BELT DE-TENSIONING IN DIPS

G. Lodewijks and Y. Pang

Delft University of Technology, The Netherlands

ABSTRACT

Essential for a reliable operational performance of a belt conveyor is its ability to maintain a minimum tension in the belt. This minimum belt tension ensures that the required drive power can be transmitted from the drive pulley onto the belt, and it restricts the belt sag to a certain limit. Normally a static design procedure is used to determine the minimum required belt tension. In practice however, belt de-tensioning can occur unexpectedly during operation, particularly in belt conveyors that have a geometrical profile which includes peaks and dips (or valleys). In this case, a static design approach alone does not suffice and a dynamic design approach is required to study the manifestation of low tension areas in the belt. This paper describes the dynamic nature of de-tensioning and the dynamic characteristics that contribute to the appearance of low tensions in dips. It provides guidelines and practical tips to prevent belt de-tensioning.

INTRODUCTION

In order to operate a belt conveyor in a reliable manner, it is essential that a minimum belt tension is maintained in the belt. There are two primary reasons for maintaining a minimum belt tension.

The first is to allow transfer of drive power from the drive train into the belt. The connection between the belt and the drive pulley is not form enclosed, as is the case in a chain and a sprocket wheel, but force enclosed. This means that the transfer of drive power from the drive train into the belt depends purely on the friction between the belt and the drive pulley and on the normal force that the belt exerts on the drive pulley surface. This normal force depends on the belt tension. Assume that T_1 is the belt tension before the drive pulley and T_2 the belt tension immediately after the drive pulley. To ensure that the drive power can be transferred into the belt, the ratio between T_1 and T_2 is limited to

$$\frac{T_1}{T_2} \leq e^{\mu\alpha} \qquad 1$$

where μ is the coefficient of friction between the drive pulley and the belt, and α the wrap angle of the belt on the drive pulley. If the ratio between T_1 and T_2 exceeds the limit given in Equation 1, then the belt will slip on the drive pulley leading to excessive belt wear.

The second major reason for maintaining a minimum belt tension is to ensure that the belt sag does not exceed certain limits. Normally, the belt sag s is limited to 1.5%

of the idler pitch when the belt is running fully loaded at a steady belt speed V or 3% during transient operation. If m'_b is the mass of the belt per metre and m'_l the mass of the bulk solid material on the belt per metre, then the minimum belt tension T to limit the belt sag s is prescribed as follows

$$T \geq \frac{(m'_b+m'_l)gS_i^2}{8s} \qquad 2$$

where S_i is the idler pitch. In Equation 2 the belt sag s is given in metres.

Besides the necessity to maintain a minimum belt tension as explained above, the belt tension should also be maximised, but only to within the applicable safety factors.

STEADY STATE BELT TENSION

Consider the horizontal belt conveyor shown in Figure 1. This non-moving belt conveyor has a typical layout with a head drive and a gravity take-up right after the drive pulley. The main reason for locating the take-up right after the drive pulley is to ensure that the belt tension after the drive pulley (the outgoing belt tension) is constant, minimising the chance of belt slip.

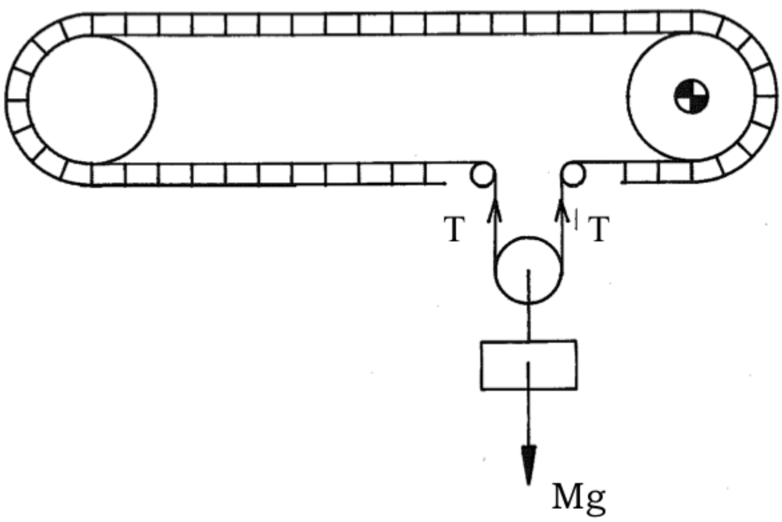

Figure 1. Typical tension distribution in a non-driven belt conveyor (Lodewijks, 2012)

If the drive is not used, then in theory, the belt tension should be distributed as shown in Figure 1. The belt is pre-tensioned by the gravity take-up. The gravity take-up consists of a dead weight with a mass M. The belt tension T therefore is equal to ½ Mg. If friction in the system is ignored for a moment, then this belt tension T will exist in the whole belt. Usually, the belt take-up tension T is symbolised by T_2.

