Chapter 3

DYNAMIC ANALYSIS - BENEFITS FOR ALL CONVEYORS

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The use of Dynamic Analysis techniques in the design of conveyor systems is extremely limited. The most frequent use of the technique is in large and expensive systems where the emphasis is on insurance against failure caused by catastrophic tension waves. This paper sets out to illustrate a much wider range of applications where dynamic analysis has been successfully employed. The examples chosen deal with identification and correction of dynamic low tension problems, the variability in take-up movement as other system parameters change, the requirement for proper tuning of starting controls and finally the absolute necessity for a complete understanding of the entire system when specifying brakes for critical applications. The intent of the paper is to get designers to view a conveyor as a system of interacting flexible components rather than as a single rigid body.

INTRODUCTION

The idea of analyzing conveyors as flexible systems rather than as rigid bodies was first published over 25 years ago [2,6]. Reliable and versatile numerical techniques for performing this type of analysis have been around for about a decade [3]. Despite this long time frame and the significant increase in the utilization and capacity of conveyors during the same period, the use of Dynamic Analysis techniques have been largely confined to very large systems. Even for the conveyors where Dynamic Analysis has been carried out, it has usually been used as insurance against catastrophic tension waves rather than as a design tool to produce a better performing more cost effective conveyor. The intent of this paper is to illustrate the use of this modern analysis tool on smaller conveyor systems with an emphasis on producing a better design rather than just insuring against catastrophe. Of course, a distinct advantage of using Dynamic Analysis to optimize a design is that any destructive dynamics that may be inherent to the system will be detected.

OUTLINE OF THE ANALYSIS TECHNIQUE

Many papers have been published detailing the theoretical basis of Dynamic Analysis, [3,4]. For the purposes of this paper, which is to generally increase awareness in the industry of the use of these techniques, a simplified more hands-on explanation of the mechanics of the technique is useful.

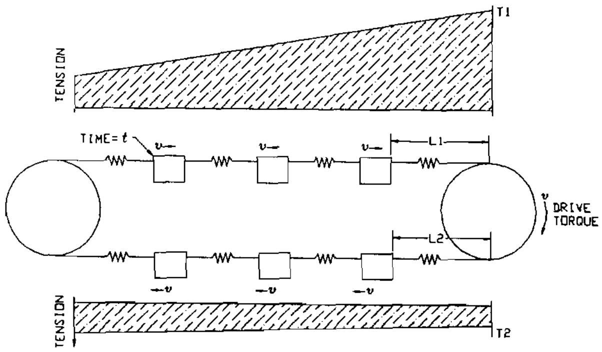

Figure 1(a) MASS-SPRING REPRESENTATION OF STEADY STATE RUNNING

Figure 1(a) shows a simple conveyor system during steady-state running, modelled as a series of masses and springs. In the steady-state condition, the belt velocity is everywhere close to a constant. In Figure 1(a), the spring extension before the drive, L1 and after the drive L2 remain constant. This is because, since the velocities are equal, in a time period δt, the rim of the drive pulley moves the same distance as the masses adjacent to it. This leaves the spring lengths unchanged.

In the steady-state condition the torque due to the effective belt tension on the pulley is matched by the torque produced by the drive.

The tension in the springs between the masses is determined by their stiffness and extension. Since during steady-state running, the distance between the masses, and hence the spring extension, remains constant, the tension in these springs also remains constant.

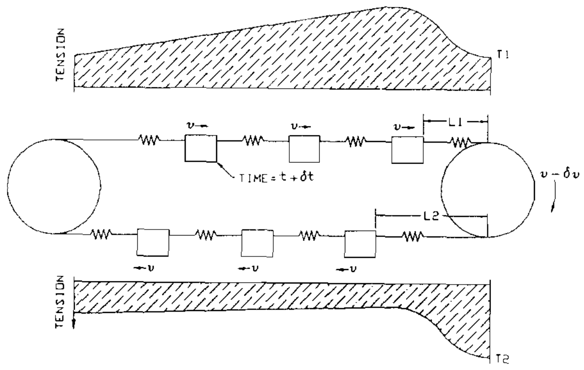

Figure 1(b) MASS SPRING REPRESENTATION OF COASTING STOP

Figure 1(b) shows the same system a short time after the drive has been turned off. The sequence of events that occur to produce the tension wave shown in Figure 1(b) are:

i) The removal of the drive torque from the pulley leaves the torque from the effective tension unbalanced. This results in a rapid deceleration of the drive pulley from v to (v-δv).

ii) The lower velocity of the drive pulley means that in the time period δt, the rim of the pulley moves a shorter distance than do the masses adjacent to it.

iii) The velocity difference results in the shortening of the spring upstream of the pulley and a lengthening of the spring downstream. The lengths of all other springs in the system remain unchanged.

iv) The change in extension of the springs adjacent to the drive pulley results in a decrease in tension between the drive and the mass upstream, and an increase in tension between the drive and the mass downstream.

v) The change in tension on only one side of the masses adjacent to the pulley, subsequently produces a force imbalance on these masses which causes them to decelerate.

vi) The deceleration of masses causes changes in the extension and hence tension of the springs on the other side of these masses. The resulting force imbalance causes the disturbance to propagate further along the conveyor.

