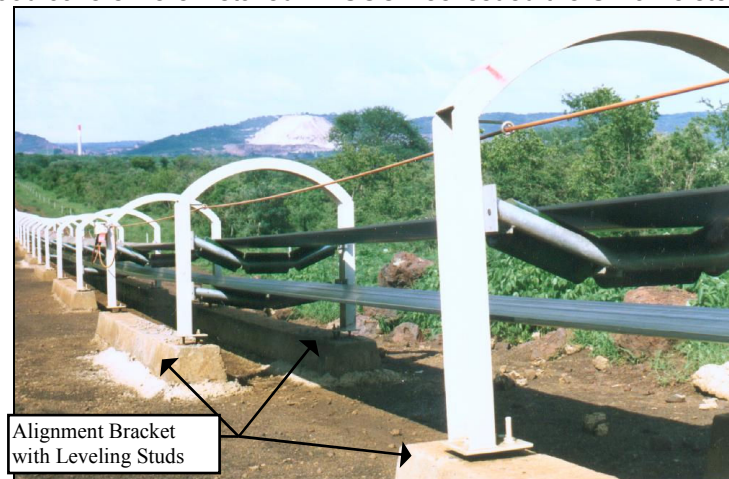

Figure 10 - U-frame Idler Supports with Large Idler Spacing

Idler diameters are 152 mm carryside and 127 mm return.

Idler bearings are 6206 series ball bearings. Idler life was engineered to exceed 100,000 hours, including allowance for alignment errors, convex and horizontal curve forces, belt flap excitation, and manufacturing tolerances. One exception was taken along the straight section with a 500 m convex curve where the life was reduced to 58,000 hours.

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3.6   Shock Wave Dynamics & Control Analysis Methods

Starting and stopping a 31 km long elastic belt requires a special understanding of belt stretch in the form of shock or elastic wave propagation and distribution.

Starting Control

Proper Starting of any long overland must include consideration of the belts tension conditions for the last stop. ZISCO's overland required special dynamics tuning to control the stopping dynamics and improve the starting dynamics. Common accelerating ramp P.I.D. algorithms can cause belt, mechanical, and structurally damaging shock wave perturbations. Fixed and load cell controlled take-up systems can likewise induce unacceptable shockwave reactions. CDI has measured a number of such shockwave reactions cases. ZISCO is one such example. Figure 11 demonstrates control instabilities induced by the drive system starting control algorithm as implemented by the VFD supplier. A number of unacceptable methods were also tried by the supplier, see Figure 12 example.
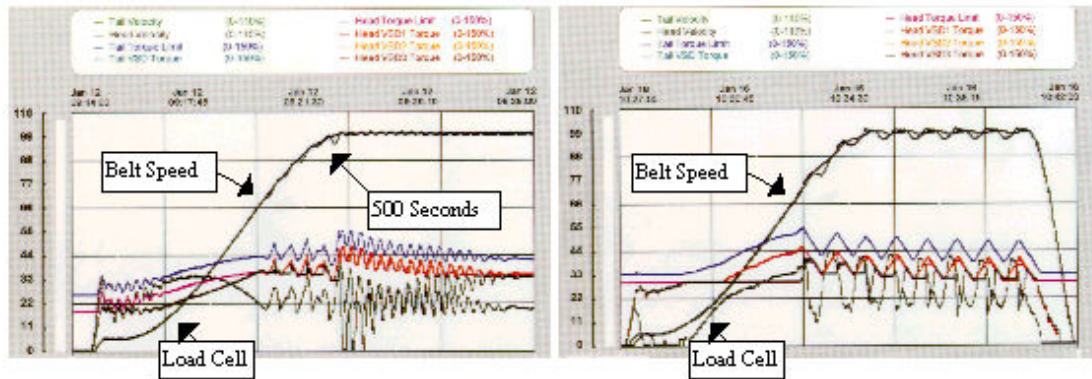

Figure 11 - Drive system control instability #1   Figure 12 - Drive system control instability #2

The instability shown in these figures is caused by the significant time delay of the elastic wave oscillation between head and tail stations. The tail drive acceleration ramp is reasonably steady, but the head drive goes unstable.

