GEARLESS DRIVE SYSTEMS – LOW INERTIA AND SHIFTING ALIGNMENTS

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INTRODUCTION

The advent of sophisticated Variable Frequency (VFD) controls and advanced electric motor designs have allowed conveyor design engineers to eliminate the gear reducer from the drive trains of many of today's high volume, high speed bulk material handling systems. Thanks to these new controls and motors, the conveyor drive motor can now run at the natural speed determined by the belt speed and the pulley diameter, while maintaining high electrical efficiency and high motor torque.

These new drive systems appear to have been whole heartedly accepted by the bulk material handling community as an unmitigated blessing. Unfortunately they do have limitations, and the conveyor designer must be aware of these limitations. The gearless drive motor is just another tool. It is important to select the right tool for the job. If you only have a hammer, the whole world looks like a nail.

Over the last few years, CDI has been involved in a number of design projects and forensic studies involving gearless drives. In the process, some important issues and problems with the gearless drives have been discovered that need to be considered as part of the initial conveyor design criteria.

These problems can be roughly divided into two types:

- Low motor inertia
- Maintaining precise rotor alignment

OVERVIEW OF CONVEYOR DRIVE TRAINS

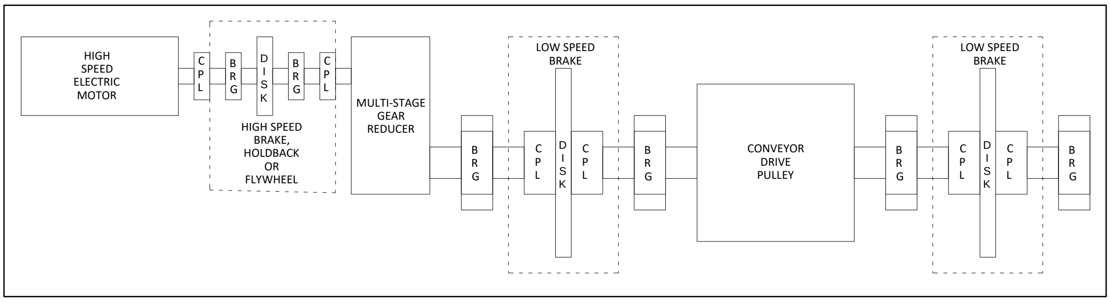


Figure 1: Conveyor Drive Train with Gear Reducer

In the past, a conveyor drive train consisted of a high speed electric motor, a multi-stage gear reducer and a drive pulley, as shown in Figure 1. The dashed lines in this figure surround what can be considered optional equipment, which may or may not be required in any particular drive train. Depending on the local electric supply and the motor design, the motor may be running as fast as 1000-1800 rpm. The pulley is typically running at a much slower speed of 50-90 rpm. Gear ratios of 15:1 to 30:1 are typical and generally require 2 to 3 stages of gearing to achieve.

Today, with the proper VFD and gearless drive motor, a much simpler drive train can designed, as shown in Figure 2. There is no longer a need for the gear reducer or the high speed coupling between the motor and the gear reducer. The entire drive train now runs at 50-90 rpm.

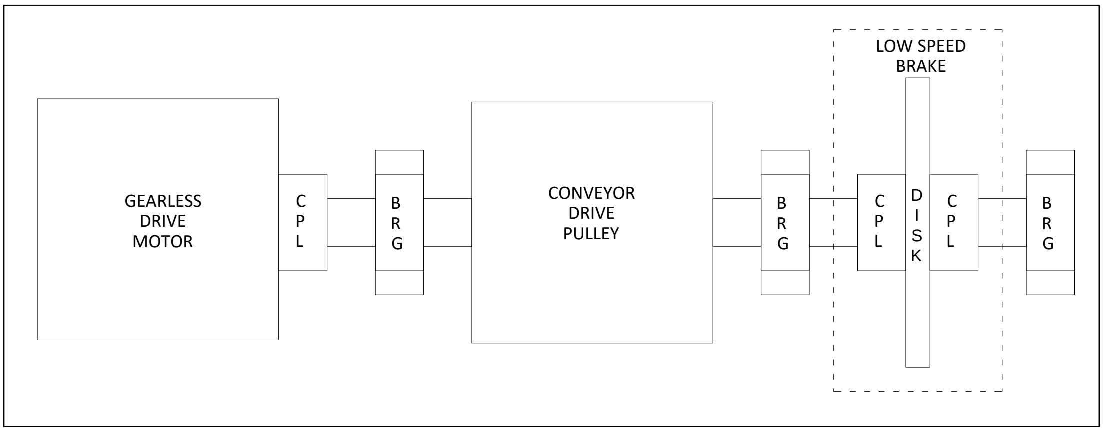


Figure 2: Conveyor Drive Train with Gearless Drive

COST BENEFITS OF THE GEARLESS DRIVE

Substantial savings in capital costs can be made by selecting the gearless drive system over the gear reducer drive system.

A recent project required replacing of a pair of multi-stage gear reducers coupled to two 726 kW high speed electric motors on the same drive pulley. The gear reducers, uninstalled, cost approximately $350,000 USD each. Other recent projects involving gearless drive motors have used multiple drive motors rated at up to 4400 kW. Gear reducers rated for this level of power are estimated to cost in excess of $2 million USD each.

There are additional savings to be had in terms of both capital and operational costs:

- No high speed couplings required.
- No gear reducer cooling system required.
- Smaller drive train footprint leads to reduced foundation costs.
- Fewer components requiring precision alignment during installation and maintenance.
- No gear lube to spill during installation and maintenance.
- No gear reducer maintenance.

DISADVANTAGES OF THE GEARLESS DRIVE SYSTEM

The use of a gearless drive system results in increased costs of some peripheral accessories required for proper dynamic control of the conveyor, and reduces the conveyor system inertia, making the conveyor more difficult to control in some circumstances.

High Speed Accessories

The lack of a gear reducer in the drive train means that brakes, backstops, and flywheels lose the mechanical advantage of the gear reducer ratio.

The gear reducer multiplies the torque ratings of the high-speed brakes and backstops by the gear ratio. For a gear ratio of 20:1, a low speed brake or backstop torque needs to be rated for 20 times as much torque.

The flywheel is an energy storage device, and the gearbox multiplies the moment of inertia of the flywheel by the square of the gear ratio. For a gear ratio of 20:1, a low speed flywheel would need to have 400 times the inertia to have the same effect on the conveyor system performance.

- Low speed brakes require larger disks, callipers and hydraulic systems.
- Low speed backstops must be considerably larger than equivalent high-speed holdbacks. Depending on the conveyor lift and load, holdbacks may not be available in the torque rating required. In such cases, fast-acting brakes may have to be used instead.
- Low speed flywheels would have to be enormously massive to have any appreciable effect.

Motor Inertia

The drive motor rotor has a moment of inertia similar to a flywheel. Table 1 shows typical motor inertias for various gearless drive motors. The rotors on these motors are quite massive, on the order of 10 to 20 metric tons, and diameters are on the order of 2 to 2.5 meters.

Table 1: Motor Inertia of Typical Gearless Drive Motors

Motor Power	High Speed Inertia	Pulley Speed	Motor Speed	Gear Ratio		Low Speed Inertia
kW	kg-m^2	rpm	rpm			kg-m^2
4400	24000	66	66	1	:1	24000
5065	24000	66	66	1	:1	24000
3500	12000	87	87	1	:1	12000

Using the 4400 kW drive as a baseline, a comparison can be made between a typical gearless drive and some possible variations using high speed motors.

Table 2 compares an equivalent single motor option and an equivalent dual motor option, at three different motor speeds, or gear ratios.

Table 2: Motor Inertia of Comparable High Speed Motors

Motor Power	High Speed Inertia	Pulley Speed	Motor Speed	Gear Ratio		Low Speed Inertia	Compared to 4400 KW Drive	
kW	$kg\text{-}m^2$	rpm	rpm			$kg\text{-}m^2$		
4400	922	66	1200	18	:1	304793	12.7	:1
4400	673	66	1500	23	:1	347624	14.5	:1
4400	507	66	1800	27	:1	377107	15.7	:1
2x2200	652	66	1200	18	:1	215537	9.0	:1
2x2200	476	66	1500	23	:1	245868	10.2	:1
2x2200	358	66	1800	27	:1	266281	11.1	:1

The use of a gearless drive system involves a significant reduction in the motor inertia. This reduced motor inertia reduces the natural drift stop time for the conveyor.

Drive Train Alignment

A comparison of Figure 1 and Figure 2 shows that the gearless drive system has far fewer components to align, which greatly eases the time and cost of installation and maintenance of these systems. However, the alignment requirements for the gearless drive system are much more critical than those for the gear reducer drive system.

For the gear reducer drive train, the primary goal with drive train alignment in to minimise vibration in the system. Contact between the motor rotor and stator is prevented by the proper alignment of the rotor shaft in the bearings mounted inside the motor frame.