The end result is that a wave of decreased tension propagates down the carry side of the conveyor and a wave of increased tension propagates down the return side. For simplicity the variations in tension will be referred to as 'compression' and 'tension' waves. These labels are not entirely accurate, since it is not possible to get true compression in a conveyor belt, but the terms are widely accepted and unambiguous.

If the magnitude of the 'compression' wave is greater than the actual steady-state tension of a region of the conveyor through which the wave passes, highly non-linear behavior will result. The belt tension in the region will not become negative but extremely low tensions and large belt sag between idlers will occur. Destructive dynamic effects frequently result from this type of occurrence.

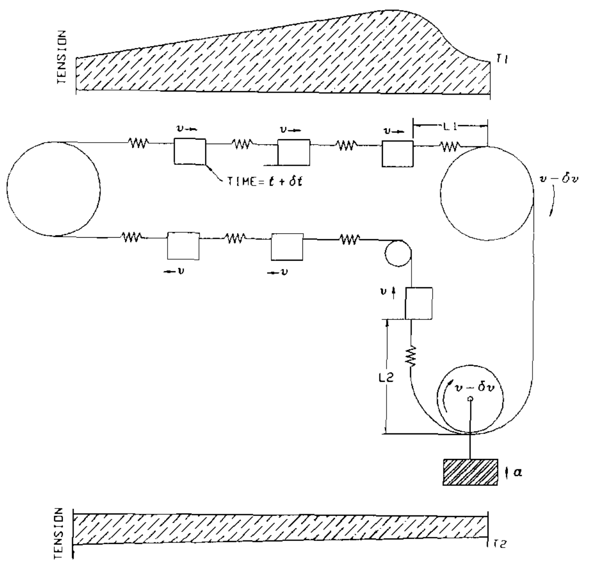

Figure 2 MASS SPRING REPRESENTATION OF COASTING STOP WITH GRAVITY TAKE-UP

Figure 2 is the same as Figure 1(b) except that a take-up is located immediately downstream of the drive. The increase in tension downstream of the pulley, instead of inducing a tension wave which propagates down the conveyor, produces a force imbalance on the take-up. The force imbalance on the take-up causes it to accelerate upwards. This upward movement of the take-up absorbs the 'tension' wave. As a result the return side of the belt is unaffected by the drive stopping until the 'compression' wave from the carry side has travelled completely around the conveyor.

The speed with which the initial waves propagate is a function of the system mass and the belt axial stiffness. The loaded side of the belt will be heavier and consequently waves on the carry side will propagate more slowly than waves on the return side.

The magnitude of the waves will depend on many factors. Two important factors are listed below.

i) The drive inertia. This is extremely important as it determines how quickly the pulley slows when the drive torque is removed. The faster the pulley slows, the greater the tension wave.

ii) The belt stiffness. This is important since for a given change in extension, a stiff belt will experience a greater tension change than a more flexible belt.

The steady-state velocity of the belt does not influence the magnitude of the stress wave since it is the change in velocity that is important, not the actual velocity.

APPLICATIONS FOR DYNAMIC ANALYSIS

Tracking the progress and interaction of the waves described in the previous section is what is commonly referred to as Dynamic Analysis. It is not the intention of this paper to dwell on the usefulness of these techniques for identifying destructive tension waves in conveyor systems. This application has been well accepted by the industry in general, and for most large projects Dynamic Analysis is carried out as insurance against catastrophe [5]. Rather it is this paper's intention to illustrate the ways in which Dynamic Analysis can be used as a design tool to enhance the performance and reliability of a much wider range of conveyors, particularly those in the lower power ranges. (ie <1000kW).

In some respects, the use of flexible dynamic analysis techniques has even greater advantages for low powered conveyors since these normally utilize fabric belts. The lower stiffness of a fabric belt accentuates the dynamic characteristics of the conveyor because of the slower propagation speed of the system and the greater elongational changes that occur.

To illustrate the usefulness of this type of analysis as a design tool, a number of case studies are presented. Some of the examples are taken from actual conveyor systems while others have been formulated to illustrate specific behavior.

