THE GOLDEN RULES BELT CONVEYOR LOADINGS AND FORCES

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INTRODUCTION

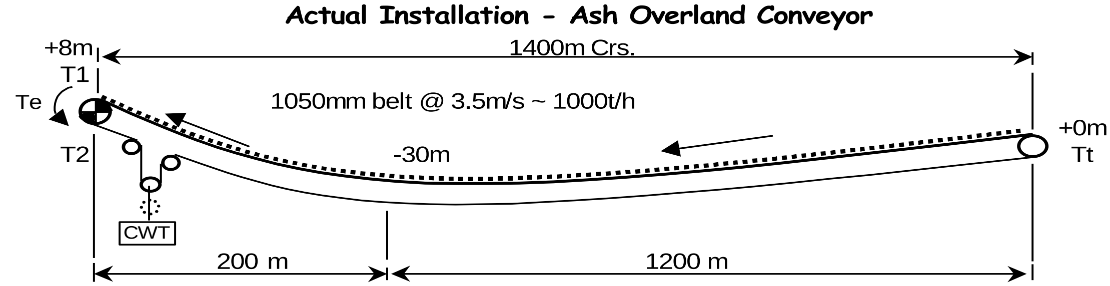
In the paper entitled " Loading and Forces brought about by belt conveyors (what Structural and Civil Engineers should be aware of)", which was presented by the author on 25th June 1996, it was shown that there is a real risk of loading misinformation emanating from conveyor design. It was shown that the normal design standards used by mechanical engineers are too superficial, and the likelihood of Civil and Structural Engineers receiving meaningful design criteria can be pretty slim. An actual overland conveyor installation, which had been designed using the conventional mechanical design standard ISO 5048, was analyzed together with more in-depth calculations for the dynamic conditions during starting and stopping. The huge discrepancy in results thus obtained demonstrated very clearly why the conveyor never really stood a chance, and had to be significantly re-built to the findings of LSL Consulting. Since then, the author has been called in to do a similar dynamic analysis on a fairly major but previously troublesome conveyor installation exporting bauxite in Guinea, West Africa. The behavior and improvements to this conveyor corresponded well to the findings of the calculations of the forces involved. A subsequent paper presented at this years loading conference, the author gave a brief summary of LSL's findings relating to the pitfalls of the original design of both conveyors, together with the criteria that should have been used. Since then, yet another dynamic analysis was carried out by the author, this time on a troublesome man-riding conveyor. This investigation gave some surprising and very significant and interesting results, which in turn are important to the structural designer as well.

The above actual case study examples, (all of which are for significant installations using steelcord belts), all demonstrate the importance of proper dynamic analysis by the mechanical engineer at the design stage. Also, the results of these case studies help define the type of standard loading information which must be made available to the Civil and Structural Engineers to avoid such pitfalls being encountered again. Finally, and perhaps of even more potential benefit, simple rule of thumb methods are given to obtain the maximum forces that Civil and Structural Engineers should use as a check on the suitability of the loading information they are given to work with.

EXAMPLE 1 : ASH OVERLAND CONVEYOR IN SOUTH AFRICA

The conveyor analysed in the above-mentioned previous paper on conveyor loadings as shown in the sketch below, was susceptible to dynamic problems as the longitudinal profile had a low point towards the head end. This has an extremely adverse affect on the coasting condition when fully loaded, as the short inclined section towards the head end tends to slow down far more quickly than the long decline section upstream. This caused the carrying belt at the lowest point to collapse during coasting causing massive spillage. It also caused the inadequate counterweight mass to travel upwards uncontrollably resulting in structural damage. Additional counterweight tension had to be added to overcome this as well as to provide the necessary tension to allow braking forces to be applied (especially with only the decline section loaded). The original counterweight tension of 33kN was calculated using the International conveyor standard calculation method ISO 5048 which normally applies to the “Static” or normal running condition only (and not the “Dynamic” conditions of starting and stopping). Due to the above conditions and problems in stopping the conveyor, the slack side tension had to be increased to 46kN. This has the effect that the tension distribution had to be increased all around the conveyor. In addition to this, the conveyor was analysed for start up conditions from fully loaded to completely empty. It was found that the original design had ignored what happens to the tail pulley when starting up empty. The amount of start up torque applied is pretty much the same for any load condition (unless an electrical constant start up time system is employed). As a result, when empty with a very short start up time, the tension at the tail required to accelerate the return belt increased dramatically. The tail station had to be re-built on new civils and an uprated tail pulley was installed. The drive station steelwork was found to be inadequate and was reinforced with additional bracing. For convenience, the sketch below shows the conveyor profile together with the dramatic range in loadings obtained from using the normal standard method to the actual loadings for the dynamic conditions. In addition, there is a possibility of a dynamic transient tension of about 23kN prorogating around the conveyor. This can be initiated by an aborted start where there is an interruption in start up torque with the belt stressed and stretched to its maximum. It can also be the result of the start up torque being applied too quickly before break away occurs. This is referred to as the Funke Ramp and is the case in our second example, which is dealt with in detail below.
From the first example we conclude the following important loading requirements:

