

Optimizing Conveyor Take-up Systems using Dynamic Analysis and the Implementation of Capstans

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ABSTRACT

Although it may be perfectly acceptable to neglect the belt's elastic behavior for many conveyor designs, this can lead to disastrous consequences for others. So how can a designer know in advance if a dynamic analysis is required? Unfortunately, in many cases it is impossible to foresee or predict dynamic problems prior to performing the actual analysis. However, a basic understanding of the principles of conveyor dynamics, as well as knowledge regarding common problem areas, can significantly help the designer make more educated decisions during the initial design stage. This paper will demonstrate how static calculations alone can incorrectly represent the true behavior of the conveyor system. It will also discuss the importance of correctly sizing and locating the take-up system to provide the most economical conveyor design. The use of capstans will also be discussed followed by two examples demonstrating their proper operation.

INTRODUCTION

Over the past 20 years numerous papers have been written on the subject of conveyor dynamics. These papers have discussed various theoretical models and numerical techniques used to properly analyze the elastic response of conveyor systems. Unfortunately to date, no complete reference book contains a complete discussion of conveyor dynamics from start to finish. Perhaps this is due to the fact that almost every conveyor is unique. Each with its own specific design requirements and challenges. In order to battle these challenges the conveyor engineer must have a thorough understanding of the dynamic behavior of the conveyor system. In the end, the equipment selection must be such that it satisfies all required design criteria while maintaining the most economical cost.

This paper will demonstrate how static calculations alone can incorrectly represent the true behavior of the conveyor system. It will also discuss the importance of correctly sizing and locating the take-up system to provide the most economical conveyor design. The use of capstans will be discussed followed by two examples demonstrating their proper operation.

STATIC vs. DYNAMIC CALCULATIONS

Static conveyor calculations are used to determine the steady state belt tensions, power consumption, and general equipment selection for a conveyor system. Static analysis assumes that the conveyor acts as an inelastic body (CEMA, 1997). With this assumption both acceleration and deceleration forces can be calculated. These forces can then be added to the steady state tensions to estimate the starting, stopping, and drifting belt tensions.

In reality however, a conveyor belt (either fabric or steel cord) is by no means an inelastic body. To correctly analyze the belt's behavior, it must be broken up into a series of elastic springs, dashpots, and discrete masses (Nordell, 1996), (Lodewijks, 1997). Figure 1 shows a typical rheological element used to model a conveyor system. Two dimensional wave theory can then be used to determine the true elastic response and dynamic behavior of the conveyor system.

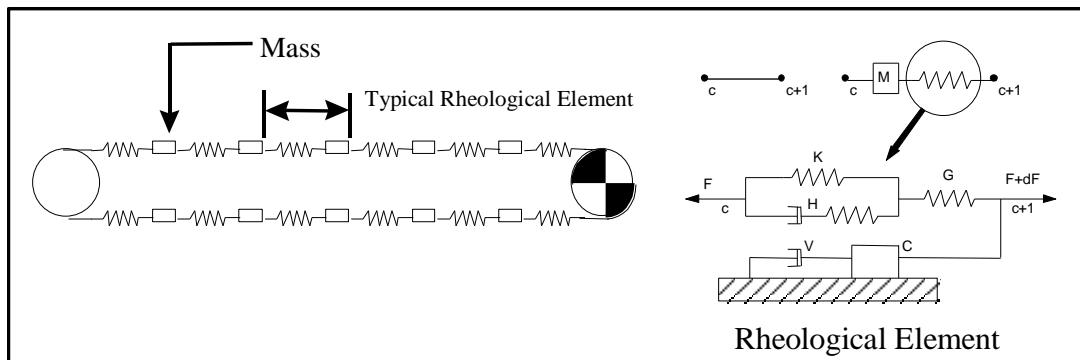


FIGURE 1 - A rheological model used to model the dynamics behavior of a conveyor system

DYNAMIC TUNNING

Dynamic tuning refers to the use of dynamic analysis to optimize the equipment selection and design of the entire conveyor system. For example, dynamic analysis may show that adding a flywheel to a particular conveyor could significantly improve the starting or stopping behavior of the conveyor. This could result in the use of a lower strength belt, reduced take-mass, and a lighter conveyor structure. These

cost saving could easily overshadow the additional flywheel cost and result in significant savings. Perhaps more importantly, the correct use of flywheels, other dynamic tuning devices, could significantly improve the reliability and safety of the conveyor. Some types of dynamic tuning are:

1. Drive and brake location
2. Take-up location (head or tail)
3. Flywheels (size and location)
4. Brakes (types, application times)
5. Capstans
6. Drive Controls (Starting and Stopping)

All of these items must be properly designed for each conveyor. Further as the conveyor geometry, tonnage, and other design parameters vary significantly from one conveyor to another, different methods of dynamic tuning will be required to achieve an optimal conveyor design. Often times, the counterweight location and mass are the most significant items. The counterweight regulates the maximum and minimum belt tensions, which in turn affect the belt strength, vertical and horizontal curve radii, and the entire conveyor structure. Minimizing the counterweight mass may result in significant cost savings on the overall system design. It should be noted however that this is not always true. In some instances it may actually be beneficial to increase the counterweight. The increased counterweight may result in the use of a larger idler spacing (thereby requiring fewer idlers), and could also decrease the power consumption of the conveyor.