Example 1. Low Tension and High Belt Sag

The conveyor used for this example carries coal over undulating terrain. It is approximately 2620m (8600ft) in length with a rise of 18m (60ft) from the tail to the head. The conveyor is powered at the head by a 300kW (400hp) motor on the primary drive and a 93kW (125hp) motor on the secondary drive. The take-up is located immediately downstream of the secondary drive pulley. The geometry of the belt is shown in Figure 3.

Since it was commissioned, this conveyor has experienced severe problems in the hollow near the tail (marked in Figure 3), whenever the conveyor stops fully loaded. Very large belt sag occurs in this region resulting in such problems as coal being dropped off the conveyor and the belt falling down onto the return strand and wrapping around idlers. An illustration of the sagging and buckled belt is shown in Figure 4. Starting the conveyor with the belt wrapped around the idlers has resulted in the destruction of many idlers and damage to the belt. Replacement of the fabric belt with a Kevlar, high modulus belt has only aggravated the problem.

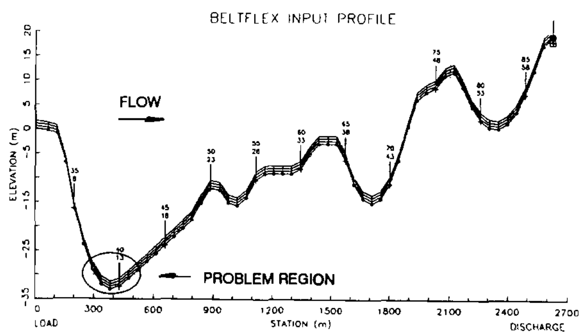

Figure 3 CONVEYOR GEOMETRY FOR EXAMPLE 1

In an attempt to control the dangerous sag in this region, a large number of additional idlers have been installed, reducing the idler spacing to less than half its original value. This modification has been of little if any benefit. In fact the problems continued even after there were more idlers in the hollow region than in the rest of the carry strand of the entire conveyor.

The problem had become so serious that after any loaded stop, a serviceman would be sent to inspect this region and, if necessary, to dig out any material that may be jammed between the belt and the idlers. The outcome of starting the jammed belt would be to destroy a number of idlers and possibly to tear the belt. Either of these situations could be extremely costly both in terms of actual cost and downtime.

A coal carrying conveyor in Northern British Columbia, suffering from exactly this problem, experienced a major failure soon after commissioning. The cost of torn belt and damaged idlers was over $US1 million and the lost production several million more. In that particular instance the actual failure had been predicted by the BELTFLEX dynamic analysis program but the contractors chose to ignore the advice.

To reach the problem section of this conveyor shown in Figure 4, the serviceman had to drive more than two miles over very rough roads. This meant that any loaded stop resulted in at least 1 hour of down time even if no corrective action was required.

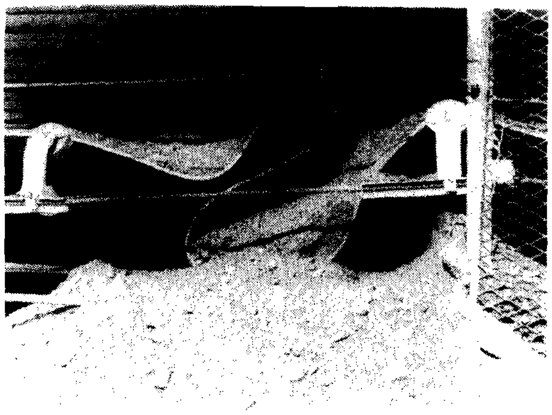

Figure 4 SEVERELY BUCKLED BELT

This problem, which is a result of the interaction between the drives, take-up and conveyor geometry, is an excellent example of the type of problem that can be easily identified at the design stage using Dynamic Analysis. Even though the problem this conveyor experienced was not a destructive tension wave, its effect on the operation of the system was very detrimental and had the potential for large capital cost if a significant length of belt was torn.

Figure 5 shows a Tension vs Time plot for the example conveyor in the hollow near the tail during a loaded stop. The two lines in the figure are belt tension before and after modification. The figure clearly shows that for the original design, very low tensions develop in the area of the hollow during a loaded stop. The minimum tension calculated to a large sag as shown in Figure 4.

Once a problem such as this has been identified, it is possible to model a number of alternate solutions. Actual simulation of a proposed solution makes it possible to install a trouble free system first time, or a successful modification on an existing conveyor. Modelling of different alternatives prior to installation is infinitely preferable to installing a series of modifications using a 'suck-it-and-see' approach in the hope that one of them will work.

