# THE VALUE OF FIELD MEASUREMENTS ON OVERLAND CONVEYORS

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# ABSTRACT

This paper discusses the importance of field measurement on conveyor belts for long term reliable operation of the conveyor system. Various field measurements acquired on different belt conveyor systems around the world are presented. Field measurements were acquired on these conveyors for a number of reasons including: commissioning; equipment and control system validation; system upgrades; and the investigation of equipment failures. Starting characteristics for various drive types are presented.

# 1. OVERVIEW

Before beginning, the author emphasizes that it is not the intention of this paper to either promote or criticize any specific manufacture, installation, design, or equipment. These specific case studies have been included to show the importance and usefulness of field measurements. As engineers, it is imperative that we understand precisely how the equipment we recommend, purchase, install, and operate actually performs in the field. It is also important to share this information with the industry in such a way that all can benefit from it.

The authors are in the fortunate position of being able to independently acquire data on a wide range of conveyor equipment and installations. This includes a variety of drive systems, a range of conveyor lengths and tonnages, ambient temperatures from -40°C to +40°C, as well as a wide assortment of bulk solids materials.

# 2. EQUIPMENT

Field measurements are perhaps the most valuable and interesting aspects of engineering. Regardless of what a manufacturer advertises about its equipment, or what "rumours and speculations" have been made about a system, or even how one's own theoretical models predict a system should behave, actual field measurement provide an unbiased view of how the system is "really" behaving.

The techniques and measurement equipment discussed in this paper have been previously discussed in detail by the authors<sup>1,2,3</sup>. However, on most conveyor systems the basic field measurements obtained are:

# Shaft Torque

Strain gauges are directly applied to the low-speed shaft of a drive, brake, or backstop pulley. Wireless telemetry equipment is used to transmit the data from the rotating shaft to a data recorder. This is an extremely accurate method of measuring

the forces being transmitted to the conveyor. It can also be used on almost any type of drive, brake, or backstop system. In some cases, strain gauges may also be applied to the high speed shaft of the drive or reducer (e.g. to measure internal backstop torque or reducer losses).

#### **Belt Velocity**

Velocity encoders are usually mounted at the head and tail ends of the conveyor to record the belt dynamics and transition shockwaves during starting and stopping.

#### **Take-up Position**

Displacement wheels and encoders are installed to track the take-up displacement during transition conditions.

#### **Horizontal Curve Displacements**

Belt tracking and side travel are recorded to determine the belt displacements during load on/off conditions as well as during starting and stopping.

#### **Belt Rotation**

Useful on pipe conveyor installations to measure the belt rotation and tracking at various locations during loading and unloading.

#### PLC Data

Weight scale data, motor amperage, drive and brake control signals and other pertinent information is normally acquired. This information is used in conjunction with the above measurements to obtain a more complete picture of the system operation.

# 3. STARTUP

The following case studies discuss a few issues that the authors have encountered on site. In most cases these issues had to do with starting a fully or partially loaded conveyor.

# 3.1 CONVEYOR ONE

The first conveyor installation is located inside the Arctic Circle with extreme temperatures variations from -40°C to +25°C. This particular conveyor was one of several new systems. It is over 3 km in length with approximately 200 m of elevation gain and transports 8000 t/h of copper ore. The conveyor has four VFD drives totalling 9200 kW of installed motor power.

During commissioning there were significant vibration issues during starting, which prevented the system from starting with tonnages above 3000 t/h. AC-Tek was requested to visit the site and obtain accurate torque measurements on each of the motor shafts.

Figure 1 shows the results of the initial measurements. The shaft torque on each of the drives is displayed along with the drive pulley velocity. During starting there were two main windows of significant vibration. The first occurred at a belt speed of approximately 0.1 to 0.7 m/s. The second occurred between a belt speed of 2.3 to 3.5 m/s. The second vibration was the largest and most substantial.