GRAVITY VERSUS FIXED TAKE-UP

The function of a take-up is to control the belt tensions at the take-up pulley. The belt loading, thermal growth or reduction, and dynamic tension waves (during starting or stopping) cause variations in belt tensions during conveyor operation. Figure 2 shows a schematic of two simple conveyors. One has a gravity take-up and the other has a fixed take-up.

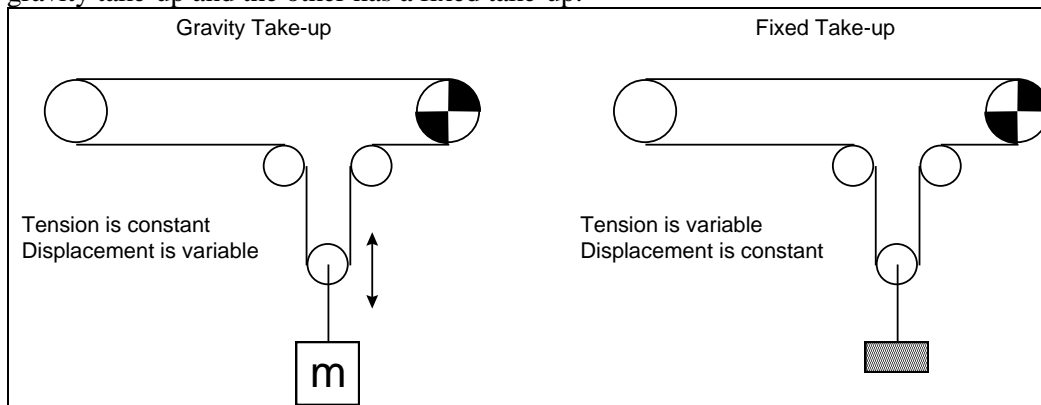


FIGURE 2 – Gravity and fixed take-up systems

The take-up pulley in a gravity take-up is connected to a free hanging counterweight. The displacement is not constrained and the tension at the take-up pulley is equal to half the counterweight mass. The gravity take-up controls the belt tensions by allowing the take-up pulley to move up or down as conditions on the belt change. The belt tension at the take-up pulley does not change except for variations caused by hysteresis (frictional losses in the sheaves or cable) in the take-up system and the counterweight inertia.

The take-up pulley in a fixed take-up is anchored to the ground. The displacement is constrained and the tension is variable. The fixed take-up does not control belt tension, however occasional manual adjustment is required when an upper or lower limit is breached. Once the pulley has been adjusted, it

does not move. Therefore, the tensions vary as the load and temperature change. During starting or stopping, the fixed take-up can experience very large variations in tension due to the conveyor dynamics.

Gravity take-up systems have the following advantages:

1. The belt tension at the take-up is constant.
2. Manual adjustment is infrequently required.
3. No PLC programming or set points are required

The primary disadvantage of a gravity system is that it is typically more expensive than a fixed take-up since at take-up tower and counterweight is required.

Although fixed take-ups are less expensive than gravity take-ups they have several disadvantage:

1. A fixed take-up has to be adjusted (either manually or automatically) as the temperature changes. This is due to the fact that the steel cables expand or shirk with the ambient temperature. Thermal growth on long overland conveyors can be significant. If adjustment is not made, the take-up may either bottom or top out.
2. The tensions at the take-up pulley vary with belt loading and temperature changes. The changes in tension must be taken into account. The variations in belt tensions can especially impact the return strand of the belt. Vertical and horizontal curves may have to be larger to accommodate the higher tensions. Belt turnovers, pulley design, vertical and horizontal curve, and structures may all be impacted.
3. An automatic adjustment take-up system requires the use of a “dead band” tension range. The dead band in itself results in higher and lower operating tensions (and therefore higher belt and pulley forces) from the optimum design set point. The system must then be over designed to accommodate these additional forces.

CAPSTANS

Capstans are devices used to increase the belt line tension at the take-up during stopping. The capstan restricts the motion of the counterweight and the take-up pulley. This results in either an increase or decrease in belt line tension at the take-up pulley. Capstans are typically designed by wrapping the counterweight cable around one or two additional sheaves. A small band or disk brake is then attached to the sheave and prevents motion of the counterweight (Figures 3 & 4). The counterweight will be frozen until the belt tensions rise or lower sufficiently to overcome the capstan brake. The capstan brake does not impede or alter the belt velocity at any time.