Actual Installation - Ash Overland Conveyor



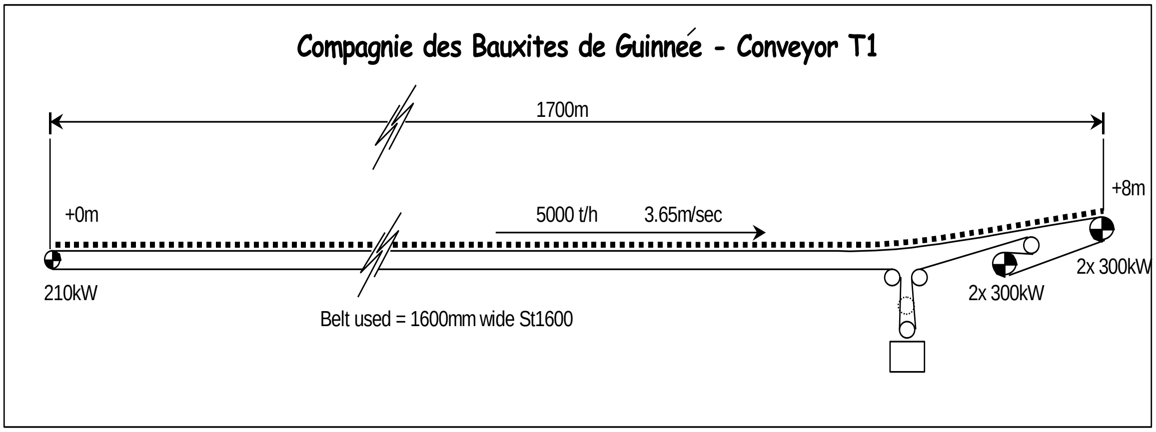
Summary of Calculated Tensions

	Te kN	T2 kN	T1 kN	Tt kN
Tensions based on ISO 5048 - fully loaded :	44	22	66	27
Tensions based on ISO 5048 - incline loaded :	47	23	69	30
Start up tensions using ISO 5048 :	66	33	99	38
Start up tensions using Dynamic Calculations :	66	33	99	43
Dynamic start up of empty belt :	66	33	99	62
Start tensions with addl. Cwt for coasting fully loaded :	66	44	110	73
Start tensions with addl, Cwt for braking -decline loaded :	66	46	112	75
Dynamic transient tension to be superimposed :	-	23	23	23

- The maximum slackside tension T2 is *twice* the normal value designed to ISO 5048 for the running (static) condition.
- The maximum tightside tension T1is *twice* the normal value designed to ISO 5048 for the running (static) condition.
- The maximum tail tension Tt is *three times* the normal value designed to ISO 5048 for the running (static) condition.
    - A dynamic transient tension equivalent to the originally calculated slackside T2 tension must also be added for the worstcase possible scenario. This tension is not however added in both belts leading to and returning from a pulley, as being a transient, can only act on one stand at a time.

EXAMPLE 2 : HARBOUR CONVEYOR T1 AT A PORT IN GUINEA

This conveyor installation is of utmost importance not only because it is the link between a bauxite treatment plant and the ships that transport the bauxite around the world, but also because the cost of waiting ships in the event of conveyor breakdown was horrendous. The start up time of the conveyor fully loaded to the present capacity was reported as being 12 to 15 seconds, and repeated problems of belt breakage on start up had occurred. Of interest was the class of steelcord belt being used as the safety factor is well in excess of normally calculated values to ISO 5048.