Figure 1. Vibration during starting – motors in load sharing control

Figure 2 zooms in on the torque of the primary drive during the second vibration. The torque varies from 0 kN-m to 540 kN-m which is almost 90% of the motor nameplate rating for that drive shaft. This figure shows that motor 2 (M2) and motor 4 (M4) are oscillating against each other. These two motors are both located on the primary drive shaft.



Figure 2. Zoom-in of primary drive torque of vibration during starting

The vibration was likely caused by a natural vibration mode of the motor and brake disk masses on the shaft.

The control of the motors was a master-slave relationship. The master VFD drive was controlled by a velocity feedback loop. The torque on the other three motors was set to load share with the master drive.

After careful analysis of the resulting measurements, and after performing several dynamic simulations, the authors recommended changing the control philosophy such that each drive was controlled individually using only speed control. It was predicted that this type of control would eliminate the vibrations occurring between motors on the same shaft. Figure 3 shows the motor torque and belt velocity after the control had been changed. Both the initial and secondary vibrations were almost entirely eliminated after the control change.



Figure 3. Motor torque and belt speed – motors in speed control

The design of the conveyor required a pre-tension step, which was not implemented in the control system. The belt should be held at 5% of full speed for a short time. This ensures the entire system is in motion before starting the main acceleration ramp. Pre-tensioning the belt provides smoother acceleration, lower belt tensions, and minimizes transient tension waves. Figure 4 shows the final start-up with the pre-tension step. The belt was fully loaded at 4750 t/h (which was the maximum available at that time). The belt started smoothly and without any problems. The system has now been in full operation for more than two years.



Figure 4. Motor torque and belt speed – start with pre-tension step

Even though the conveyor was equipped with VFD drive control, direct strain gage measurements were still crucial in accurately identifying and correcting the drive control issues. Combined with dynamic analysis and theoretical predictions, a stable starting control was implemented quickly and successfully.

# 3.2 CONVEYOR TWO

As this conveyor demonstrates, even though a conveyor appears to be operating correctly and the supplier asserts that everything is working, this may not be the case. There may still be serious problems which need to be resolved.

This conveyor transports 6500 t/h of material at just under 6.0 m/s. The belt length is approximately 3.5 km and the system has a total installed power of 4350 kW. There are two drives on the primary drive pulley and one drive on the secondary. All three drives are "actively" controlled. The authors were consulted directly by the client at the end of the commission stage to ensure the system was indeed operating properly before being handed over to them. At this time the system was supposed to be functioning normally. However, after installing the test equipment the first empty and loaded starts proved otherwise.

Figure 5 shows the empty belt start-up. The design specified the required starting time of the conveyor to be 160 seconds with an initial 20 second pre-tensioning step at 4% speed. The actual measured starting time was only 62 seconds. The initial pre-tension step was very poorly controlled and although the second part of the velocity curve was relatively smooth, the acceleration rate was fast (28 seconds).



Figure 5. Motor torque and belt speed during empty belt start-up

The fully loaded belt start-up is shown in Figure 6. Although the starting time had increased to almost 112 seconds, this was not acceptable. Furthermore, the velocity profile contained several regions of "dwell" periods with no acceleration followed by periods of rapid acceleration. These abrupt acceleration regions are not acceptable as they result in significantly reduced belt safety factors and induce dynamic shockwaves into the system.



Figure 6. Motor torque and belt speed during loaded belt start-up

After discussions with the drive supplier, the authors were assured that the start-up settings only needed to be slightly modified in order to provide a smooth acceleration curve. Various changes were made and several more fully loaded starts were measured. Although these changes slightly improved the velocity acceleration ramp, they resulted in even larger torque fluctuations during starting (Figure 7). The modifications also reduced the empty belt starting time to less than 20 seconds. This was clearly not acceptable.