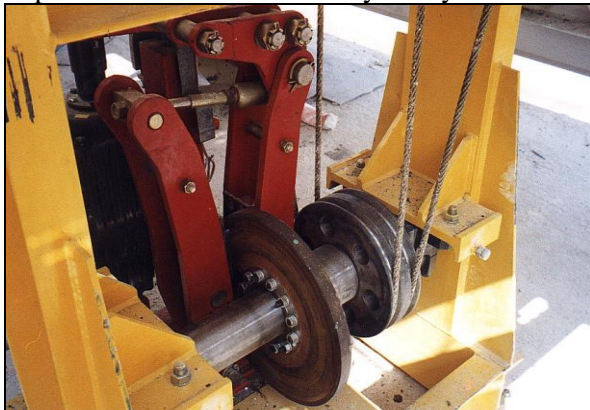


FIGURE 3 – A capstan installed using a disk brake FIGURE 4 – A capstan installed using a band brake

For example, Figure 5 shows a take-up with a tension of 87 kN. It is desired to raise the take-up tension by 35 kN during stopping. A capstan brake is placed on the counterweight cable and is applied during the stop. The figure shows tensions T1 and T2 on the cable. Tension T2 is always 174 kN. When the capstan is not applied, tension T1 is also approximately 174 kN and the counterweight will move up and down to equalize the tensions in the belt at the take-up pulley. The capstan brake applies 70 kN of force to the cable, which in turn applies 35 kN of force to the belt line tension. When the belt line tension is between 52 kN and 122 kN the take-up pulley will not move. If the belt tension is either lower or higher than these values, then the pulley will move until the tension stabilizes between these set points.

Since the capstan is only applied at specific times (i.e. during stopping), the overall conveyor can be designed around a much lower take-up tension.

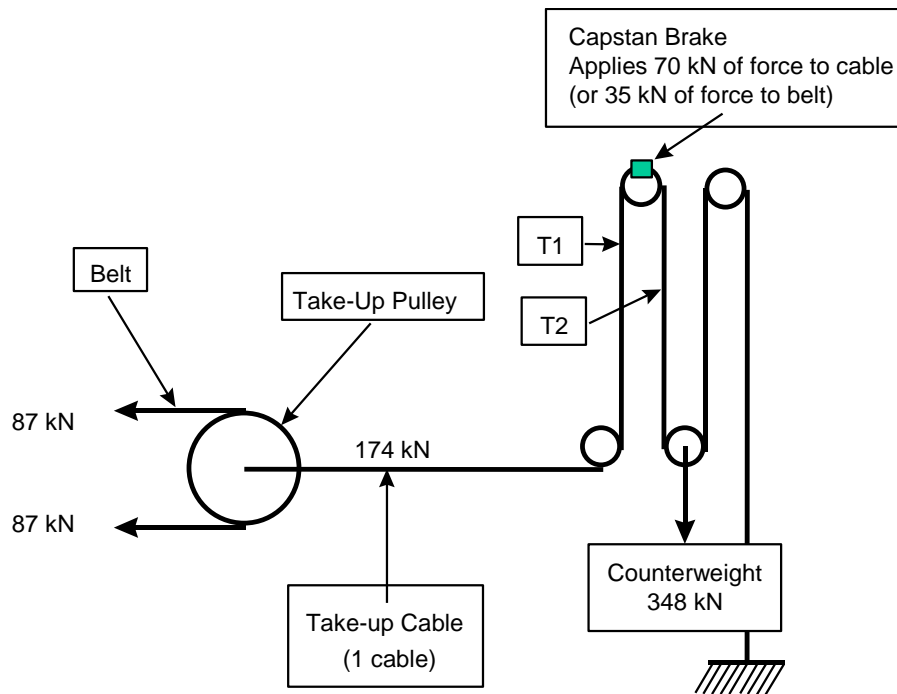


FIGURE 5 – Capstan schematic showing the beltline and cable tensions

EXAMPLE 1 – INCLINE CONVEYOR

The first example is a fictitious underground conveyor, created as an example to demonstrate the basic concept and application of a capstan. It also demonstrates the importance of take-up location and the pitfalls of designing a system using static calculations alone. Figure 6 shows the conveyor geometry. The conveyor is 1000m in length with 150m of lift. The design tonnage was set as 1500 T/H.