A full analysis was carried out and it was found that the critical aspects of the conveyor were:

- Application of breakaway torque was too rapid and well short of the requirements of the "Funke Ramp" (see below).
- Severe dynamic transients were set up due to the above.
- The high belt class made the transient problem worse.
- The conveyor was over powered.
- The start up drive sequence was incorrect.
- Due to the presence of the dynamic transient, the belt's safety factor when starting was too low.

The Funke Ramp relates to the time it takes to apply the start up torque, before the conveyor starts to move. This can be compared to the effect when you are being towed in a motor car when you have broken down. If the car towing you takes off too quickly, there is the inevitable jerk and jolting which often breaks the tow-rope. A gentle pull away with gradual application of towing force negates this. According to Funke, the application of breakaway torque must take at least five times the time it takes for a transient wave to reach the tail pulley. The rule of thumb here is that the time required by the Funke Ramp approximates to 3 seconds per 1000m of conveyor length.

What should be of interest to the structural engineer is that despite the above, and even though the "over-designed" belt actually failed in practice, the conveyor structure has survived without damage due to the overloading. This can in all probability be attributed to the high wind and corrosion allowance used in the design of the steelwork. It is surprising that after 20 years of operation there is no rust of any significance on the main steelwork, despite the conveyor's location on a jetty out to sea. The steelwork is painted and covered with a seemingly permanent layer of bauxite dust.

The following three figures explain the conveyors dynamic problems graphically and how they were solved. In figure A, the preferred start up curve is shown for starting the conveyor fully loaded. The conveyor is 1700m long therefore requiring a Funke Ramp time of 1700/1000 x 3 seconds to achieve breakaway torque of 4179 N-m. This equates to 5.1 seconds after which the torque applied increases uniformly to 160% Full Load Torque. The conveyor accelerates due to the extra 60% torque available until full speed is reached and the input torque reverts back to 100% to transport the full load.

Figure B represents the actual start up situation which caused all the problems. The drives were brought in stepwise with the delay between their starts varying from 0 to 0.5 seconds thus totally unsuitable for the Funke Ramp. The short measured start up time indicated that the starting torque was in excess of 200% of the Full Load Torque which in conjunction with the transient tension, has caused the belt to break when the weakest splice was in the most unfavourable position .

Figure C shows how the problem was solved first removing one of the secondary drives, and then by adjusting the maximum torque per drive in conjunction with the time delay between the drive starts. Although not ideal due to the somewhat outdated torque control system, the revised arrangement provided quite dramatic improvements to the start up and the belt is not expected to break again.