Figure 7. Motor torque and belt speed during loaded belt start-up

Unfortunately this was not the only issue discovered during testing. As the conveyor was loaded, occasionally the motor load sharing would be lost between drives. Generally (but not always) Motor 1 would essentially "do its own thing" while the remaining drives continued to load share (Figure 8). At other times during empty steady state running, the drive torque on one motor would drop to almost zero, remain there for 30–60 seconds, and then resume load sharing normally.



Figure 8. Loss of -motor load sharing on primary drive during steady state running

None of these anomalies were known by either the drive supplier or client at this point (and again this was at the end of the commissioning stage).

These measurements highlight the importance of field measurement at the time of commissioning. It can be very difficult to ascertain drive torque and belt speed during starting and stopping without field measurements. It's easy to see that the belt is starting, however, it is nearly impossible to "see" or "feel" problems such as load sharing or large torque oscillations unless the proper instrumentation has been installed. If such problems exist, they can reduce the reliability of the system and cause premature failures down the road. Without having completed field measurements, the client would not have known of the existing problems in the drive system.

# 3.3 CONVEYOR THREE

This conveyor was over 12 km long, transporting 2400 t/h of coal. The authors were again requested to be present for the full load testing and verify the installation was operating within the design parameters.

After all equipment had been installed, several empty belt starts were performed to ensure the conveyor was starting properly. The system was controlled using VFD drives (two drive pulleys at the head, and one at the tail). Although the empty belt starting curve was very smooth and the required pre-tensioning step had been implemented, the start-up time was only 110 seconds (Figure 9). The design report had specified a start-up time of 350 seconds. This simple, yet critical, parameter had been checked and verified during empty belt testing, but somehow it had been changed. Fortunately this correction was easily made and the fully-loaded starting curve was nearly ideal (Figure 10).



Figure 9. 110 second start-up of 12 km overland conveyor



Figure 10. Corrected 350 second start-up and excellent load sharing

When a power failure occurred, it was discovered that the tail brake was not being immediately applied. When the authors inquired about this they were told by the PLC programmers that this brake was only a "parking brake" to be applied once the conveyor had come to rest. But this was not correct. The reason for the brake is to prevent excessively long drift times during a power failure stop when the decline section of the belt is loaded. Without reducing the drift time of the belt, the volume of the head chute is insufficient and would likely overfill. This issue, while easily corrected, shows the value in conducting an independent set of field measurements. It also demonstrates the need for periodic measurements to verify that the system is still operating as initially specified.

# 4. INCREASED CAPACITY

The next case studies discuss how measurements were used to increase the capacity of the different conveyor systems.

#### 4.1 CONVEYOR FOUR

This system consisted of multiple conveyors (with a total approximate length of 6000 m). The design capacity of the conveyors is 18000 t/h, however, the client could only achieve a maximum tonnage of 12000 t/h due to several problems. The authors were requested to perform measurements and a design audit of the conveyor to determine what actions were required to run full design tonnage on the conveyors.

Field measurements were an essential part of determining the path forward. They were to be used to determine the safe operating tonnage of the currently installed equipment, and also to establish what equipment upgrades would be necessary to meet the increased demand capacity.

The conveyors were driven by 850 kW motors. Each conveyor had two to four motors depending on its length. Fixed filled fluid couplings were used on all conveyors. The oil fill level in these types of couplings is not actively controlled. The couplings contained a delay chamber so that the oil drained from the working chamber to the delay chamber when the motor was not rotating. In the final stopped position, a certain amount of oil remains in the working chamber and the rest is in the delay chamber. During start, the oil is pushed into the working chamber of the fluid coupling by centrifugal force. The rate of transfer from the delay chamber to the working chamber depends on various factors such as oil viscosity (which is a function of temperature) and the diameter of the nozzle holes (for oil transfer from delay chamber to working chamber).

Figure 11 shows motor torque and belt velocity on one of the conveyors at an average loading of 9500 t/h (approximately 50% loaded). This starting curve is characteristic of the starts of all conveyors on this particular system when loaded.