For the gearless drive train, the primary goal becomes preventing contact between the motor rotor and stator. There are no bearings inside the motor case. The rotor is suspended at the end of a cantilevered drive shaft and the rotor alignment is only maintained by the same bearings that support the drive pulley. This creates large bending moments in the shaft. The resulting shaft deflection reduces the air gap between the rotor and the stator.

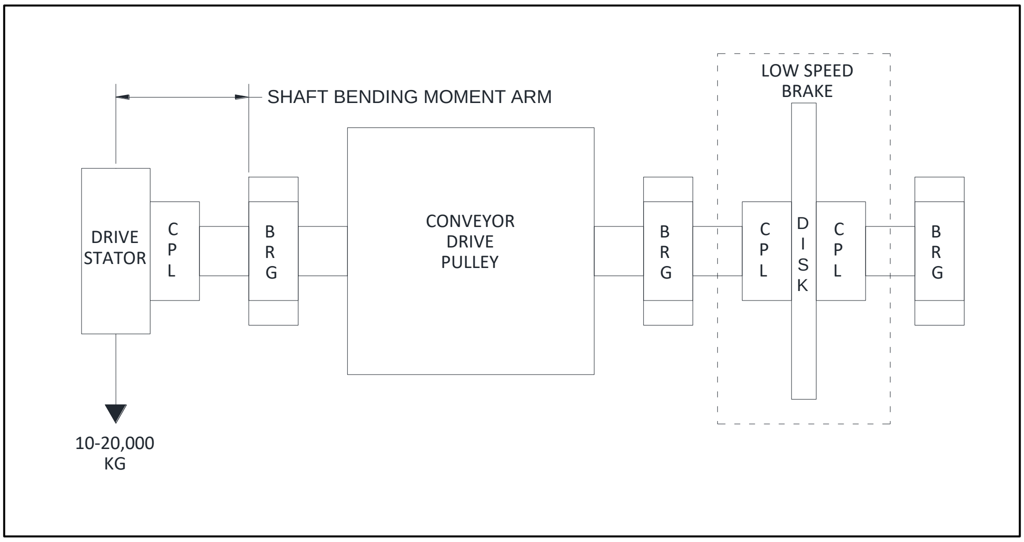


Figure 3: Gearless Drive Train with Stator Removed

There are many factors that affect the alignment between the conveyor drive pulley shaft and the rest of the drive system:

- Shaft deflection in both types of drive trains can be affected by
  - Bearing alignment changes due to structural damage or deformation, foundation settling, or disassembly for maintenance
  - Pulley weight
  - Overhung loads from swing base mounted gear reducers, brake disks, backstops, couplings and so forth
  - Varying belt tensions due to changes in conveyor load
- Shaft deflection in gearless drive motors can be also be affected by
  - Overhung load from the motor rotor
  - Magnetic attraction between rotor and stator

Some of the forces causing these deflections are more or less constant in magnitude and direction, such as the pulley weight and the weights of the components creating the overhung loads. These forces are present during installation, and can be factored into the initial drive train alignment. Other forces can vary considerably over time, especially the belt tensions. These varying deflections created by these dynamic forces must be allowed for in the drive train alignment tolerances.

There are three principal strategies in current use to reduce the magnitude of the variations in dynamic pulley shaft deflections:

- Flexible couplings
- Swing base mounted drive systems
- Increased pulley shaft diameter

Only the last strategy can be used with the gearless drive system.

Additionally, since the rotor must be attached directly to the drive pulley shaft to minimise shaft deflections, the option of placing a low speed brake system between the motor and the pulley no longer exists.

Summary

The gearless drive system provides an enormous reduction in the cost of the conveyor drive train by eliminating expensive gear reducers and related auxiliary equipment. These cost savings are offset by

- Increased costs of brake systems due to the higher, low speed torque ratings required.
- Decreased flexibility in selection of brake disc locations along the drive train.
- Increased cost and decreased availability of backstops with sufficient torque capacity.
- Elimination of the use of reasonably sized flywheels to increase drift stop times on inclined conveyors.
- Reduction in the drift stop times of all conveyors.
  - Inclined conveyor performance is degraded.
  - Low-lift conveyor performance is relatively unchanged.
  - Decline (downhill) conveyor performance is enhanced.
- Decreased tolerance for pulley shaft deflections.
- Elimination of the use of flexible couplings and drive swing bases to offset dynamic variations in pulley shaft deflection.

CASE STUDIES

The following case studies will illustrate some of the problems associated with the use of gearless drive motors.

- Case 1 – Stopping time coordination between an upstream decline conveyor and an incline conveyor.
- Case 2 – Dynamic tension waves created by low motor inertia in a power out stop on an incline conveyor.
- Case 3 – Critical motor alignment issues on an overland conveyor system.

CASE 1 – STOPPING TIME COORDINATION

This case involved a bid design for 2 conveyors in a series of 5 conveyors.

Figure 4 and Figure 5 show the profiles for the 2 conveyors in question: A Decline conveyor directly upstream from an Incline conveyor. The geometry of this situation was dictated by the local topography, which is fairly typical of mountainous regions around the world. Table 3 gives general specifications for each conveyor.

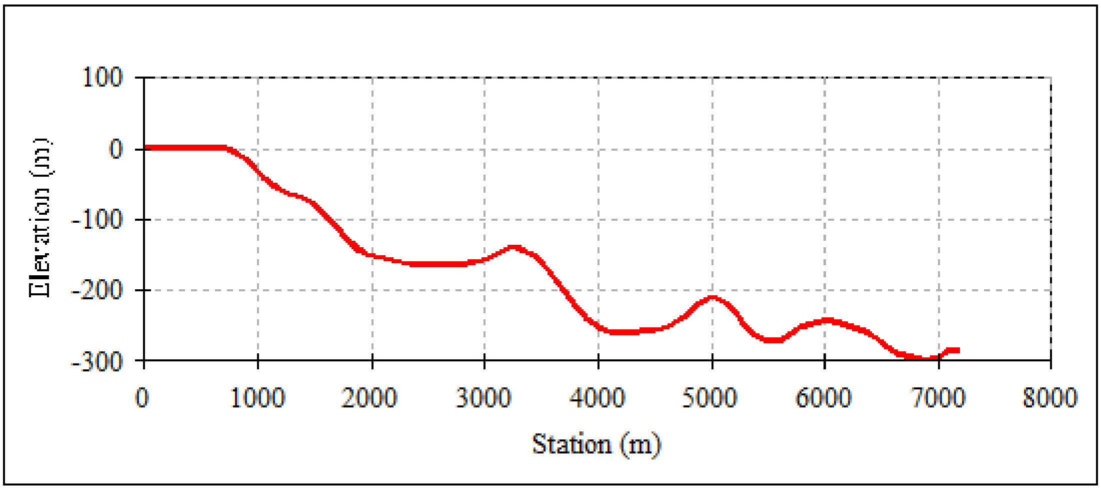


Figure 4: Profile of Decline Conveyor – Case 1

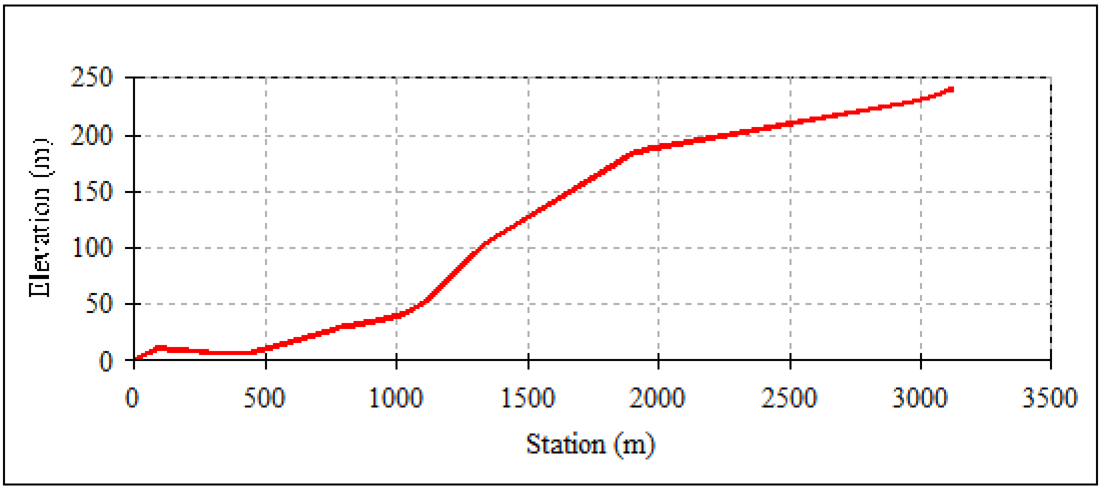


Figure 5: Profile of Incline Conveyor – Case 1

Table 3: Conveyor Specifications – Case 1

Conveyor	Capacity	Speed	Belt Strength	Gearless Motors	Motor Speed	Total Motor Inertia	No. of Drive Pulleys	Stopping Time
	tph	m/s		kW	rpm	kg-m^2		sec
Decline	9400	7	ST-5000	2x3500	87	24000	1	66
Incline	9400	7	ST-5000	3x3500	87	36000	2	11

The 66 second stopping time for the decline was the fastest brake time that could be achieved while maintaining a reasonable dynamic safety factor on the ST-5000 belt. The 11 second stopping time for the incline conveyor was the drift time before the backstop or brake system had to engage to prevent runback.