Figure 13 shows a proper control procedure implemented by CDI outside of the VFD in the PLC. Head and tail accelerations are stable and the load-cell oscillation damps quickly after the maximum strain rate is applied.
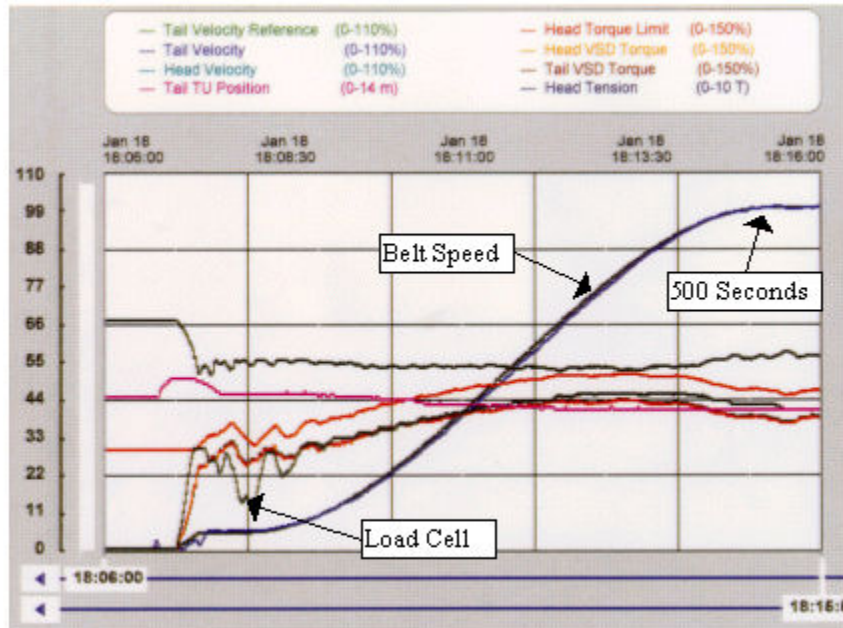

Figure 13 - Corrected started ramp control method

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Stopping Control

Normal empty and loaded belt stopping is controlled by the drives using a 100 second deceleration ramp. With all loading and temperature variations, the belt coasts to a stop in less than 90 seconds. Applying a longer deceleration ramp insures reasonable belt tensions will be distributed throughout the conveyor length.

When drive power is lost, the belt loaded carry strand inertia can exceed the return strand inertia except for the last 1 km causing a coasting distance imbalance between the upper and lower strands. Belt is pulled from the take-up, feeding the carry strand providing large belt sag conditions on the carry strand. Return belt slows down faster causing belt to be pulled from the take-up. Without "special control methods" large belt sag and ore spillage will occur in the noted regions in Figure 4 and look much worse than the conveyor shown in Figures 14 and 15.

Figure 14 - Belt Spillage in progress

Figure 15 - Sag Spillage Stockpile

Flywheels, about 9 times the motors inertia, are added to each of the head drives as part of the "special control method". In addition, a 20kN-m tail region brake is applied, also shown in Figure 4, on a 820mm diameter bend pulley after the take-up. Without the tail brake, the head zone belt sag will approach 5% (250mm @ 5.0m) Without the flywheels, belt sag is far greater. The trick is to: 1) apply enough brake to make a significant contribution to sag control while 2) balancing the stopping time necessary to offset flywheel inertia without 3) over tensioning the horizontal curve region. Fortunately, there was a good balance without unacceptable horizontal curve lateral displacement.

3.7   Horizontal Curve and Belt Turnover Design

Horizontal curve engineering is one of the more precise engineering functions in conveyor design. Centralized running of a loaded conveyor is the main criteria. Inaccurate tension analysis and horizontal and vertical curve physics will lead to improper loaded belt displacement and result in shortened idler life, damage to belt edges, or spillage. When the curve radius magnitude is constrained by property right-of-way or other obstacles, the main means of regulating belt tracking is by banking the idlers against the normal force that's pulling the belt toward the curve's radius center. The banking should not exceed 8 degrees for most angular surface shaped materials. Central running of the belt minimizes ore lateral riling motion and ultimately spillage, and prevents mal-tracking as the belt leaves the horizontal curve region. Idler banking angle on this overland are:

Carry Sides - 2° except concave curves are at 6°
Return Side - 3° except concave curves are at 5°

The return strand requires a 3-roll carry trough configuration to control the large variation in tensions.

Belt Turnover

Belt turnovers are located at the head station following the drive, and at the tail region prior to the tail drive. Its use was well justified due to the heavy wet ore muck building up on the carry side belt surface.

Bateman provided an effective design with adjustable quarter point turning rolls as shown in Figure 16. Its design was not in CDI's scope.

Figure 16 - Belt turnover shown at quarter point of rotation

4.0   CONCLUSION

The conveyors commissioning was executed without any major delays, design faults or manufacturing defects. Its performance exceeds the design criteria. The rainy weather imposed severe overload conditions, which the system survived. ZISCO has extended the envelope of technology and acceptance of longer and more cost effective troughed belt conveyors.

Acknowledgment

I cordially thank Bateman Materials Handling Ltd. for the opportunity to share in this successful venture. Their design support teamwork was very professional and efficient in executing all phases of the project

Curriculum Vitae

Lawrence Nordell is president of Conveyor Dynamics, Inc. CDI was incorporated in 1981. Although formally educated as a applied statistician he has spent almost 30 years in conveyor design. ZISCO is the longest conveyor, designed by CDI, followed by the Channar 20km two flite overland and the 24 km Indokodeco 5 flite overland (9 km longest flite). Conveyor Dynamics, Inc. has been involved in many of the world's longest and highest powered conveyors as the engineer, design auditor, or fix-it crew.

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