Figure 11. Typical loaded starting curve using fixed filled couplings

When the motors first turn on, the initial torque on each motor quickly rises to approximately 85%, 95%, 125%, and 150% of the motor nameplate rating. From the graph, the primary motor on the left side has a torque output of 85% nameplate torque at 12 seconds whereas the secondary motor on the right side has a torque output of 148% nameplate torque at 14 seconds. However, a few seconds after the peak initial torque, the output torque drops to 50% to 85% nameplate torque for the motors. It then lingers at this low torque values for some time. Finally, when the belt speed increases to 60% to 70% of full speed, the torque suddenly rises again.

Figure 11 shows high torque at the beginning and end of the start that can result in belt slip on the drive pulley if it is too high. Second, due to the low torque in the middle of the start, the fluid coupling may have insufficient torque and the belt may stall without starting. This low torque is a limiting factor to the maximum capacity of the conveyors and must be understood.

A fixed fill fluid coupling has a characteristic torque starting curve which is a physical attribute of the fluid coupling. The starting torque can be increased or lowered by changing the oil fill level in the fluid coupling. It must be understood that these curves do NOT change - it is a physical attribute of the fluid coupling. Each fluid coupling will have its own lambda curve; however, all the fluid couplings of the same model will be somewhat similar.

It must be noted that these couplings were from one specific manufacturer and thus are not representative of all couplings. The authors have taken measurements on a wide range of coupling types, manufactures, and sizes, and the resulting behaviours can be significantly different from one another. Figure 12 shows the "normal" start-up procedure used at the mine when the conveyor stops and the belt is more than 50% loaded. The couplings are sprayed with a water hose in order to keep them cool while starting. This prevents the thermal fuses from blowing and having to refill the coupling oil levels. This is necessary due to the low torque that occurs when the couplings are starting, resulting in very slow acceleration. The field measurements were able to identify the allowable loading in the conveyor under which the belt could be safety started. Unfortunately the safe design capacity of the conveyors, due to the physical characteristics of fixed filled fluid couplings, is much lower than the design tonnage.



Figure 12. Water cooling of the fluid coupling during starting

Over the course of several weeks, long term trending and steady state data was acquired from the PLCs and compared to the field measurements. Figure 13 shows the results from one conveyor over a range of tonnages. Each data point represents a fixed steady state period where a constant loading of the conveyor was achieved over a specific time period. This data, combined with the loaded starting information, was essential in determining the required modifications to increase the conveyor capacities to 18000 t/h throughput.



Figure 13. Demand power vs. tonnage

# 4.2 CONVEYOR FIVE

The next conveyor is approximately 3.1 km in length, transporting copper ore at 7200 t/h. The client wanted to increase the capacity to 8800 t/h. Field measures were made to determine the current operating condition of the conveyor and what, if any, changes would be required to accomplish this update. The measured power consumption is shown in Figure 14. The conveyor has four 1305 kW variable speed motors installed. The demand power at the current belt speed and 7200 t/h condition was approximately 75% nameplate. As such, the predicted power consumption at 8800 t/h was still acceptable and just under 90% of the motor nameplate rating.

As Figure 14 shows, the starting control and motor load sharing was excellent. The dynamic behaviour of the belt was also verified under a fully loaded power failure condition. Using this information, the belt dynamics were again simulated at the higher capacity. The results showed that no dynamic problems would be expected at the higher tonnage.

However, the belt safety factor at the increased tonnage was below the acceptable level for the client. Furthermore, the cross-sectional material loading on the belt exceeded the 110% CEMA loading. It was therefore recommended that the drive pulleys be replaced with the next largest diameter. This resulted in an increased belt speed which in turn decreased the material cross-sectional loading to acceptable levels. The belt safety factors also increased to satisfactory levels, and there was still sufficient power to meet the design tonnage.







Figure 15. VFD Starting with four drives at 7000 t/h

# 5. DETECTING PROBLEMS

The next examples show a few systems which had experienced a failure of one type or another. Field measurements were used to verify the belt tensions and uncover any anomalies.