The torque rating for the backstop or brake on the Incline conveyor was 537 kN-m. This is well within the range of torque ratings for backstops offered by several manufacturers, although not all of the backstops in this torque range can also handle the motor rpm.

Figure 6 shows the size of the surge bin required to contain the excess material discharged by the decline conveyor.

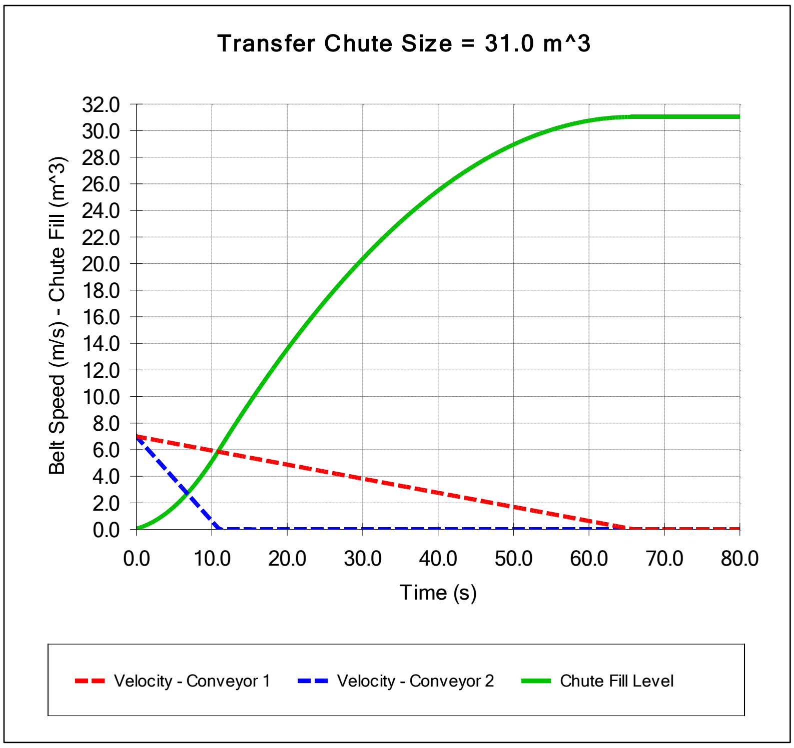


Figure 6: Required Surge Bin Size - Case 1

For comparison, for an assumed angle of repose of 40^o, 31 m^3 of material would create a conical stockpile about 2 m high and 6 m in diameter.

The standard strategy for coping with this material overflow is to add a flywheel to the Incline conveyor to match its stopping time to the upstream decline conveyor. Dynamic analysis of this conveyor determined that the gearless drive motors would have to run at 1100 rpm to achieve a stopping of 65 seconds for the incline conveyor. This is equivalent to a gear ratio of 12.6:1. At the given 87 rpm motor speed, the total moment of inertia of the drive system, including flywheel, would be approximately 160 times the moment of inertia of the gearless drive with no flywheel.

If each gearless drive has its own flywheel, each flywheel would require a moment of inertia of

$$I = 160\ x\ 12\ 000\ kgm^2 = 1\ 920\ 000\ kgm^2$$

Considering a solid cylindrical flywheel, the moment of inertia, is calculated on the basis of

$$I = \frac{1}{2}Mr^2$$

Where

$$M = mass$$

$$r = radius$$

Table 4 shows specifications for a range of cylindrical steel flywheel sizes with this moment of inertia. Using one of these huge flywheels is not a reasonable proposition. If it were not for other problems the low motor inertia creates for an Incline conveyor, the most cost effective choice would be to opt for the 31 m^3 surge bin instead.

Table 4: Low Speed Flywheel Specifications – Case 1

Flywheel Diameter	Flywheel Radius	Required Flywheel Mass	Required Flywheel Length
m	m	kg	m
1	0.5	15360000	2507
2	1	3840000	157
3	1.5	1706667	31

CASE 2 – DYNAMIC TENSION WAVES DURING STOPS

This case involved a forensic audit of an existing incline conveyor with a gearless drive system. The principal problem was excess belt wear that had nothing to do with the gearless drive system. During the audit, problems were discovered with conveyor performance during power out stops that were creating transient belt tensions well above the return side low tension bend pulley design limits. Additionally, these tension waves were causing excessive side motion and forces along the conveyor's horizontal curve section.

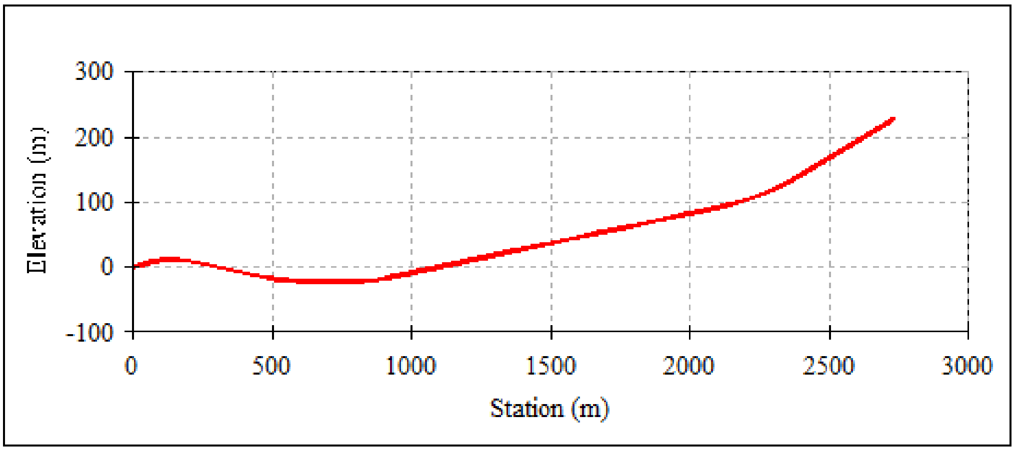


Figure 7: Profile of Incline Conveyor – Case 2

Table 5: Conveyor Specifications – Case 2

Conveyor	Capacity	Speed	Belt Strength	Gearless Motors	Motor Speed	Total Motor Inertia	No. of Drive Pulleys	Stopping Time
	tph	m/s		kW	rpm	kg-m^2		sec
Incline	9400	6.5	ST-5300	2x4400	66	48000	2	11

The horizontal curve occurs between Station 1360 and Station 2016.

The conveyor uses a motor-assisted stop ramp when power is available, with a parking brake that activates when the belt speed reaches about 1% of the nominal belt speed (0.06 m/s). The fast-acting parking brakes on each drive pulley are used instead of a backstop, and each brake has a minimum torque rating of about 820 kN-m. An equivalent backstop for this conveyor would require a torque rating of 807 kN-m. Backstops of this size are pushing the limits of backstop manufacturer's current catalogue offerings.

The take-up is fixed, with an empty running tension of approximately 326 kN. The dynamic analysis results shown in Figure 8 compare the extreme variations in tension that occur during a fully loaded power out stop, at various pulley locations along the return side of the belt. The take-up (TU) is near the tail (Tail) and the other return side low tension bend pulleys (P5) are near the drive station, around Station 2520.