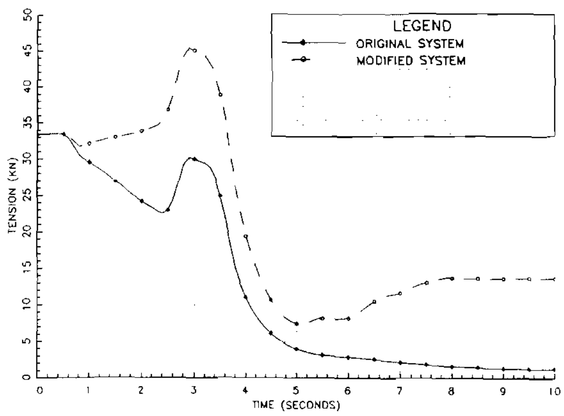

Figure 5 EX. No 1 CONVEYOR TENSION vs TIME

The recommended modification to correct the low tension problem in the hollow was to install a brake on the tail pulley. The second line in Figure 5 shows the tension predictions in the problem region after the installation of the brake. The new minimum predicted tension of 8kN corresponds to a sag of about 50mm (2in) and lasts for only a few seconds.

Dynamic modelling of the modified system also assists in determining the required characteristics of the tail brake such as its torque capacity and application rate. Any other technique to determine these characteristics is likely to result in either a more costly brake than necessary, a brake that will not do the job, or possibly a brake that will itself cause problems. Further explanation of the need for correct modelling of brake performance will be discussed later.

Example 2. Working Take-Up Displacement

The second example has been chosen to demonstrate the manner in which the dynamics of the conveyor effect the take-up, and to illustrate the way the take-up location can dramatically influence the amount of working take-up travel required.

In general, the total take-up travel allowed for at the design stage is influenced more by factors such as permanent belt stretch and allowance for set up variation and spare splices than by the movement of the take-up while starting and stopping. For situations where the take-up pulley moves horizontally on a carriage and the counter weight is located remotely in a tower, the movement allowance in the tower need only be sufficient for the working travel. This is because the position of the weight can be adjusted from time to time as required to allow for the other factors.

In either situation, more take-up movement than expected can be disastrous if either the weight or pulley should bottom out during a start or stop. Static calculations can give a good idea of the steady-state position of the take-up during normal running, and the variation in take-up position with changes in load. Static models cannot give any indication of how the take-up will behave during starting and stopping.

The example chosen for this section is a 16° incline conveyor, 425m (1400ft) long with a lift of 128m (420ft). For this conveyor, the take-up position when the belt is at rest or during steady-state running differs by only 400mm (16in). This is because the tension distribution is very similar during steady-state running and when stopped. Figure 6 shows the take-up movement during a loaded stop for three different conditions:

- Take-Up located at the tail
- Take-Up located at the head
- Take-up located at the head with the drive inertia doubled

The variance in take-up movement during the different stops is quite dramatic. For this conveyor, fixing all other parameters and relocating the take-up from the head to the tail more than halved the take-up travel.

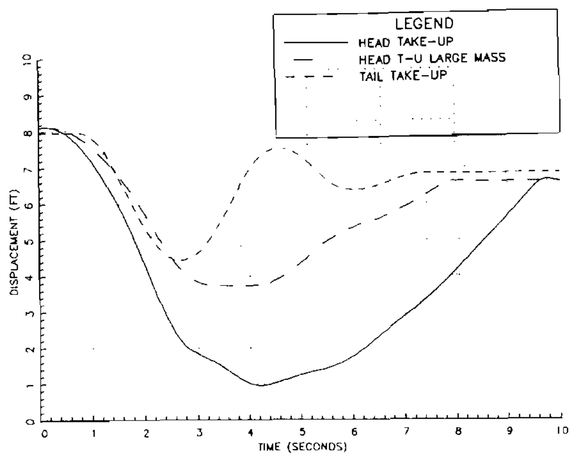

Figure 6 TAKE-UP DISPLACEMENT FOR DIFFERENT CONFIGURATIONS

Figure 6 also shows that leaving all parameters identical and simply increasing the inertia of the drive, usually by the addition of a flywheel, dramatically decreases the take-up movement.

In situations where the amount of take-up movement must be limited because of space constraints, such as is frequently the case in underground installations or where an existing facility is to be upgraded and the take-up tower is to be retained, the type of modelling described here is essential if the conveyor is to operate satisfactorily.

The graphs plotted in Figure 6 were all calculated using a Dynamic Analysis program since the conveyor in question has not yet been built. One of the attractive aspect of modelling take-up displacement is that it is one of the few parameters which can be easily measured. Parameters such as belt tension are of great interest but are extremely difficult to measure in the field. This makes take-up displacement an ideal parameter with which to check the accuracy of a numerical model. Figure 7 shows a comparison of actual measurements and the numerical predictions of take-up displacement for the conveyor described in Example 1. The close correlation both qualitatively and quantitatively is very encouraging and gives substantial justification for confidence in the modelling.