Figure A **Harbour conveyor T1: Ideal Start up Curve**

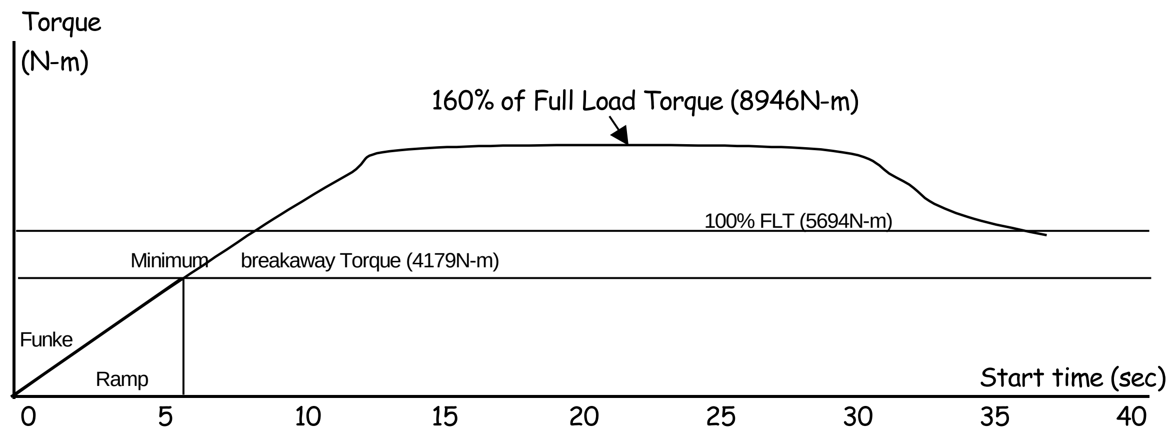


Figure B **Harbour conveyor T1: Actual start up curve**

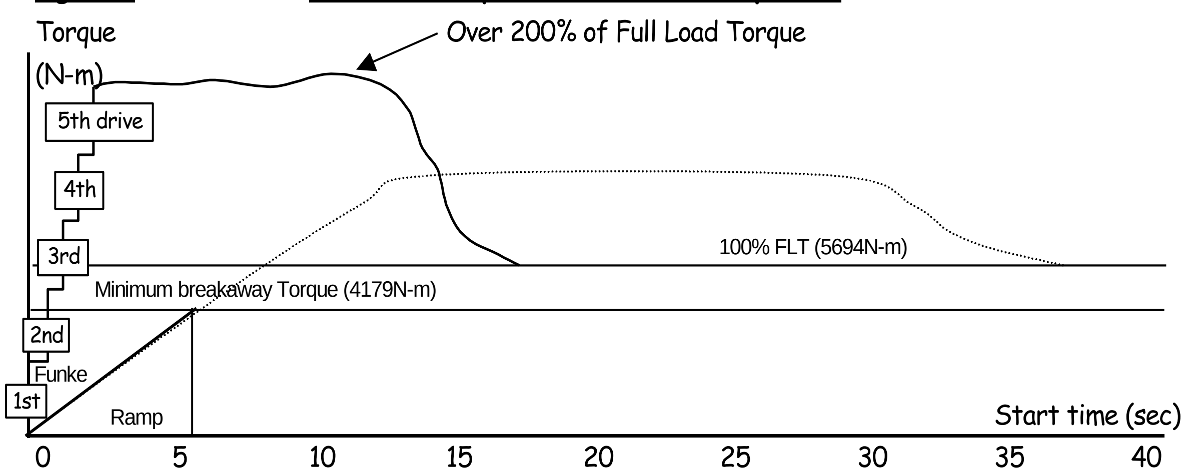
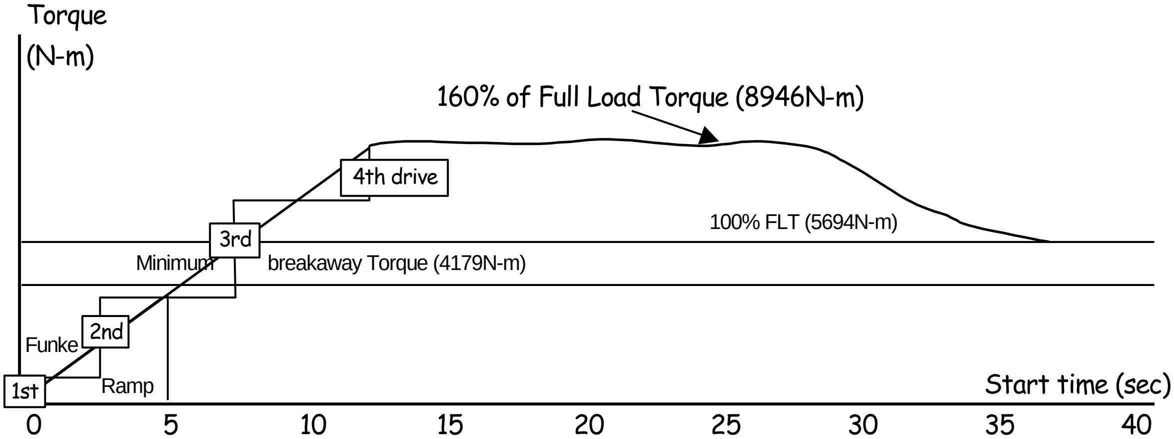


Figure C **Harbour conveyor T1: Revised start up curve**



From the above and more especially from the values calculated during the analysis of the conveyor's loadings the values of the belt's safety factor was derived for start up as well as for being subjected to the dynamic transient tension at the same time. This analysis is shown schematically below in figure D in graphical format. The graph is divided into three zones namely safe, unsafe and failure and shows the improvement achieved by reducing the start factor. At the previous start factor of over 200%, the start time was measured to be only 12 seconds and this implied that the calculated safety factor for the belt was 4.2. The safety factor for normal running of this belt is relatively high at 8.3 (which is well in excess of the usually taken value of 6.7) and this explains the start value of 4.2 being theoretically tolerable. The problem however, is the reduced safety factor after the dynamic transient is taken into account. This has the effect of reducing the safety factor to only 3. Dr Harrison states that when selecting belts, the minimum safety factor should be calculated from the following:

$$\text{Safety factor} = 3\,(1+Td/T1)$$

Where Td is the transient dynamic tension and T1 is the maximum running tension.