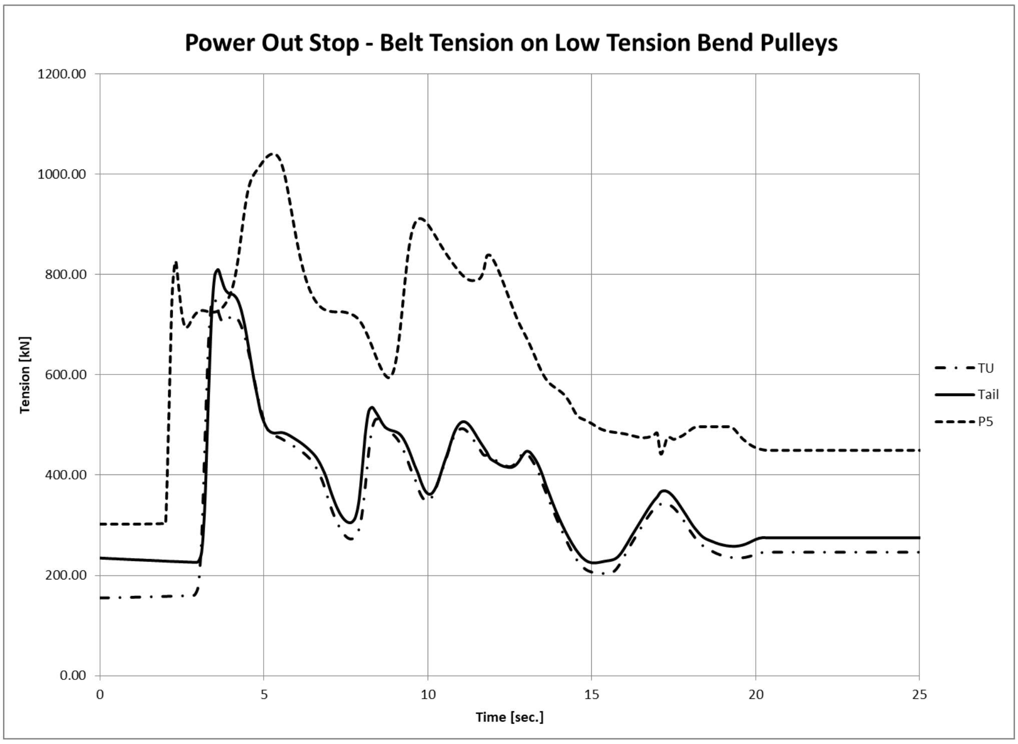


Figure 8: Take-up Tension during a Power Out Stop – Case 2

The peak design tensions used for the low-tension bend pulleys was 760 kN, which is barely acceptable for the take-up tensions, but not quite enough for the tail and other low tension bend pulleys in the system.

Figure 9 shows that that this conveyor also suffers from substantial runback at the tail (Tail) and at the drive station (DR 2). The drive station is included here as this is where the belt speed is monitored for motor and brake control signals.

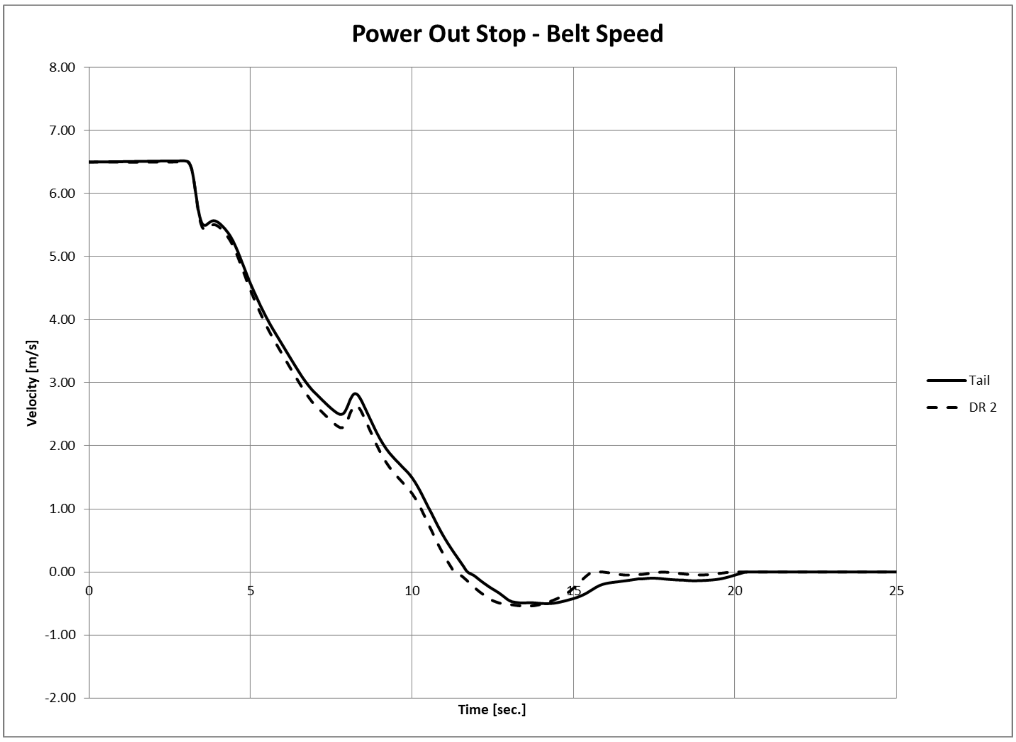


Figure 9: Runback during Power Out Stop – Case 2

An analysis of the belt side travel in the horizontal curves, and the impact forces on the return side idlers frames is shown in Table 6. There are no side guide rolls along this curve, and any displacement of 194 mm in either direction indicates that the belt has contacted the idler frame. This analysis indicates that as the belt tension oscillates during the power out stop, the belt repeatedly slams into the idler frames.

Table 6: Displacements and Forces in Horizontal Curve – Case 2

Start Sta.	End Sta.	Displacement [mm]				Contact Force [N]			
[m]	[m]	Min	Empty	Run	Max	Min	Empty	Run	Max
2016	1981	10	194	194	194	0	160	150	1199
1981	1877	-169	111	111	194	0	0	0	896
1877	1774	-155	122	122	194	0	0	0	900
1774	1670	-142	133	133	194	0	0	0	902
1670	1567	-129	144	147	194	0	0	0	905
1567	1360	-127	155	160	194	0	0	0	906

A comparison of actual operational data to a dynamic computer model is shown in

Figure 10: Operation Data for Power Out Stop at 5959 tph – Case 2

and

Figure 11. While the dynamic model shows a 10% higher peak tension than the operational data indicates, the overall similarity of the take-up tension curves shows that the dynamic model provides a fair representation of actual operating conditions.

For the dynamic model in

Figure 11 the brake timing was adjusted in an attempt to eliminate runback. While runback at the drive station was successfully eliminated, some runback at the tail is still occurring, and could not be completely eliminated without activating the brake at a very high belt speed, and causing higher belt tension peaks.