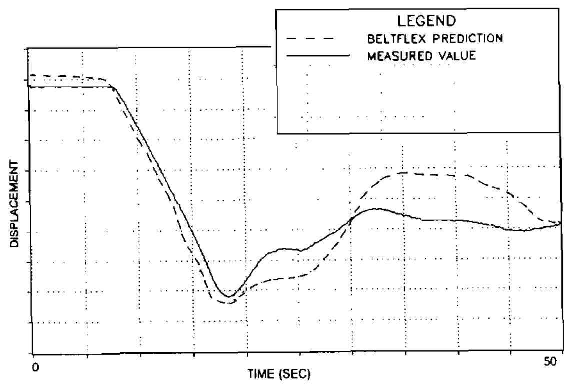

Figure 7 COMPARISON OF MEASUREMENT AND PREDICTION OF TAKE-UP DISPLACEMENT

Example 3 Effect of Control Tuning on System Operation

Selection of appropriate starting equipment and controls is essential for satisfactory operation of any conveyor system. Tuning of the starting controls for optimum performance is an exercise that is frequently overlooked despite the fact that poor tuning can cause as many problems as selection of inappropriate starting equipment. Tuning refers to the adjustment of parameters to optimize the performance of the system. These parameters include such things as timing of starts, synchronization between motors and brakes, fill levels and fill rates on fluid couplings, duration of starting and stopping ramps etc. In many situations, sophisticated starting controls are specified and installed but the tuning of the controls is so poor that the system gains no benefit from the additional capital expenditure.

A wide range of problems can arise from poor tuning of the controls. The result of the problems can range from overstressing of the system components which reduces their life, to catastrophic tension waves destroying the conveyor. Attention to detail and the use of a Dynamic Analysis program allows tuning of controls at the design stage. This leads to the elimination of destructive tension waves, minimization of starting loads and saves much site time and expense during commissioning.

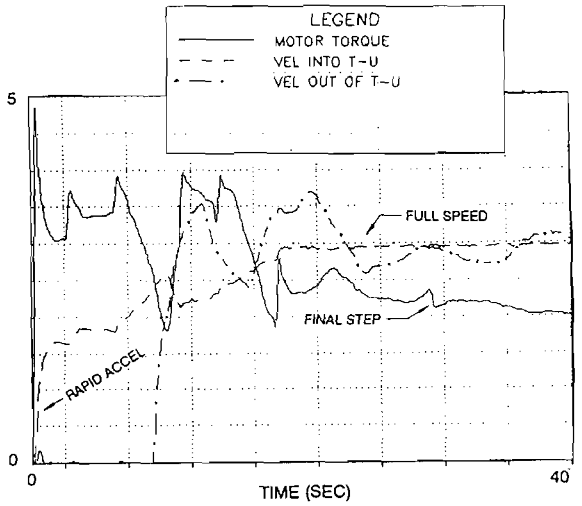

Figure 8(a) STARTING TORQUE AND VELOCITY OF A POORLY TUNED SYSTEM

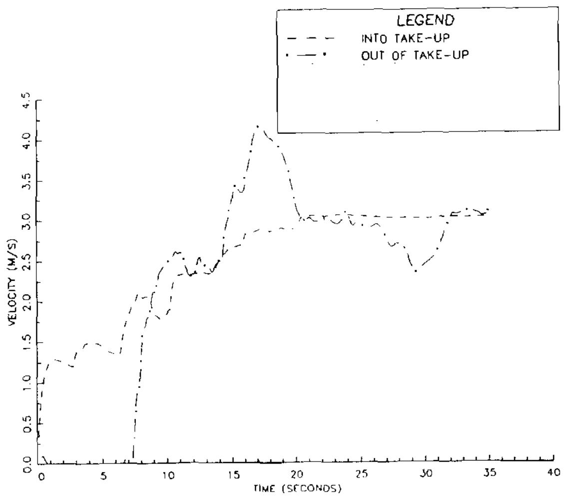

Figure 8(b) NUMERICAL PREDICTIONS OF VELOCITY FOR THE SYSTEM SHOWN IN FIGURE 8(a).

The examples shown here are from three different conveyors all utilizing stepped wound rotor starting controls. This type of starter controls torque by placing additional resistance in series with the rotor. At various times during the start the rotor resistance is changed by switching electrical contactors. The selection of the resistors and the times they are switched determines the starting characteristics of the system.

For each of the three cases described below, the choice of starting equipment is appropriate, but the tuning of the systems give very different starting behavior.