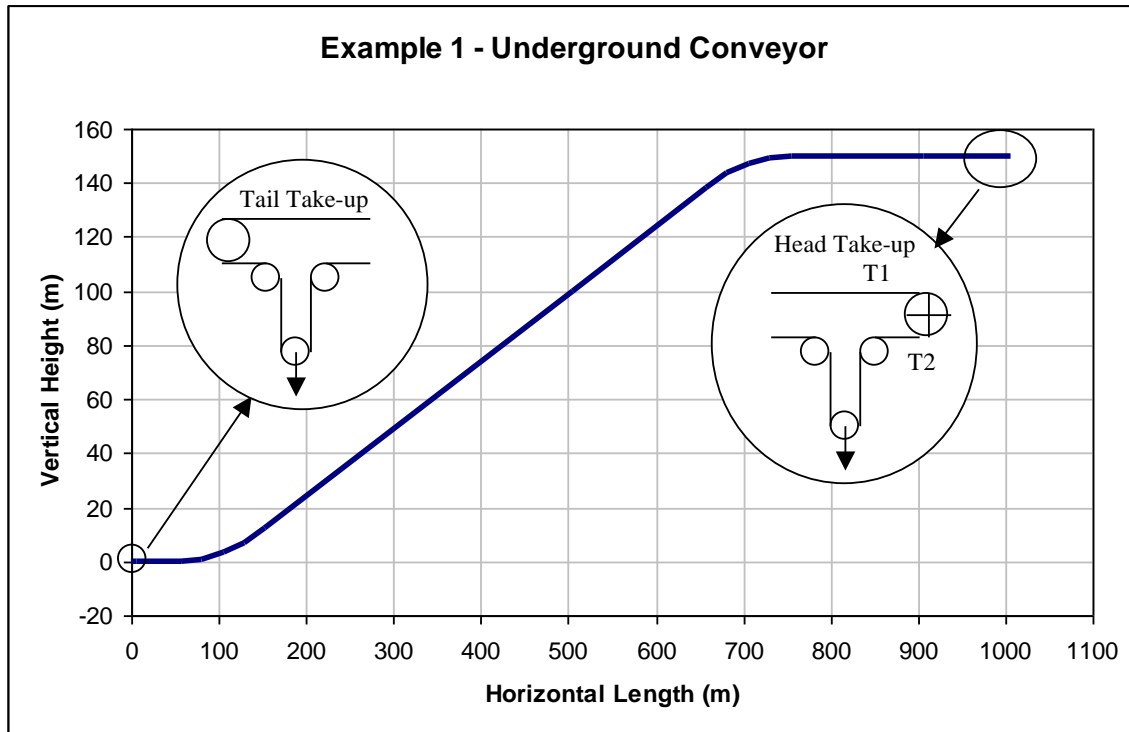


FIGURE 6 – Example underground conveyor geometry

The drive is located at the head of the conveyor, and a T2 slack side tension of approximately 85 kN is required. If the take-up is installed after the drive (at the head end) then the required take-up tension would be 85 kN. However, if the take-up is installed at the tail end, then the counterweight could be reduced to 55 kN. This is due to the fact that the return side belt tensions are increasing from the tail to head conveyor. Therefore the 55 kN tail tension would result in the same 85 kN T2 tension.

Although a tail take-up is ideal for this type of conveyor geometry, in many underground systems this is not physically possible. In this case the take-up must be moved to the head. This is where the true nature of static and dynamic analysis becomes apparent. If the engineer only takes into account the static analysis, then the two systems appear almost identical. Dynamically however, these two conveyors are worlds apart. When the conveyor shuts down the high T1 drive tension collapses. This in turn sends a dropping tension shockwave down to the tail end of the conveyor. If the take-up is located at the tail, then this shockwave can be absorbed and the minimum belt tensions will remain reasonably unchanged. However, if the take-up is located at the head, the traveling shockwave impacts and reflect off the tail pulley. This causes extremely low belt tensions that may result in material spillage along the tail end of the conveyor. Figure 7 shows the belt tensions during the first 2.5 seconds of a drift stop. During this time, the tail tensions drop from 55 kN to 3.5, which is unacceptable low.