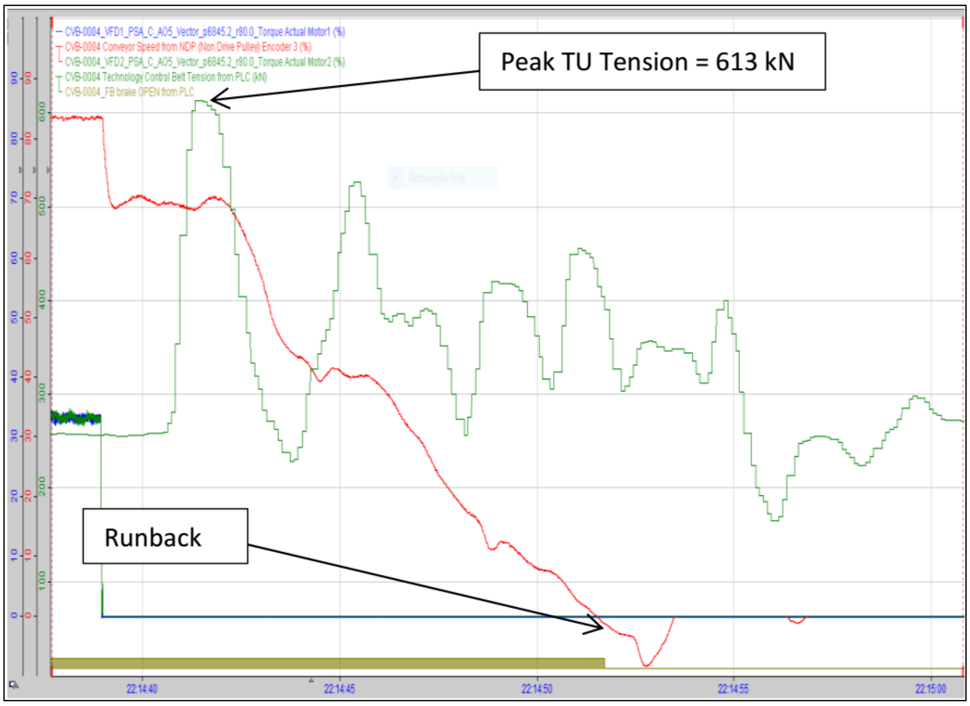


Figure 10: Operation Data for Power Out Stop at 5959 tph – Case 2

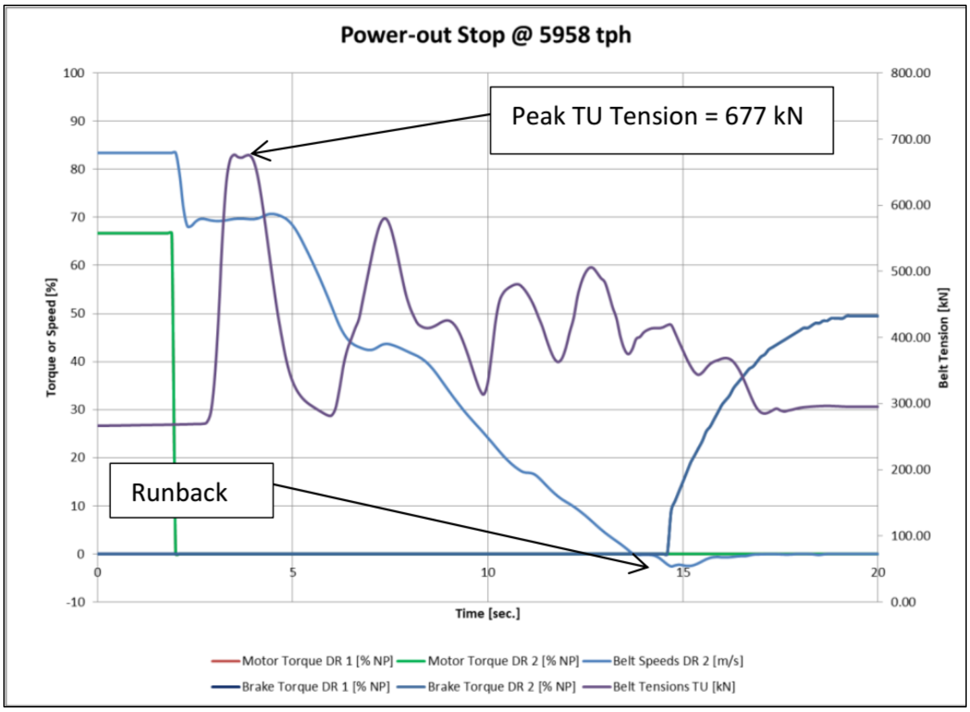


Figure 11: Dynamic Simulation for Power Out Stop at 5958 tph – Case 2

The peak belt tensions can be significantly reduced and the runback eliminated by applying additional inertia to the drive trains. Figure 12 and Figure 13 show the results of the dynamic analysis if a flywheel with sufficient moment of inertia to extend the drift stop time to 30 seconds is added to each drive pulley.

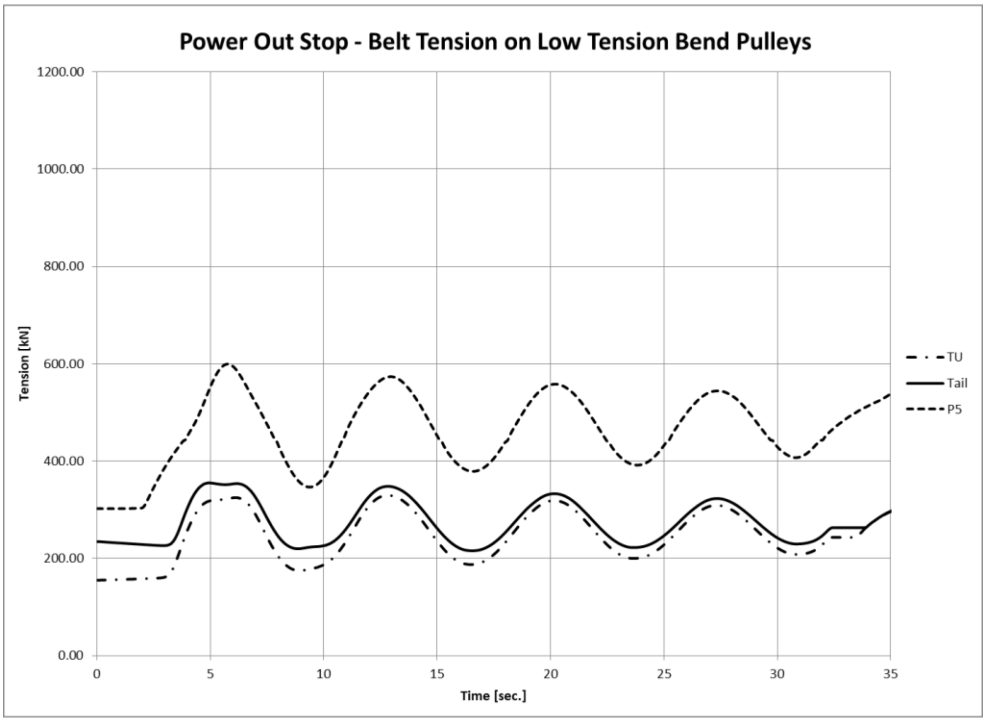


Figure 12: Take-up Tension during a Power Out Stop – Case 2 with Flywheel

Figure 13: Runback during Power Out Stop – Case 2 with Flywheel

Since this conveyor had no stopping time coordination issue with the upstream overland conveyor during normal, motor-assisted stops, the goal here was to achieve a drift stopping time equal to the normal stopping time of 30 seconds. Analysis indicated that this would require a moment of inertia equivalent to the gearless drive motors running at 560 rpm. This is equivalent to a gear ratio of about 8.5:1. At the given 66 rpm motor speed, the total moment of inertia of the drive system, including flywheel, would be approximately 72 times the moment of inertia of the gearless drive with no flywheel.

The results of dynamic analysis with such a flywheel are shown in Figure 12 and Figure 13. Peak tensions during the power out stop have been reduced by over 40%, and runback has been eliminated. Side forces in the horizontal curve have also been reduced to more reasonable levels, if not eliminated entirely.

Table 7: Displacements and Forces in Horizontal Curve – Case 2 with Flywheel

Start Sta.	End Sta.	Displacement [mm]				Contact Force [N]			
[m]	[m]	Min	Empty	Run	Max	Min	Empty	Run	Max
2016	1981	-30	154	154	194	0	160	150	42
1981	1877	-194	-21	-21	37	34	0	0	0
1877	1774	-193	-7	-7	51	0	0	0	0
1774	1670	-176	7	7	65	0	0	0	0
1670	1567	-160	21	23	79	0	0	0	0
1567	1360	-150	35	39	93	0	0	0	0

If each gearless drive has its own flywheel, each flywheel would require a moment of inertia of

$$I = 72\ x\ 24\ 000\ kgm^2 = 1\ 728\ 000\ kgm^2$$

Table 8 shows the range of low speed flywheel sizes required. These are similar in size to the low speed flywheels required for Case 1.

Table 8: Low Speed Flywheel Specifications – Case 2

Flywheel Diameter	Flywheel Radius	Required Flywheel Mass	Required Flywheel Length
m	m	kg	m
1	0.5	13824000	2257
2	1	3456000	141
3	1.5	1536000	28

This leaves 1 of 2 options:

- Beef up the pulleys to withstand the peak belt tensions, and install almost 1000 side guide rolls to reduce the impact forces on the return side idler frames.
- Install a high speed flywheel.

A flywheel running at 1800 rpm would require a gear reducer with a gear ratio of 27.27:1, rated at 4400 kW power and 636 kN-m torque. This gear ratio would allow the flywheel to have a high speed inertia of only 2324 kg-m^2.

Table 9 shows a range of possible flywheel sizes. While the larger diameters would have serious flexibility issues, the smaller diameter flywheels are certainly within the range of flywheels already in operation on other incline conveyors with gear reducer drives.

Table 9: High Speed Flywheel Specifications – Case 2

Flywheel Diameter	Flywheel Radius	Required Flywheel Mass	Required Flywheel Length
m	m	kg	m
1	0.5	18589	3.03
2	1	4647	0.19
3	1.5	2065	0.04

Unfortunately, this solution eliminates the cost savings obtained by using the gearless drive motors in the first place.

CASE 3 – CRITICAL MOTOR ALIGNMENT ISSUES

Whereas low inertia only has adverse effects for incline conveyors, motor alignment issues are the same for all gearless drive conveyors, regardless of the material lift. This case involves an essentially flat overland conveyor about 3.4 km long with a 22 m lift.