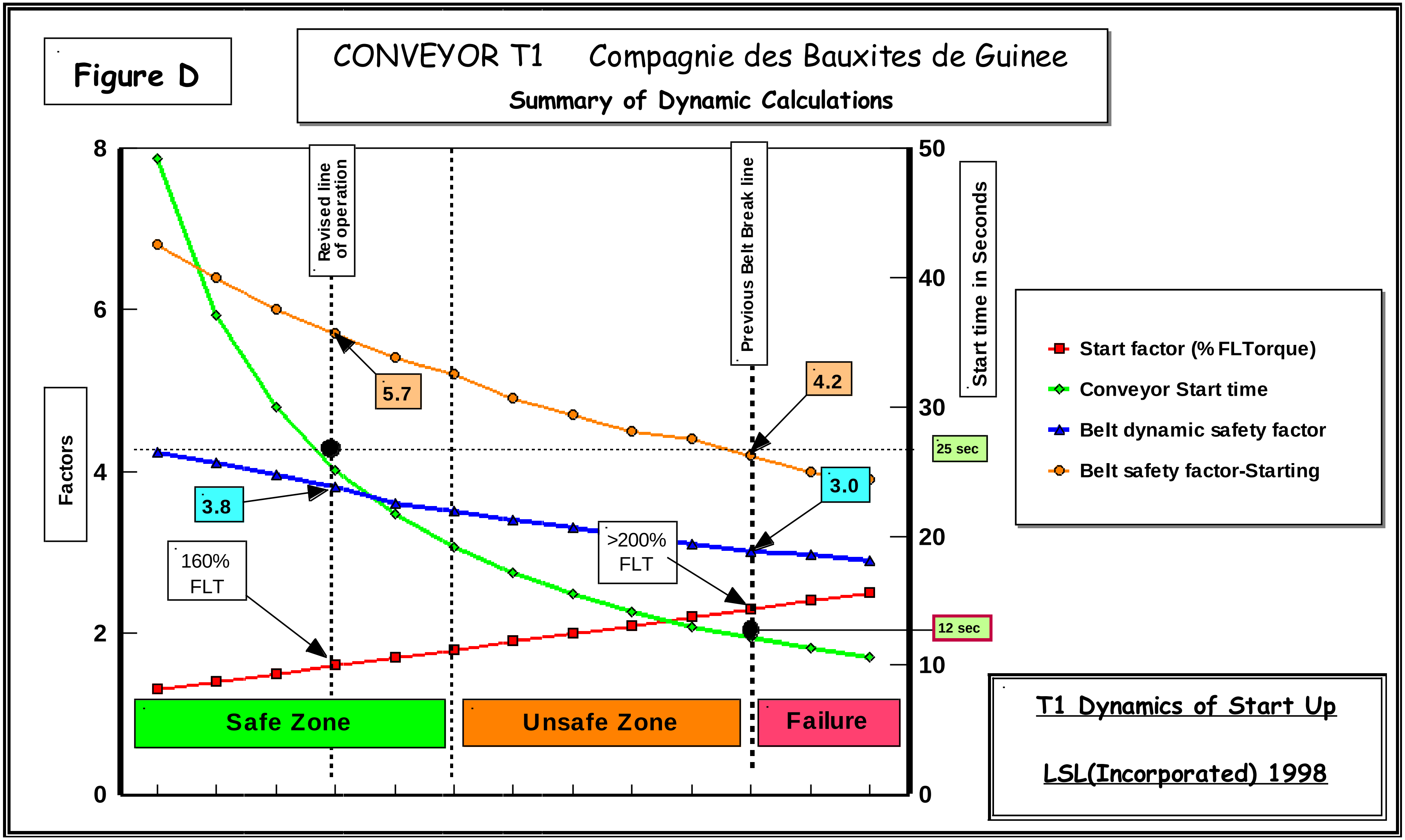


Figure D
CONVEYOR T1 Compagnie des Bauxites de Guinee
Summary of Dynamic Calculations
Factors
Start time in Seconds
Revised line of operation
Previous Belt Break line
5.7
4.2
3.8
3.0
25 sec
12 sec
160% FLT
>200% FLT
Safe Zone
Unsafe Zone
Failure
Start factor (% FLTorque)
Conveyor Start time
Belt dynamic safety factor
Belt safety factor-Starting
T1 Dynamics of Start Up
LSL(Incorporated) 1998

From this it can be seen that the minimum possible safety factor that can be used in practice is 3. The reason for this is that the deformation of the belt around the pulleys, edge stress through transitions and vertical curves, and for misalignments etc. demands a minimum safety factor, and the above example of conveyor T1 shows this very well.