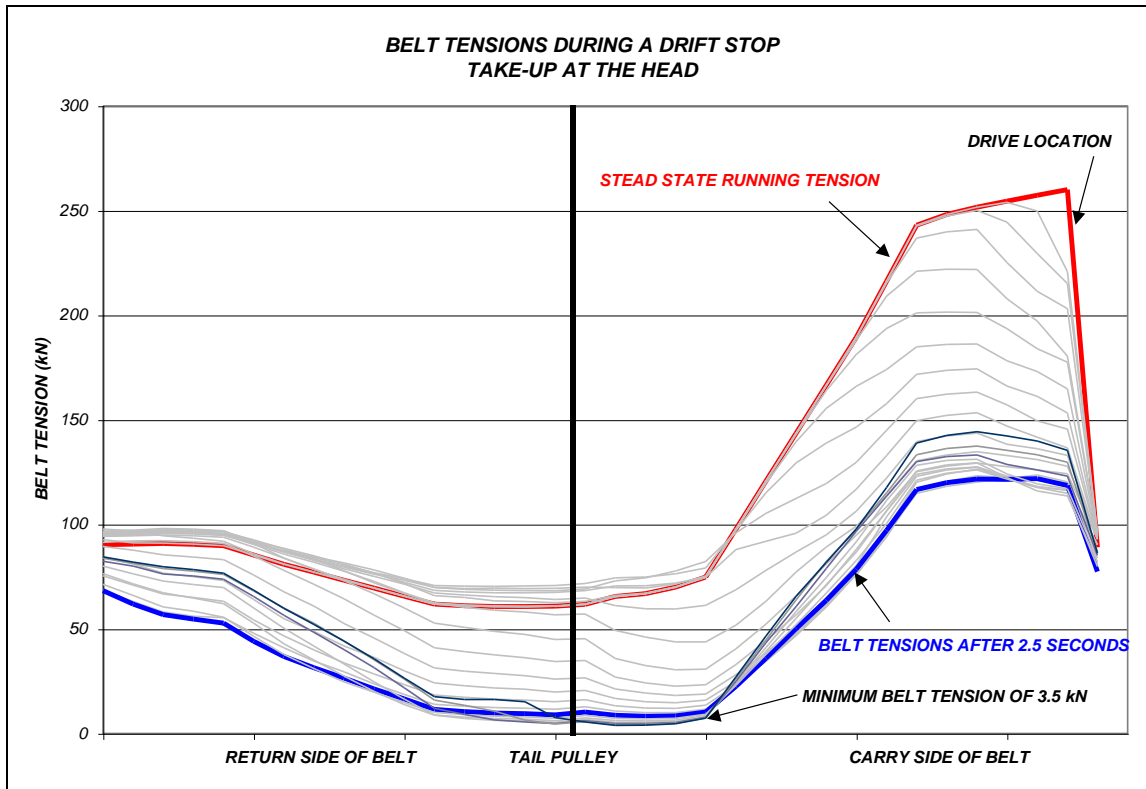


FIGURE 7 – Belt tension during a drift stop with a head take-up system

The dynamic tension wave must travel from the drive pulley down the carry strand and then back up the return strand to the take-up. Once the tension wave reaches the head take-up and the counterweight starts moving to equalize the tension, it is already too late for the tail pulley. The tensions at the tail will drop below acceptable levels before the head take-up can react. This dangerous situation can turn from bad to worse. The take-up will now start to pull the belt up the slope in response to the initial tension wave. This creates another tension wave in which the tensions at the tail go from unacceptably low to very high. This second tension wave may cause idler or pulley damage.

Table 1 shows the minimum belt tensions for static and dynamic calculations. Note that in the case of the head take-up, the minimum belt tension for a drift stop using static calculations is substantially different than calculated using the dynamics analysis. Had the engineer only used static calculations the resulting conveyor design could be disastrous, as described above.

	Static Calculations Minimum Drift Tension (kN)	Dynamic Calculations Minimum Drift Tension (kN)
Tail Take-up	50.6	20.4
Head Take-up	30.6	3.5
Head Take-up with Capstan	-	16.5

Table 1 – Static and dynamic minimum belt tensions during a drift stop

If the take-up must be placed at the head end (due to geometric or other restriction at the tail), this problem could be solved by further increasing the counterweight mass. This solution is used the majority of the time in current conveyor design. Unfortunately, this results in higher belt tensions, a stronger belt (or lower safety factor), larger structures, heavier duty pulleys, etc. In fact, for this simple example, the

take-up mass would need to be increased by over 30% (130kN) to achieve an acceptable design. Some conveyor systems may require an unacceptable high increase in take-up tension.

A second solution is to add flywheels at the head drive. In many cases this is an acceptable solution. However, depending on the size and inertia of the conveyor, the flywheels can quickly become very large. The addition of flywheels may also result in unacceptably long stopping times when the conveyor is empty, or only the head end is loaded.

A third solution is the installation of a capstan. In this case, the gravity take-up would function normally except during stopping. When the motor is turned off, the T1 and T2 tensions try and equilibrate themselves. The T1 tension will drop, and the T2 will increase. If the capstan is NOT applied then the increasing T2 tension will cause the take-up to move, and most of the tension increase will be absorbed by the gravity take-up system. If the capstan is applied, then the increasing T2 tensions will travel down the return side of the belt. Additionally, the speed at which this tension wave travels is faster for an empty belt than a loaded belt (the material on the belt dampens and slows the wave propagation speed). This allows the return side tension wave to reach the tail section before the collapsing T1 shockwave arrives. Figure 8 shows the belt tensions using a 40 kN capstan.