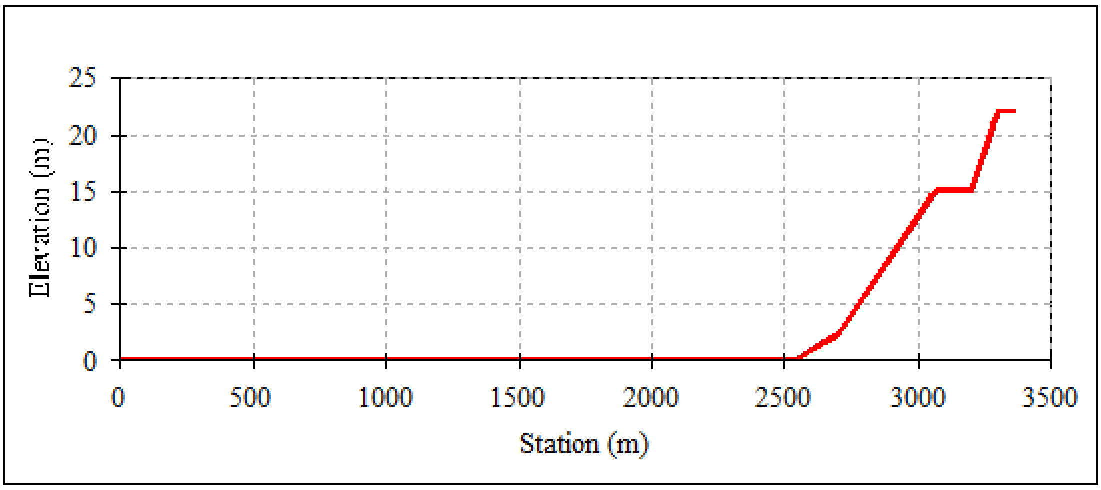


Figure 14: Profile of Flat Overland Conveyor – Case 3

Table 10: Conveyor Specifications – Case 3

Conveyor	Capacity	Speed	Belt Strength	Gearless Motors	Motor Speed	Total Motor Inertia	No. of Drive Pulleys	Stopping Time
	tph	m/s		kW	rpm	kg-m^2		sec
Flat OL	12550	6.72	ST-4500	1x5500	131	24000	1	29

The principal problem in this case was measured drift in the rotor alignment over several months that had the potential to exceed the allowable air gap tolerance between the rotor and the stator. A detailed review and analysis was made of data collected by the maintenance and operations personnel during installation and subsequent operations over a several month period. This analysis identified several physical factors that affect the air gap during installation and operations.

Installation Tolerances and Allowable Deviations

During installation, measurements are made with no tension on the belt, no torque on the motor, and the motor at ambient temperature. Any deflections of the shaft due to the masses of the shaft or the pulley, or due to the overhung load of the rotor mass are already present during the alignment process.

The air gap has a nominal measurement of 8 mm, an allowable deviation during installation of 0.2 mm between centres, and an allowable deviation of the centre position of up to 1.6 mm. In practice this can be difficult to achieve. Figure 15 shows the locations where air gaps are typically measured. There is an inductive sensor at each of these locations which measures air gaps during operations. Except for the 07:00 location, these same locations are measured using physical instruments during installation. There is only one sensor at each location, but during installation, separate measurements are taken at the ring end and coupling end of the rotor.

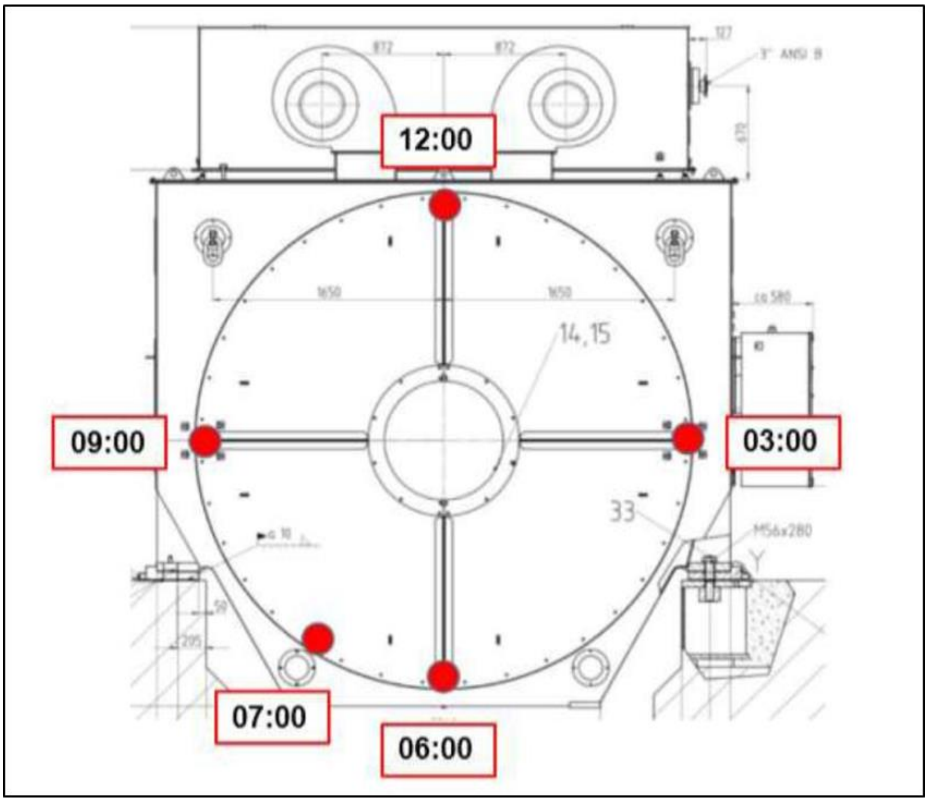


Figure 15: Air Gap Measurement Locations.

Table 10 shows the results of the installation measurements. The difference in the In Plane Average between the Vertical and Horizontal Planes indicates that either the rotor or the stator (or both) are slightly out-of-round (not perfectly circular) by about 0.5 mm. The Vertical Offset shows that the rotor is hanging slightly lower, toward the 06:00 sector than the stator centre. The Horizontal Offset shows that the rotor is slightly closer to the 09:00 sector. These alignment errors will create magnetic forces between the rotor and the stator that will increase the rotor deflection when the motor is running. For the out-of-round condition, the magnitude of these deflections will vary in direction as the motor spins. For the vertical and horizontal offsets, the deflections will always be in the direction of the offset.

Table 10: Air Gap Measurements during Installation

Sector	**Pole No.**	**Ring Side**	**Coupling Side**	**Sector Average**	**In Plane Average**
		[mm]	**[mm]**	**[mm]**	**[mm]**
Vertical Plane					
0^{o} (6:00)	16	7.0	7.4	7.2	7.5
180^{o} (12:00)	8	7.6	8.0	7.8	
Rotor End Average		7.3	7.7		
Vertical Offset		0.3	0.3		
Horizontal Plane					
90^{o} (3:00)	11	8.0	7.8	7.9	8.0
270^{o} (9:00)	4	8.2	8.0	8.1	
Rotor End Average		8.1	7.9		
Horizontal Offset		0.1	0.1		
All Around Average		7.7	7.8	7.8	

Running Deflections

The stator exerts a magnetic force on the rotor with a magnitude proportional to the air gap, and a force constant of 121 kN/mm. Figure 16 shows the principal dimensions of the pulley and rotor assembly. This diagram was used to perform a simple beam deflection analysis based on the magnetic forces developed for the out-of-round and offset conditions set by the initial installation deviations. The results of these calculations are shown in Table 11

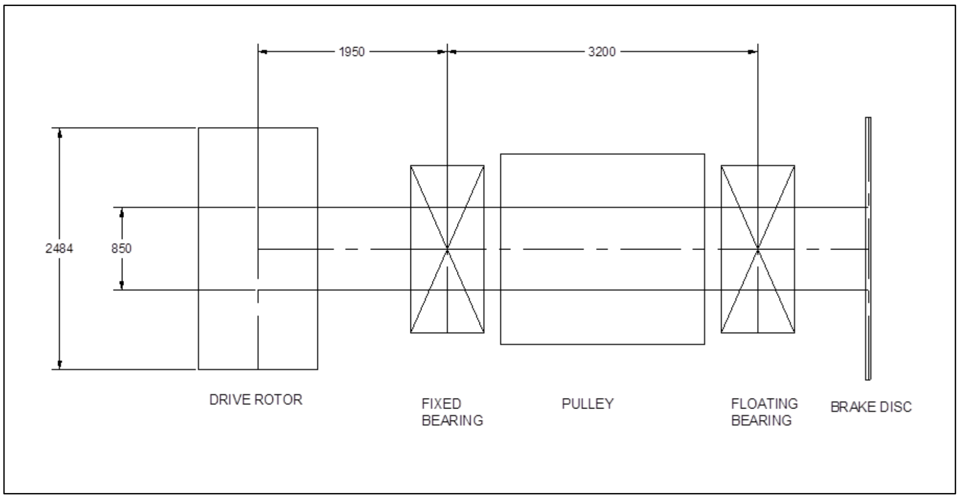


Figure 16: Pulley and Rotor Assembly

Table 11: Magnetic Deflections due to Initial Alignment

Source	Alignment Error	Unilateral Magnetic Force	Additional Deflection
	[mm]	[kN]	[mm]
Out of Round	0.5	60.5	0.08
Vertical	0.3	36.3	0.05
Horizontal	0.1	12.1	0.02

Figure 17 shows the drive system configuration. In this particluar case, the clockface reference used for the sensor array is on the opposite side of the motor, resulting in the reversal of the 3:00 and 9:00 positions.