From this, the loading criteria to which conveyor steelwork and civils should be designed becomes much clearer. The loadings for the high tension (drive) end of the conveyor must be based on calculated tensions as well as *based on the class of steelcord belt installed.* In other words, should the belt fail, the structure must not, and to ensure this the following is proposed:

$$\text{Maximum design T1 Tension} = \frac{\text{Belt class (kN/m) x belt width (m)}}{3}$$

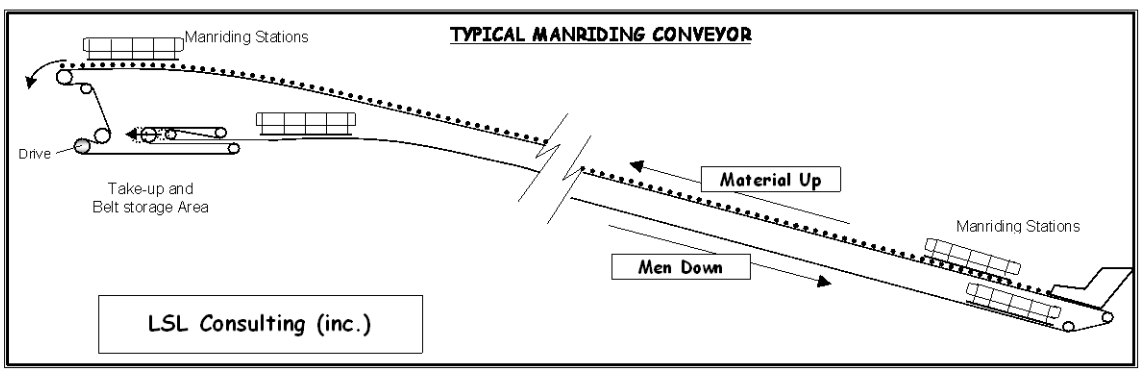
The start factor for conveyor T1 was reduced to 160% which had the effect of increasing the start up safety factor to 5.7 and 3.8 with the dynamic transient taken into the equation. This now means that the belt will no longer break (unless something goes wrong) and the structure will not be overstressed.

EXAMPLE 3: MANRIDING CONVEYOR IN A PLATINUM MINE

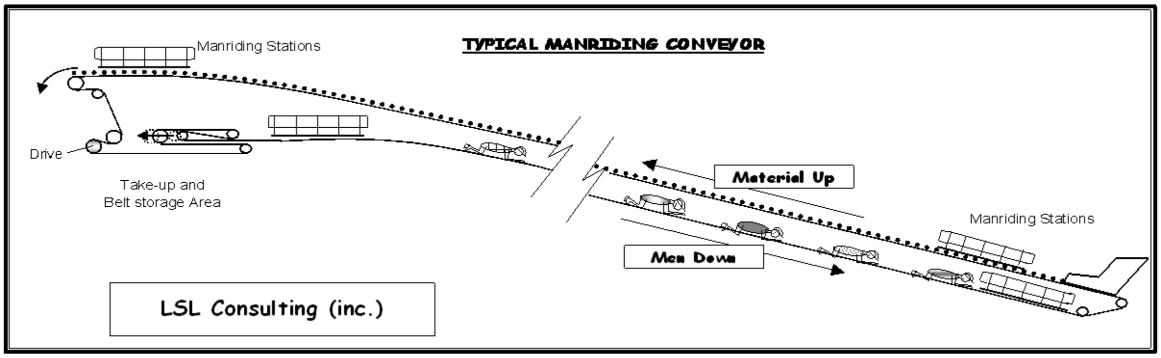
Manriding conveyors require a more stringent “set of rules” than short in-plant or overland conveyors, as they carry <u>people</u> as well as material on either strand, i.e. carry or return or both. They are normally found in mines and are inclined so as to transport workers in and out, as well as carrying crushed material out in the normal way. In order to operate safely, the maximum possible tensions that can be produced must be calculated for the following conditions:

- Fully loaded with material (carry side)
- Fully loaded with material as well as men (carry and return side)
- Fully loaded with men (return side)