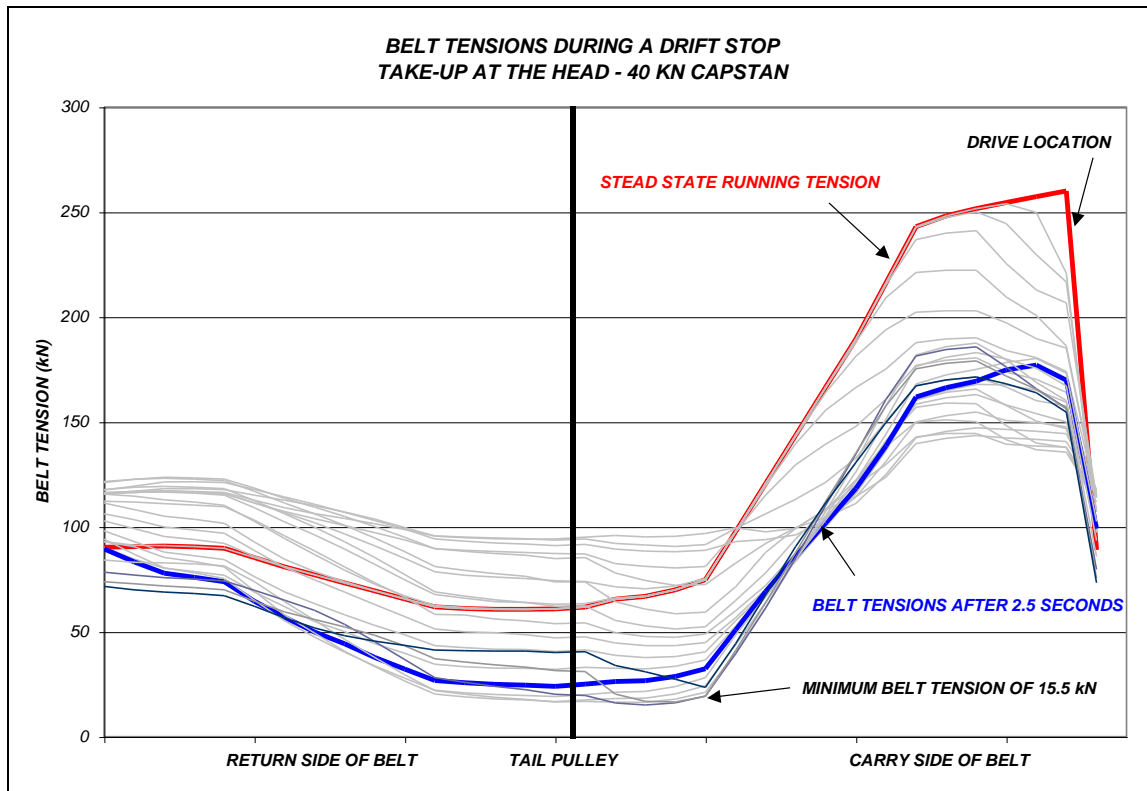


FIGURE 8 – Belt tensions for a head take-up system and capstan – Minimum belt tension of 15.5 kN

Even though this was a relatively simple example, it clearly demonstrated the need for performing a dynamic analysis. By properly sizing and installing a capstan the following benefits were achieved:

1. Reduced take-up mass
2. Reduce belt strength
3. Possible increase in carry side idler spacing (fewer idlers)

4. Reduced overall system tensions & structure requirements

Although some of these benefits may be minor for a small conveyor system, the basic capstan concept can be use on much larger conveyor systems resulting in significant cost benefits.

EXAMPLE 2– 6.1 KM OVERLAND COVNEYOR

This example demonstrates the successful installation of a capstan on a long overland conveyor, and also compares theoretical calculations to actual field measurements (Lodewijks & Kruse, 1998). The conveyor is located in Western Australia and transports coal 6.1 km from the coalfields to the Muja power plant. The general conveyor specifications are:

- ST-800 N/mm belting running at 4.5 m/s
- 2 x 315 kW primary and secondary head drives
- 14 concave curves & 15 convex curves
- 4500m horizontal curves
- 30 kN take-up belt line tension

The conveyor profile is shown in Figure 9 below.

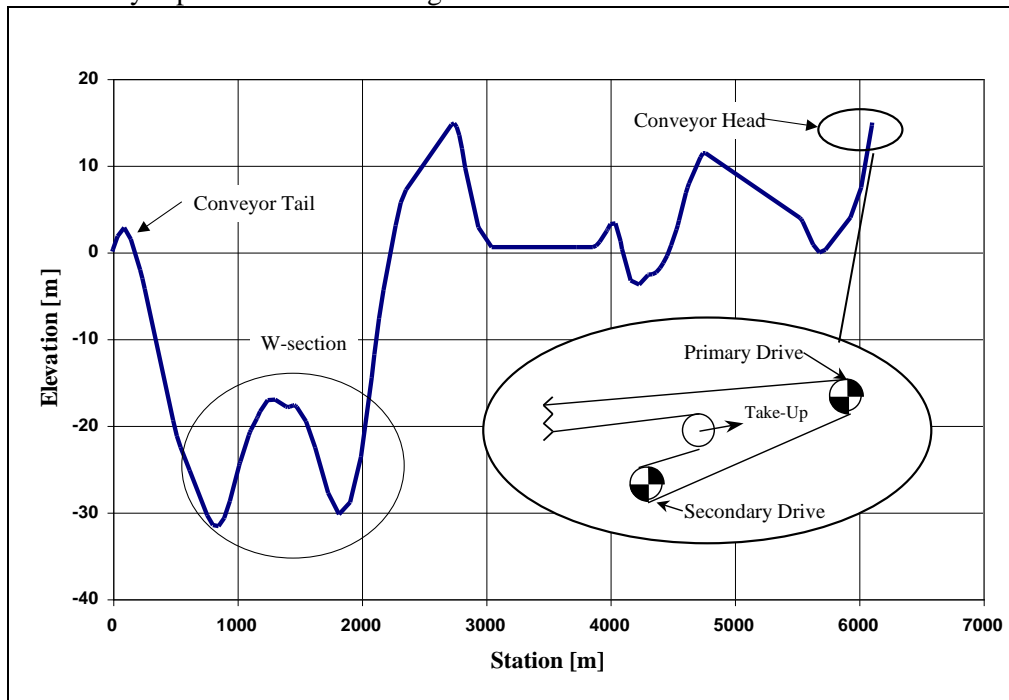


FIGURE 9 – 6.1 km overland conveyor profile and drive arrangement

The primary design concern with this system is the W-section located at 1,000 m and 2,000 m from the tail end of the conveyor. When the drives are turned off, the T1 belt tensions drop causing a tension shockwave from the head to the tail. When this tension wave arrives at the W-section, the belt tensions drop below acceptable design levels and excessive belt sag is occurs.

A solution to this problem would be to increase the overall take-up tension. Again this is a poor solution, as it would require a higher strength belt, and other structural components. Additionally, an increase in take-up tension would cause belt lift-off in some concave curve areas and an increase in the horizontal curve would be required. A better solution would be to install a brake that is applied during shutdown to prevent the low belt tensions from occurring. The ideal location (in terms of maintenance) for a brake would be at the head drive, after the take-up. This would allow all the equipment to be installed in a

single location. Unfortunately this is not possible since the collapsing tension wave from the head would reach the W-section before the brake tension wave. The brake was therefore installed at the tail.