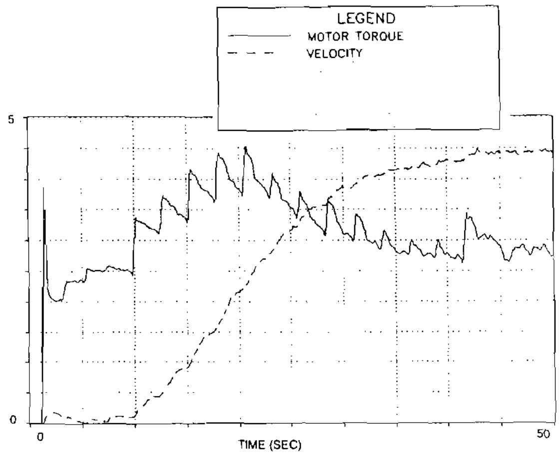

Figure 8(c) STARTING TORQUE AND VELOCITY FOR A REASONABLY TUNED SYSTEM

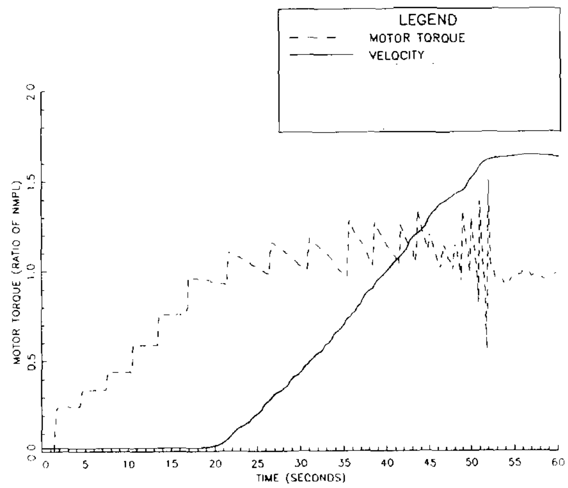

Figure 8(d) STARTING TORQUE AND VELOCITY FOR A WELL TUNED SYSTEM

Figure 8(a) shows measurements of motor torque, velocity into and out of the take-up for a conveyor where the timing and torque values of the resistor steps have been poorly selected. The loading on the first few steps is very high and the final stages of the start are really not required. The resistor steps occur over a period of 35 seconds and it is reasonable to assume that the designers intended the conveyor to start in this time. It is also reasonable to assume that stress calculations etc were based on uniform acceleration over 35 seconds. As can be seen from Figure 8(a), the belt is up to half speed in about 1 second up to full speed in about 20 seconds. The initial acceleration of the belt is almost 20 times the mean acceleration for a 35 second start.

An important consequence of the rapid acceleration is that the starting stresses in the conveyor will be substantially greater than allowed for by the designer. In this particular situation, there were no catastrophic repercussions from the poor tuning of the controls but this was purely good luck. Other systems have not been so fortunate.

Figure 8(b) shows the numerical predictions of the measured parameters shown in Figure 8(a). Again the correlation between the modelling and actual measurements is encouraging.

Figure 8(c) shows measurements of a large wound rotor motor starting. It would appear that in this case the designers have gone to some trouble to control the start and have been reasonably successful. Figure 8(c) is a good illustration of how well a starting sequence can be designed using standard static analysis techniques. The general form of this start is acceptable. The velocity curve is smooth with no gross discontinuities. However the starting sequence still produces peak torques about 25% higher than necessary.

To reduce the peak starting torque, a modified timing sequence must be developed that reduces the load on the early steps in the start and increases the load on the later steps. To develop such a sequence at the design stage is almost impossible using rigid body analysis techniques but is straight forward using Dynamic Analysis. The only other means of tuning these types of controls is on site during commissioning. This is time consuming and very expensive even if the appropriate people are available. These are the main reasons site tuning is rarely done satisfactorily.

Using Dynamic Analysis techniques it is straight forward to formulate an optimum starting sequence that will give good control under all load conditions and maintain the maximum belt tension at the lowest level possible. A starting sequence with these characteristics is shown in Figure 8(d). Note the very flat torque range in the later stages of the start and the uniform acceleration once the belt begins to move. The sharp spikes in the final seconds are rapid switching of resistors that allows the motors to run with very little residual slip (1.5%-2%). These spikes are essentially electrical and due to motor inertia do not show up at the belt as tension. The conveyor measured in Figure 8(c) does not show this behavior as the designers installed the system with 5% residual slip. Using more slip makes design of the system easier but exacts a considerable cost penalty in electricity over the life of the conveyor.

Example 4 Dynamic Tuning Of Critical Brake Applications

Many conveyor systems require brakes for safe operation or to assist in control during starting or stopping. The conveyor described in Example 1 used the brakes only to control the belt tension during stopping The brake was not necessary to reduce the stopping time of the conveyor. In cases such as these where no mass/time criteria is available to determine the required brake torque, a Dynamic Analysis is essential to develop a selection criterion for the brake.