In addition to the above, the conveyor has to be able to start and stop as well as trip out safely under any of the above load conditions. It is imperative that the belt never lifts off the idlers as this could crush a worker against either the conveyor structure or the hanging wall. This implies a possible preference for keeping the belt tensions low, but it is also imperative however, that the tensions never fall below the minimum allowable again with risk to human life. For this reason, it is very important to note the position of the take-up system. On the conveyor in this example, the designer chose to position the take-up at the **head end** for convenience. The cost of doing this was extremely high for reasons that only become apparent when you look at the effect each of the above load cases has on each other. A simple way to explain this is as follows:



- Select a slack side tension at the head drive end to enable the belt to start up with the greatest power requirement, i.e. fully loaded with material (carry side) as per the above sketch.
- Check that this tension is sufficiently adequate to prevent the belt collapsing at the tail when coasting to rest fully loaded with material as well as men (carry and return side). This condition is indicated in the sketch below and is dynamically awful. The material on the way up on the carry side wants to stop in a couple of seconds, whereas the people traveling downhill on the return belt cause the belt to want to cruise along indefinitely. The result is of course for the belt to want to collapse and buckle at the low tension tail end. A substantial head end take-up mass is therefore required to compensate for this.



- For the above case, fully loaded with material as well as men (carry and return side), when the conveyor is running, the tension is set at the head end by the counterweight. The tension in the return system reduces rapidly towards the tail by virtue of the masses of the people riding down the incline on the return belt. On the way back up, by virtue of the mass of material on the carry side, the tension increases markedly towards the head, thus giving a T1 or high tension at the head pulley.
- Now consider the other case, fully loaded with material (carry side). Again, when the conveyor is running, the tension is set at the head end by the counterweight. The tension in the return system reduces towards the tail by virtue of the mass of the belt alone. This reduction in tension towards the tail is considerably less than with people riding down the incline on the return belt. As a result, the tension at the tail when fully loaded with material only (carry side), is considerably higher. In turn therefore, the T1 or high tension at the head pulley is increased dramatically.
- On the man riding conveyor in this example, the designer attempted to counteract the above effect, brought about by putting the take-up at the head end, by fitting large flywheels to the conveyors drives. In so doing, the effect of the material on the way up on the carry side wanting to stop in a couple of seconds was counteracted.
- To compensate for the added inertia of the flywheels however, the brakes had to be increased. In turn, this caused the counterweight to misbehave and bottom out.

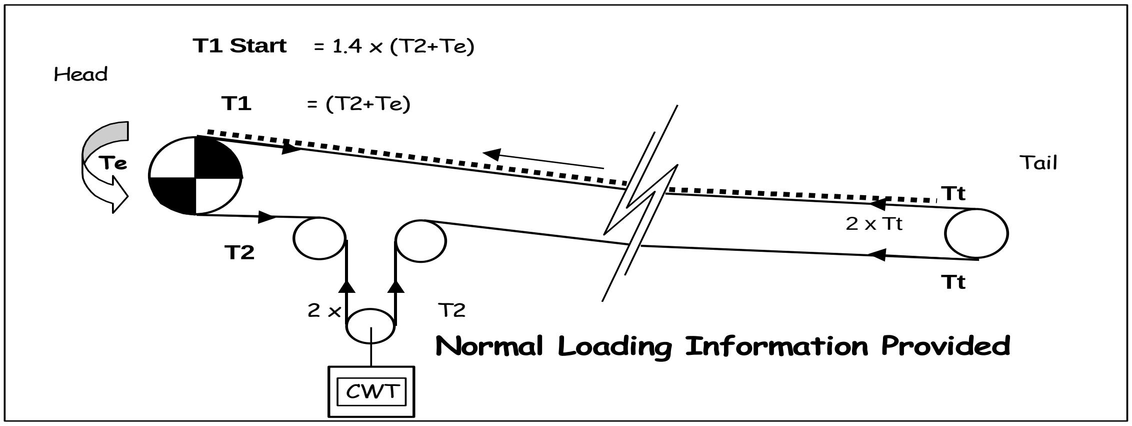
The overall effect of selecting a head take-up was to increase the tensions and lower the safety factor in the system by 24%. An St 2500 belt was used whereas an St2000 would have sufficed had a simple arrangement and **tail take-up** been employed. The number of pulleys in the conveyor would have been halved and the splices would have benefited *enormously* and the sophisticated control systems would not have been required. Also the conveyor would possibly not have been the subject of trouble-shooting by the author had they simply selected a tail take-up instead of one at the head end