# 5.1 CONVEYOR SIX

The system consisted of three overland conveyors which were encountering multiple premature pulley and bearing failures. The client required field measurements be performed to rule out dynamic shockwaves or other anomalies that could be contributing to the excessive pulley failures. The measurements would also be used against the original pulley design tensions to ensure the belt tensions were indeed as expected.

The conveyors had extremely undulating ground profiles with multiple incline and decline sections up to +/- 120 m. The total system length was just under 9 km. DC motors were used on all conveyors. Figure 16 shows the DC starting torque and resulting velocity profile. This was well within the original design tensions.



Figure 16. Starting of fully loaded belt using DC motor control

Historical steady state data was also provided. An algorithm to search the data files for constant steady state tonnage and torque was developed. The results for one conveyor are shown in Figure 17 at specific tonnages and as a function of ambient temperature. This type of data is extremely useful for verification of theoretical models.



Figure 17. Demand power at various tonnages and a range of temperatures

After a wide range of measurements were recorded under various worst-case conditions, no significant problems were encountered. All belt tensions were within the expected design ranges. This information was provided to the pulley manufacturer to further indicate that the pulley designs needed to be re-evaluated.

Although the field measurements did not indicate any excessive belt tensions, there were several control issues which needed attention. During testing, five operational stops (e.g. the belt was decelerated using the motors) were recorded. Three of these stops were acceptable. However twice during the motor deceleration, the belt suddenly sped up before decelerating again (Figure 18). The reason for this control instability was not known and may be related to programming or faulty equipment (problem with a tachometer).

The motor torque during the operational stop was acceptable (except as noted above). However, when the motors turn off there is torque backlash on the backstop due to the sudden release of torque on the motors (Figure 19). The peak backlash torque was significantly higher than the running torque. The authors recommended that the torque be ramped down over a short time period at the end of the operational stop in order to avoid this backlash and increase the life of the backstop.



Figure 18. Control instability during operational stop



Figure 19. Torque backlash during an operational stop

#### 5.2 CONVEYOR SEVEN

This conveyor is a relatively short inclined conveyor. The conveyor has a single drive located about a third of the way back from the head pulley. The conveyor not only utilised an electronic soft start, but also had a delay filled fluid coupling installed. This was one of many conveyors on which the authors were contracted to take measurements to ensure proper functionality after commissioning.

Figure 19 shows the motor torque for two back-to- back fully loaded starts. The conveyor was purposely loaded to the maximum potential throughput of the system and an emergency stop was initiated. When the conveyor was initially restarted it "tripped out" resulting in an aborted start-up. After the trip, the conveyor was successfully restarted without a problem. During both starts the peak motor torque was only 114% of the motor nameplate rating. This is exceptionally good for a belt conveyor, and one would normally expect 140% or more from either a fluid coupling or soft start control on their own.

The combination of the fluid coupling and soft starter together seem to perform very well. Peak motor torque values for several other loaded belt starts were measured and varied from 105% to 109%.



Figure 20. Conveyor start using a combined soft start and fluid coupling

Unfortunately, the very first night after the field measurement equipment had been installed, this conveyor experienced a rather troublesome series of events. For unknown reasons the downstream conveyor was manually locked out and this signal was not being sent back to the controlling PLC system. The system was started and

began transporting material onto a stopped receiving conveyor. In another turn of events, the plugged chute detectors had been disabled and thus the conveyor head chute simply continued to fill with material. Once the head chute was completely filled, more and more material continued to be jammed into the head chute. This process persisted until a supporting structure ultimately failed and the conveyor finally tripped out.

The next morning was filled with speculation and theories of what had actually happened. Fortunately, the field measurement equipment had recorded the entire event. Figure 21 shows the approximate timeline for the failure. At 145 seconds, material begins loading on the feeding conveyor and passes over the weight scale. This material begins loading onto the incline conveyor at approximately 180 seconds and the motor torque increases. By 315 seconds, the belt is fully loaded and the transfer chute begins filling with material. At this point the motor torque has stabilised at its steady state, fully loaded running values.