Although the tail brake does prevent the low tension in the W-section, a new problem arises. During stopping, a low-tension wave on the return side of the conveyor causes the tail brake tension ratio to exceed 7.0:1. This tension ratio is not acceptable, and would cause the belt to slip on the tail pulley. In order to resolve this, a capstan was installed on the take-up. A 22.5 kN capstan was required to sufficiently increase the tail tension and keep the tail brake tension ratios within acceptable limits.

During commissioning it was discovered that the belt was slipping on the tail pulley during an emergency stop. Additionally there was some minor material spillage in the W-section of the belt. A load cell was installed at the take-up to measure the take-up tension. Additionally the tail braking torque, belt velocity, and tail pulley velocity were also measured. The results are shown in figures 10-12. Figure 10 shows the belt and the tail pulley velocity. The belt begins to slip on the tail pulley approximately 9 seconds after the stop is initiated. Figure 11 shows the brake torque during stopping. The measured brake torque was 10 kN-m below the design specification and therefore was not responsible for the belt slippage. However, the reduced brake torque would be responsible for the excessive belt sag in the W-section. Figure 12 shows the belt line tension at the take-up and pulley displacement. The take-up tension only increased by approximately 5 kN, instead of the 22.5 kN as designed. This would indeed cause lower than expected return side belt tensions and possible belt slippage at the tail pulley. Although the tail brake tension ratio could not be directly measured, it was determined using dynamic analysis. The resulting tension ratio is overlaid on figure 10. It shows that with the reduced capstan force the tail drive tension ratio could easily exceed 5.0:1. This is precisely when the belt began to slip. The design called for a maximum tension ratio of 3.5:1.

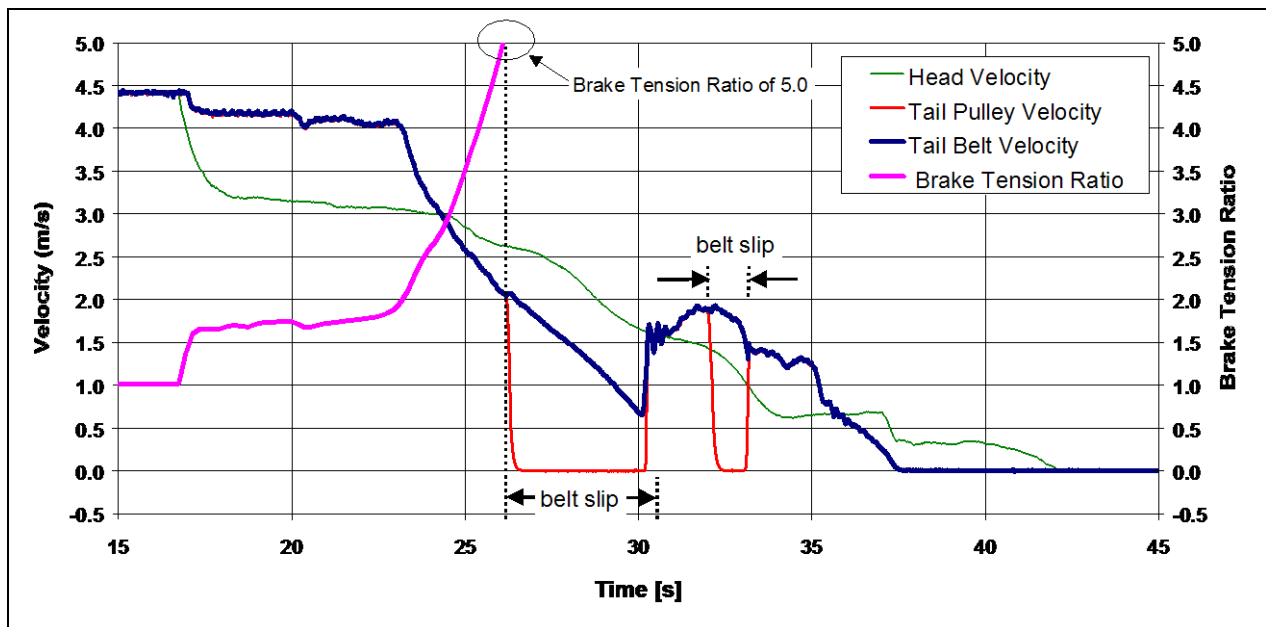


FIGURE 10 – Tail belt velocity, tail pulley velocity, and Tail brake tension ratio