The golden rule which manifests itself from the above example, is not to even attempt a structural design of a man-riding conveyor unless you are sure that the mechanical designer is experienced in this regard and has passed on meaningful design forces based on good dynamic analysis and based on a *good and practical conveyor arrangement.*

CONCLUSIONS AND SUMMARY OF GOLDEN RULES

Current Situation

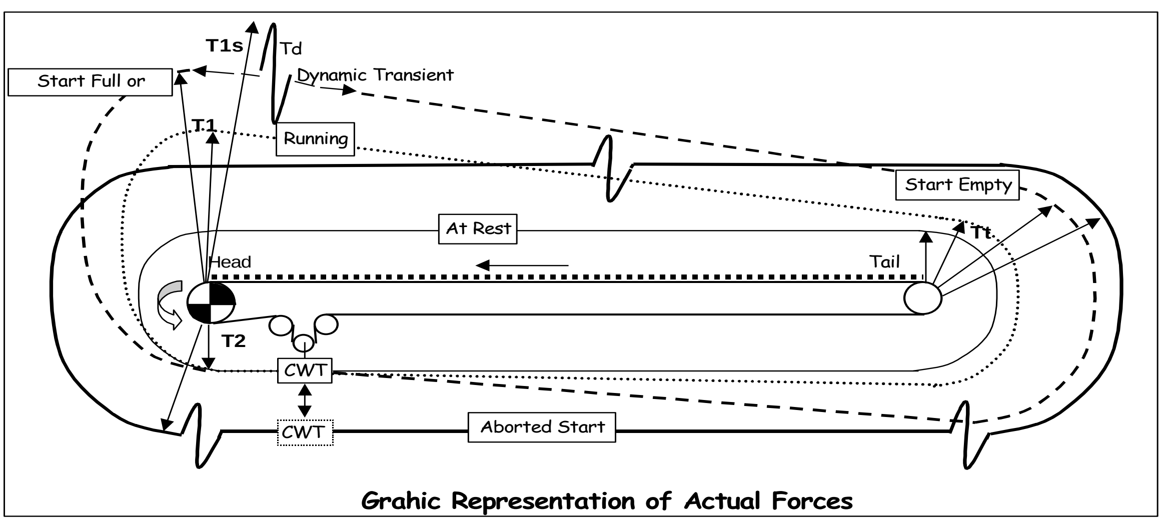
It is often the case that the mechanical engineer gives the Structural and Civil engineers a conveyor schematic diagram which he has marked up with loadings from tensions T1, T1 Start, T2 and Tt as follows:



The values given above are normally calculated using the ISO 5048 calculation method for “static” conditions, or in other words for when the conveyor has settled down at full speed and load. To cater for start up, it is common to increase T1 by a factor of 1.4. To this the Structural and Civil engineer will add the normal design load factors in accordance with SABS 0160.

Proposed Golden Rules

From the above examples as well as many others not mentioned here, it is concluded that for significant conveyors using steelcord belts, the following is recommended:



- The maximum slackside tension T2 is taken as *twice* the normal value designed to ISO 5048 for the running static condition.

- The maximum tail tension Tt is taken as *three times* the normal value designed to ISO 5048 for the running static condition.

- A dynamic transient tension taken as equivalent to the originally calculated slackside tension must be added for the worst case possible low tension area scenario. This tension is not however added in both belts leading to and returning from a pulley, as being a transient, can only act on one strand at a time when considering the force imposed on pulley frames.

- The maximum tight-side tension is taken as the greater of:

  Twice the normal value designed to ISO 5048 for the running static condition or,

  Calculated from Maximum Design Tension = $\dfrac{\text{Belt class (kN/m) x belt width (m)}}{3}$

It is proposed that in the absence of detailed mechanical analysis, the above rules of thumb be used in establishing the actual worst case loadings to ensure that conveyor structures are not under designed and do not fail due to the actual full range of conditions they have to endure. In the case of manriding conveyors, take no chances whatsoever and insist on proof of full and meaningful dynamic analysis for each and every load case.

Acknowledgements and references:

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Diallo M.T. For permission to print the findings of conveyor T1 at CBG in Guinnea.