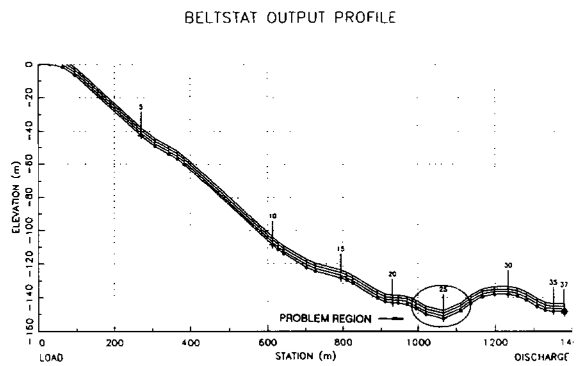

Figure 9 CONVEYOR GEOMETRY FOR EXAMPLE 4

In many slope belt applications, additional drive inertia is required to control dynamic tensions when stopping with a full load on the belt. This additional drive inertia frequently has the unwanted effect of substantially increasing the coasting time when the belt is empty. For safety reasons and to prevent tearing long sections of the belt if a rip occurs when the conveyor is empty, brakes are usually installed. The torque capacity of this type of brake is easy to calculate but the application strategy is not. For instance, if the belt is empty, full braking is required but if the belt is fully loaded it is essential that no brake is applied. A fully proportional brake using servo valves to control brake torque and fast response tachometers for closed loop control will be able to apply the correct brake torque to give uniform stopping time regardless of the belt loading, but the cost and complexity of such a system is often unnecessary. A much simpler braking system can often be made to work as long as the application timing and braking stages are correctly selected. Correct selection of the brake characteristics for all load situations is not possible without a proper understanding of the dynamics of the conveyor system.

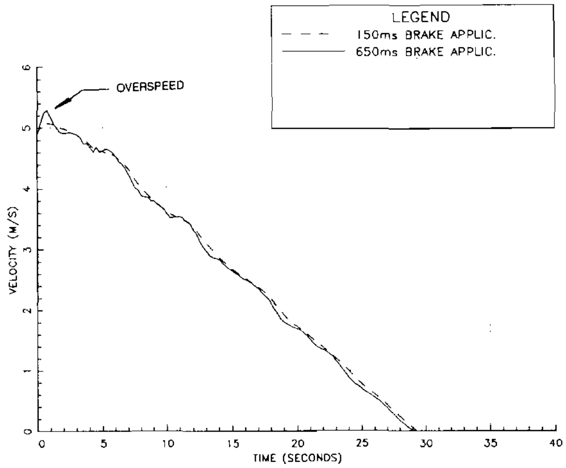

Figure 10(a) VELOCITY PLOTS FOR DIFFERENT BRAKE RESPONSE TIMES

The brake requirements for steep downhill systems are even more critical. Again, the torque requirements differ vastly depending on whether the belt is loaded or empty, but a downhill conveyor has the additional complication that timing considerations are critical. If the brakes do not reach the necessary torque level sufficiently quickly, usually within a few tenths of a second for a steep system, the conveyor may overspeed enough to be uncontrollable or to cause destruction of the drive assembly. Any rapid overspeed introduces a tension wave into the conveyor [1]. The magnitude of this stress wave is

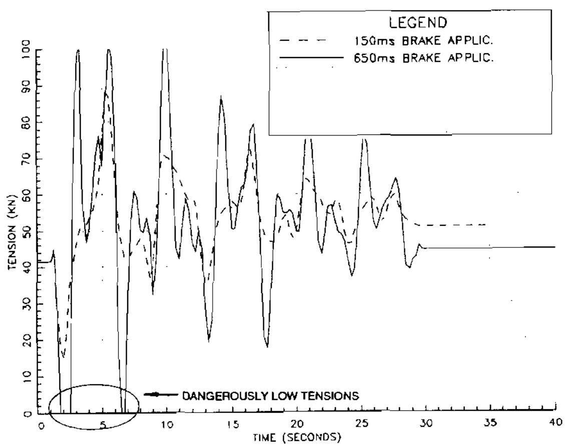

Figure 10(b) TENSION PLOTS FOR DIFFERENT BRAKE RESPONSE TIMES

dependent on the brake application speed, the steepness of the conveyor and the ratio of drive inertia to the total system inertia. Manipulation of the variable parameters to develop a safe and workable design is not possible without the use of Dynamic Analysis to give the designer an adequate understanding of the system.

The conveyor chosen as an example for this section is a short but steep downhill system shown in Figure 9. The conveyor is 1400m (4600ft) long, has a total fall from head to tail of 150m (490ft) with a dip of about 15m (50ft) just before the head pulley. The drives are located at the tail with a total installed power of 1050kW. The take-up is located at the head of the conveyor. The dip close to the head end introduced further complications in the control of this conveyor.