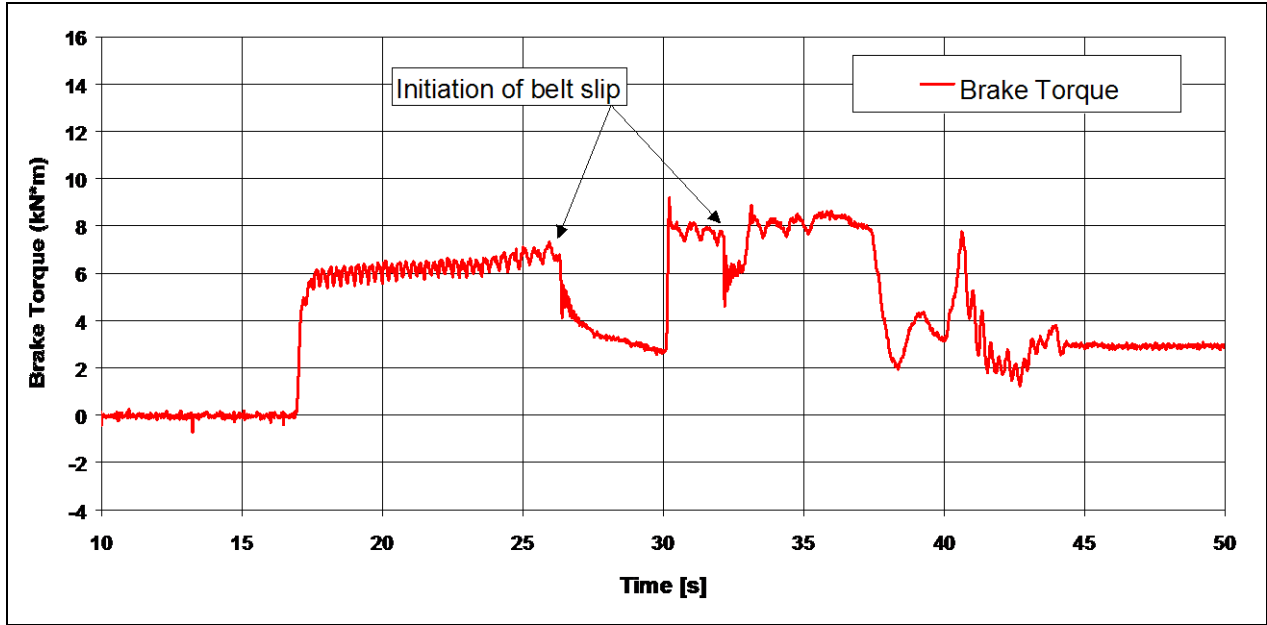


FIGURE 11 – Tail brake torque

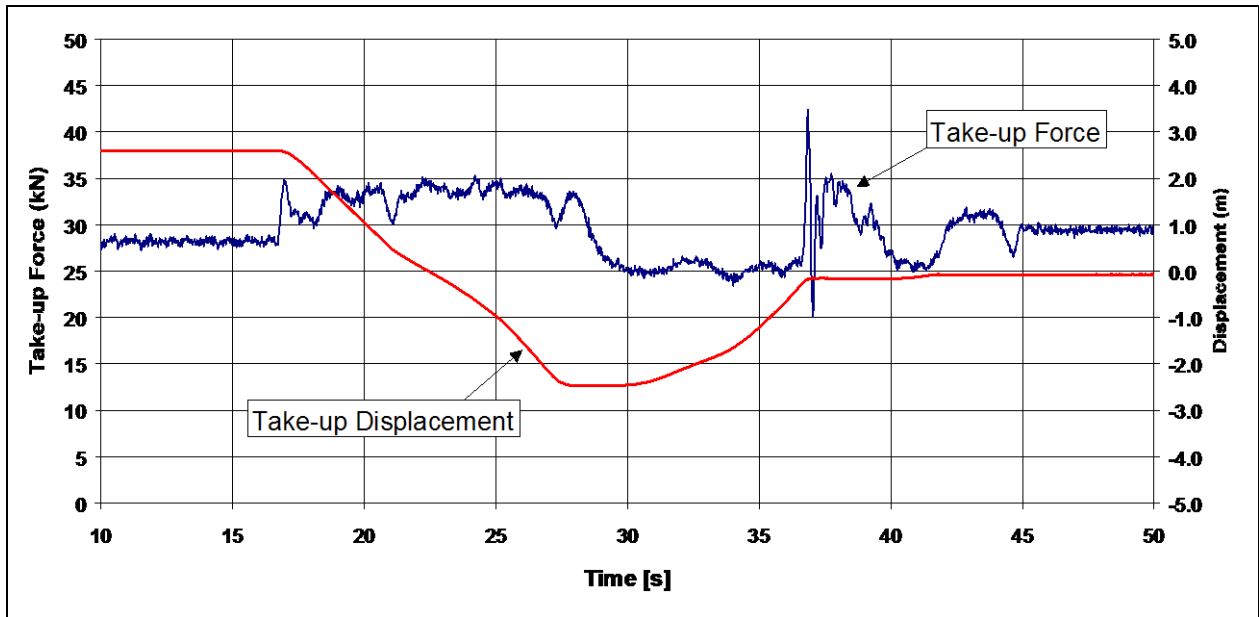


FIGURE 12 – Take-up force and displacement

Shortly after these measurements were taken both the capstan brake torque and the tail brake were increased to the correct design specifications. Once this was done both the material spillage and the belt slippage disappeared.

CONCLUSION

This paper has emphasized the need to perform dynamic analysis on both relatively short and long conveyor systems. It has also shown that in certain instances a properly designed capstan can significantly reduce the required counterweight mass and improve the dynamic behavior of the conveyor belt. This in turn results in considerable cost savings and provides a smoother and safer operating conveyor system.

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