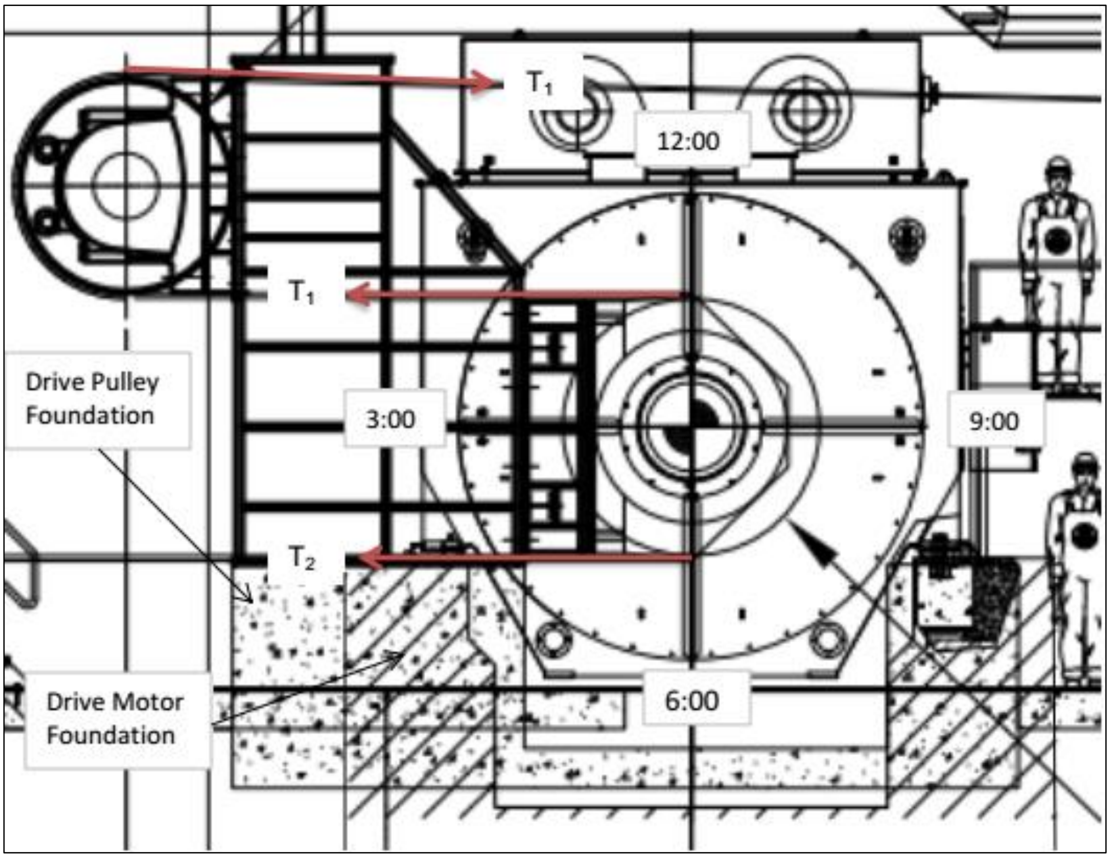


Figure 17: Drive System Configuration

The belt tensions on the drive pulley when the belt is running create a pulley shaft deflection at the center of the rotor that is opposite in direction to the belt tension vectors shown. Estimates of the expected deflections over the range of operating loads were made, using simple beam analysis and assuming the belt tension is evenly distributed along the shaft between the bearings face. The distributed load is not really a good assumption, so a factor of 2 was included in the calculation as a safety factor for this initial estimate. These deflections will increase the magnetic forces on the rotor, and an additonal increase in deflection from this cause must be allowed for. Table 12 shows the estimated magnitude of the deflections for the Empty and Full load cases. These deflections will increase the air gap at the 3:00 sector and decrease them at the 9:00 sector.

Table 12: Deflections Due to Belt Tension

Belt Load	$T_1 + T_2$	Belt Tension Deflection	Magnetic Deflection
[tph]	[kN]	[mm]	[mm]
0	1353	0.4	0.07
12550	2050	0.7	0.10

Looking back at Figure 17, the T1 belt tension on the bend pulley combined with the T2 tension on the drive pulley create a huge bending moment on the pulley support post, which is transmitted to the supporting foundation. This foundation is separate from the drive motor foundation. Any shift in the pulley foundation due to the over-turning moment caused by the belt tension, or any differential settling of the underlying ground between these 2 foundations can lead to unpredictable changes in the rotor to stator air gap.