Figure 10(a) shows the velocity at the tail pulley with the conveyor stopping fully loaded for brake application times of 150ms and 650ms. The sharp overspeed for the longer application time is clearly apparent. This rapid acceleration induces a severe "compression" wave into the conveyor. Figure 10(b) shows the corresponding plots of tension in the dip near the head for both brake application times. The "compression" wave induced by the slow application of the brakes produces very low tensions in the dip with all the associated hazards as outlined in Example 1. It is also worthwhile noting that the peak tension is significantly greater for the slow application of the brake.

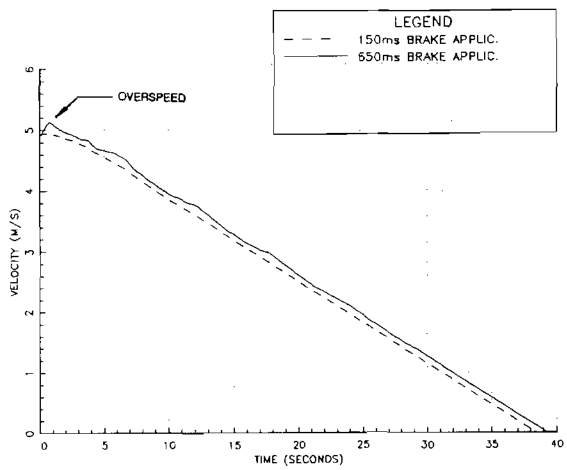

Figure 11(a) VELOCITY PLOTS FOR HIGH SYSTEM INERTIA BRAKING

Figure 11(a) and 11(b) are the same as the previous figures except that for these simulations the drive inertia has been substantially increased. With the increased inertia, the acceleration and overspeed of the tail pulley is less severe, resulting in a smaller magnitude "compression" wave being induced into the belt. Even for the slower brake application, the minimum tension is acceptable and the maximum belt tension is significantly lower than with the lower drive inertia.

Of course, for the same brake torque, if the system inertia is increased, the conveyor stopping time will increase. This is the reason Figure 10(a) shows a stopping of 30 seconds while Figure 11(a) shows 40 seconds.

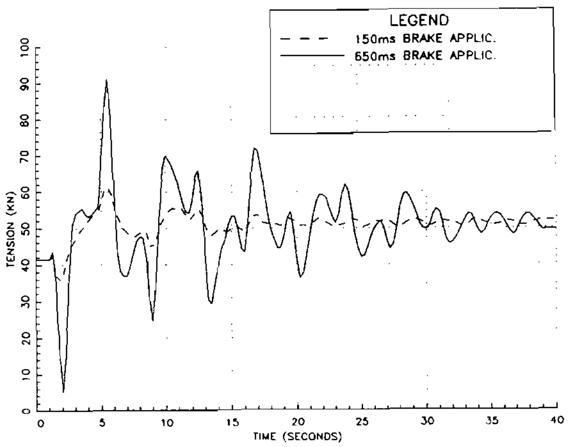

Figure 11(b) TENSION PLOTS FOR HIGH SYSTEM INERTIA BRAKING

From these figures the importance of the relationship between the application speed of the brake and the system inertia can be clearly seen. Greater inertia makes the brake response less critical in the control of the conveyor. It should also be clear that failure to consider these parameters when specifying a brake system can have disastrous results.

Tuning of the brake system for this conveyor was essential for the safety and viability of the system. Tuning of the starting controls was just as important for satisfactory operation. To start the section of belt in the dip near the head of the conveyor, the drives lower the tension in the bottom of the dip and the take-up pulls the belt up the small rise. In this situation, applying more drive torque will simply feed slack belt into the hollow and not assist at all in starting the conveyor. The rate at which the take-up can pull the loaded belt up the rise fixes the maximum acceleration rate that can be achieved. If this rate is exceeded dangerously large belt sag, like that illustrated in Example 1, are likely. Determining this maximum acceleration rate can only be done by a flexible analysis of the entire system.

CONCLUSION

This paper has presented a number of examples where Dynamic Analysis has been utilized to develop better functioning conveyor systems at the design stage or to eliminate problems after commissioning. The emphasis in these problems has not been on the identification or elimination of catastrophically destructive tension waves, but on illustrating a far wider range of dynamic problems that are frequently encountered in modern conveyor systems.

The only effective way of avoiding, or even being aware at the design stage, of problems such as those illustrated, is through the competent use of Dynamic Analysis.

Perhaps the greatest benefit that can be derived from the use of these analysis tools is the "feeling" an experienced conveyor designer can develop for a conveyor as a flexible arrangement of many interacting components. From this feel, the designer can from the very beginning of the design process, arrange the system so as to minimize or eliminate unwanted dynamic effects, thereby producing a more reliable and robust conveyor system.

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