At 360 seconds, the material has built up sufficiently in the chute so that material is contacting the drive pulley causing the motor power to increase. As material continues to build up in the chute, the motor torque and corresponding demand power continues to increase. Likewise, as the motor torque builds up, the belt speed decreases due to the motor slip curve and increased slip across the fluid coupling.

Finally, at 442 seconds, the structure catastrophically fails. At this point, the conveyor comes to an abrupt stop (0.85 seconds), and the internal reducer backstop engages, holding the loaded incline belt from running backwards.



Figure 21. Timeline of structural failure

Figure 22 shows a zoomed in view of the failure. Data was recorded at a rate of 100 samples per second. As such the entire failure was captured with very high resolution.



Figure 22. Zoomed-in view of motor torque and belt speed

Obviously this information was extremely beneficial when trying to determine what had happened. Furthermore, the torque measurements could easily be backcalculated to determine the exact belt tensions at all locations along the conveyor. These values were critical when re-examining the structural failure and subsequent repair.

# 6. VALIDATION OF THEORY

It is critical for the design engineer to understand accuracy and limits of theoretical equations. As the engineer gains confidence in the software and understands any limitation that may exist, he will be better equipped to optimise the design of new systems without compromising safety or reliability. This is a critical characteristic of a good design engineer. As such, the authors have always compared field measurements to design calculations.

Field measurements are used to validate the demand power. On long overland conveyors the rolling resistance of the bottom rubber over the idlers can account for 40% to 70% of the demand power of the conveyor. The rolling resistance calculation is not a straightforward theory and has many variables that affect the outcome. Therefore, it is imperative to know the limits and how accurate the theory is. An example of this is Conveyor Number Four (Section 4.1). Figure 13 shows that 100% nameplate power is reached at approximately 14000 t/h. Unfortunately, the design

tonnage was 18000 t/h. The designer of this system clearly did not know the limitation of the calculation method that was used to design these conveyors.

Dynamic analysis is of critical importance in the design of conveyors to correctly predict belt velocities and tensions during starting and stopping. However, this analysis uses non-linear equations in the spring formulation, which directly affects the belt velocities and tensions. The designer must know if the equations are correct, and what the limits of these equations are.

Field measurements are always compared to the theoretical equations. For example, Figure 22 shows shaft torque measurements compared to dynamic analysis. This measurement is made on an incline belt that was equipped with both brakes and backstops. The measurement is from an emergency stop where the brakes were applied to stop the belt. In this figure, the solid lines are the field measurements and the dashed lines are the predicted values from dynamic analysis. The figure shows both torque at the low speed shaft of the motor (blue), and belt velocity (red is head velocity, and brown is tail velocity). As can be seen the dynamic analysis sufficiently predicts the behaviour of the belt during stopping.

Figure 23 shows the steady state power consumption with a fully loaded belt on a 1.5 km overland "pipe" belt conveyor. This was the last pipe conveyor in a series of three totalling almost 5 km in length. Pipe belt conveyor design, power consumption, and dynamic behaviour require special engineering equations and technology. They are generally less understood than conventional troughed belt conveyors, and have their own set of particularities, characteristics and design challenges. The figure shows at seven seconds an emergency stop is initiated and the resulting dynamic behaviour is recorded at the head and tail ends of the conveyor. Belt rotation was also measured at various locations along the conveyors.

The information from field measurements is used to continuously improve theoretical models in order to further optimise new systems that are both safe and reliable.



Figure 23. Comparison of dynamic analysis results to field measurements



Figure 24. Pipe conveyor system - steady state running followed by an emergency stop

# 7. CONCLUSION

This paper has discussed various case studies and shown several real world examples of how field measurements have been used to improve the performance and reliability of overland conveyor systems. Motor starting characteristics from an assortment of systems were presented.

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