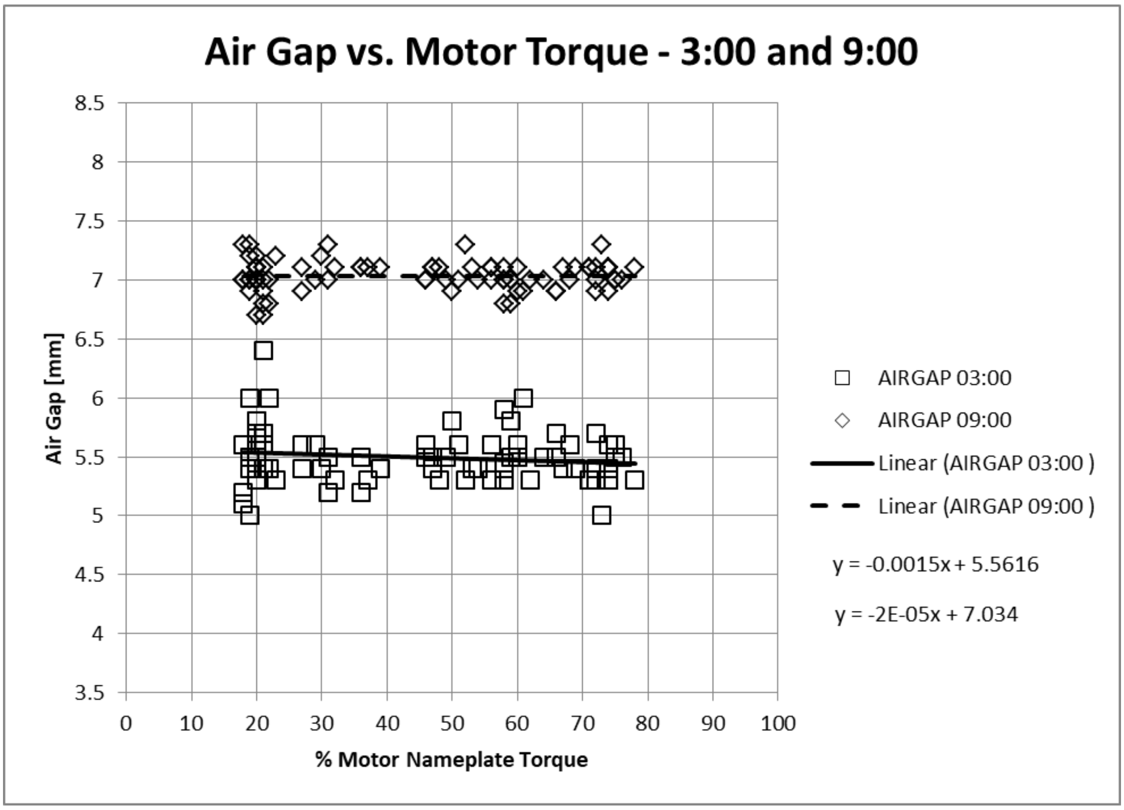


Figure 18: Air Gap vs. Motor Torque – Horizontal

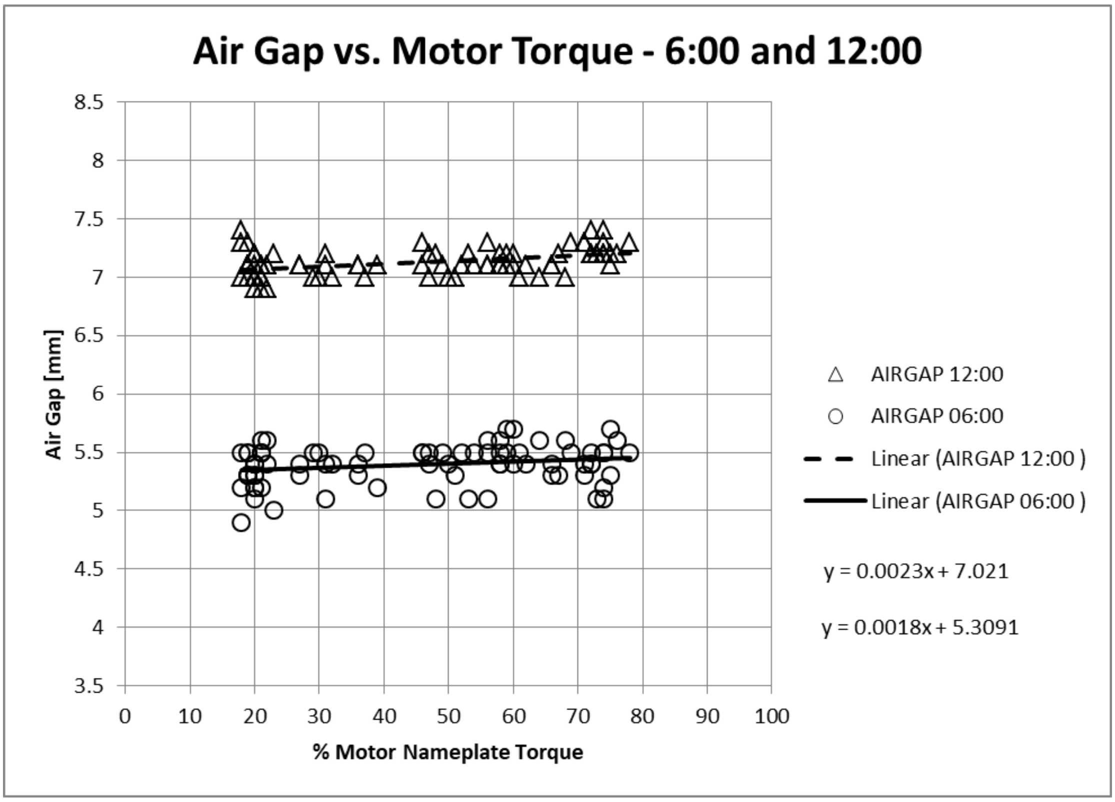


Figure 19: Air gap vs. Motor Torque – Vertical

Reviewing the available operational data to compare with our estimates shows some anomalous results. **Error! Reference source not found.** shows very slight decreases in the 3:00 and 9:00 air gaps as the motor torque and belt tension increase. On the other hand, Figure 19 shows unexpected increases in both the 6:00 and 12:00 air gaps. The spread in the range of data points, especially at the 20% torque, empty load condition leads to the conclusion that the motor torque is not the dominant factor in the air gap variance observed here.

Thermal Expansion

As the motor runs, the rotor and stator both experience resistance heating, and expand. Due to the physical constraints imposed by motor frame and the rotor coupling, this expansion reduces the air gap.

There is no easy way to estimate this in advance, but operational data can provide an indication of the magnitude of this effect. The operational data collected over a 2 month period was taken at different times of the day, under a range of loading conditions and ambient temperatures. Assuming that variations due ambient temperature changes and belt tensions are averaged out over the course of time, what is left can be attributed to thermal expansion of the rotor and stator. **Error! Reference source not found.**13 shows that there is a quite uniform reduction in the air gaps between the installed measurements and the sensor readings taken when the belt is running.

Table 13: Air Gaps – Installed vs. Running

		Vertical			Horizontal		
Sector		0^{o} (6:00)	180^{o} (12:00)	Avg.	90^{o} (3:00)	270^{o} (9:00)	Total
Installed Air Gap	[mm]	7.2	7.8	7.5	7.9	8.1	8.0
Running Air Gap	[mm]	5.4	7.1	6.2	5.5	7.0	6.2
Thermal Expansion	[mm]			1.3			1.8

Factoring out the 0.5 mm out-of-round condition observed previously in Table 10, the estimated decrease in air gap that can be attributed to thermal expansion is approximately 1.5 mm. Any addition due to increased magnetic forces in already included in this estimate.

Since the operational data was collected at different times of the day, it is possible to estimate the effects of thermal expansion due to diurnal variations in the ambient temperature. At the location of this conveyor, the ambient temperature varies from about 5°C to about 20°C between day and night. The operational data shown in Figure 19 shows a roughly sinusoidal variation in the air gap vs. the time of day the air gap data was collected. The magnitude of this variation is approximately 0.3 mm, or about 0.02 mm/°C.

Between these 2 estimates, it is possible to derive an average motor temperature of

$$1.3\ mm/(0.02\ mm/°\text{C}) = 65°\text{C}$$

above ambient.

For a reality check, this is consistent with the allowable temperature rise for a large motor (> 1120 kW) with NEMA Class A insulation.

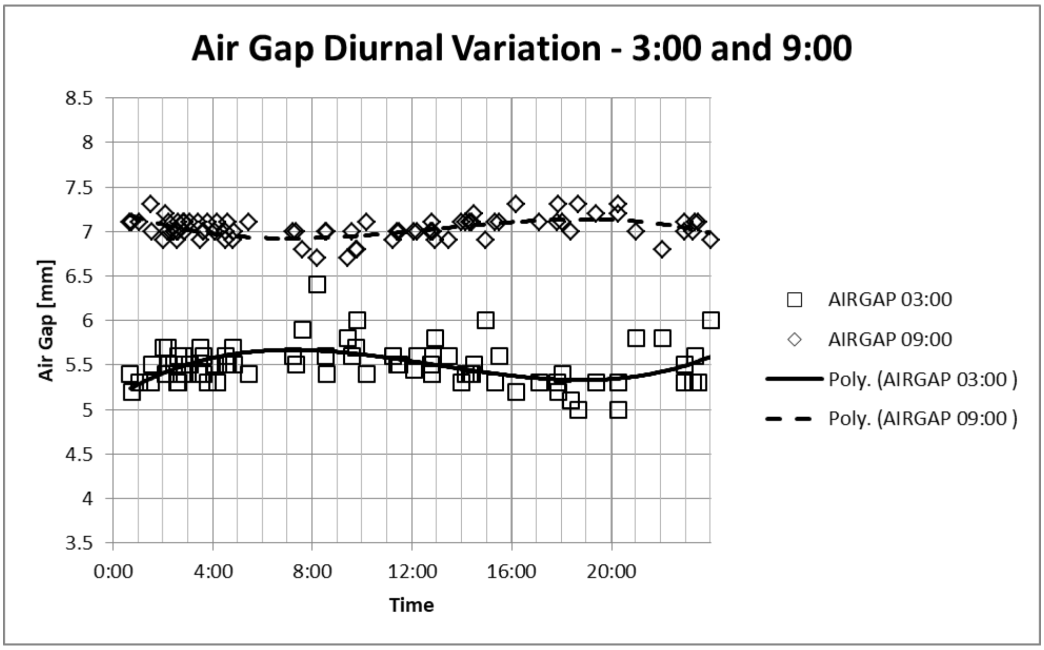


Figure 19: Diurnal Air Gap Variations

Summary of Shaft Deflection Estimates

Table 14 includes all of the predictable shaft deflections.

Table 14: Sum of Shaft Deflections

Source of Deflection	Approximate Magnitude
	[mm]
Installation tolerances	1.3
Magnetic forces - Installation tolerances	0.1
Belt Tension at full load	0.7
Magnetic forces - Belt tension	0.1
Thermal expansion	1.3
TOTAL	3.5

The nominal air gap during installation is 8.0 mm. The estimated total reduction in air gap due to all predictable sources is about 3.5 mm, with 4.5 mm of air gap left after all these reductions are accounted for. The air gap sensors are programmed to signal an "Alarm" condition whenever the air gap is less than 4 mm, and will trip the motors if the air gap is less than 3.5 mm. The safety margin for any other source of shaft deflection is about 1 mm.

This makes structural rigidity and stability a critical issue. It is imperative that the structural and foundation design be designed based on stringent deflection criteria, and not on an allowable stress or strength of material basis.

To guard against differential settling and foundation shifts due to over-turning moments, it would be best if the motor and pulley foundations were strongly joined, so that any shift in the foundation moved the entire drive train as a single unit.

Pulley Shaft Design

Referring back to Figure 16, the shaft shown here is simplified to make the deflection estimates easier to calculate. The 850 mm shaft diameter indicated is the shaft diameter at the locking ring assembly. Not shown on this drawing is the pulley diameter of 1962 mm.

The large shaft diameter exceeds most locking ring manufacturers catalogue offerings, decreasing availability and increasing the lead time required for pulley manufacture. The large shaft also decreases the available distance for adequate hub thickness to keep hub stresses below yield, or enough room to develop a suitable end disc profile that allows the end disc enough flexibility to prevent fatigue failure. The large shafts, large locking devices and large bearings required also increase the pulley weight substantially.

CONCLUSIONS

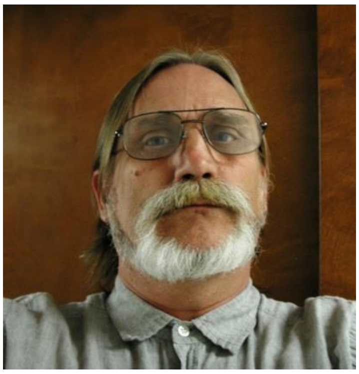
Gearless drive systems are a cost effective alternative for high capacity downhill and low lift overland conveyors.

Due to their reduced motor inertia, gearless drive systems are not suitable for high lift conveyors.

Special consideration must be taken with the design of drive pulley support structures and pulley and motor foundations to prevent loss of rotor alignment due to over-turning moments induced by belt tensions or other structural loads, or due to differential settling of the ground under the foundations.

Special consideration must be taken in the design of pulley shafts to limit shaft deflections and maintain an adequate air gap between the gearless drive rotor and stator.

ABOUT THE AUTHOR



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