If the drive is applied, then a drive force is exerted on the belt. The drive force F_d required to overcome all the friction in the belt conveyor is equal to using DIN 22101

$$F_d = CfgL(m'_r + (2m'_b + m'_l)cos\delta) + m'_l gH \qquad 3$$

Where C is the ratio between main and side resistances, f a fictive coefficient of friction, L the length of the belt conveyor, m'_r is the reduced mass of the carry and return idler rolls per metre belt length, δ the inclination or declination angle of the belt conveyor and H the elevation change between head and tail pulley. To distinguish between the resistances in the carry side F_C and return side F_R Equation 3 can be rewritten as

$$F_d = F_C + F_R \tag{4}$$

If only the carry side of the belt is loaded with bulk solid material, then F_C and F_R are given by

$$F_C = CfgL(m'_{rc} + (m'_b + m'_l)cos\delta) + m'_l gH \tag{5}$$

$$F_R = CfgL(m'_{rr} + m'_b cos\delta) \tag{6}$$

where m'_{rc} and m'_{rr} respectively are the reduced masses of the carry and return idler rolls per metre belt length. Therefore

$$m'_r = m'_{rc} + m'_{rr} \tag{7}$$

With the take-up tension T_2 and a required drive force F_d the maximum belt tension T_1 is equal to

$$T_1 = T_2 + F_d \tag{8}$$

The maximum belt tension T_1, in the example of the belt conveyor sketched in Figure 1, develops right before/on the drive pulley. This, and the belt tension development throughout the conveyor, is shown in Figure 2.

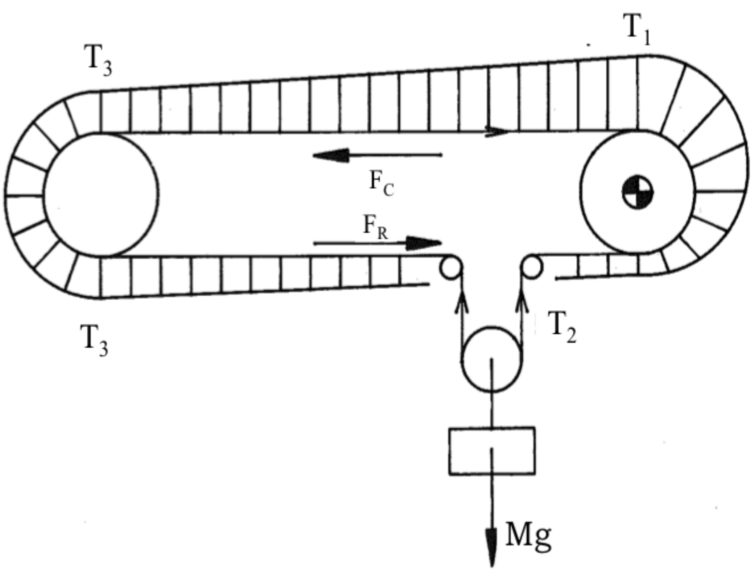

Figure 2. Typical tension distribution in a driven belt conveyor
(Lodewijks, 2012)

The belt tension T_3 at the tail pulley can easily be calculated from

$$T_3 = T_2 + F_R \tag{9}$$

or

$$T_3 = T_1 - F_C \tag{10}$$

Note that the belt tension around the drive pulley varies from T_1 to T_2 due to the drive force transmission, whereas the belt tension T_3 around the non-driven tail pulley is constant.

TRANSIENT BELT TENSION

During the transient operational phase of a belt conveyor, which is either during a start or a stop, the tensions in the belt deviate from the steady state belt tensions. The reason for this is that during the transient operational state, the belt's mass needs to be accelerated in the case of a start, or decelerated in the case of a stop. Acceleration requires energy whereas energy can be recovered during deceleration. The acceleration and deceleration forces can be calculated by using Newton's second law: force is mass times acceleration or deceleration.

In a belt conveyor two key masses exist. The first key conveyor mass M_C is the total mass of the rollers, the belt itself and the bulk solid material on the belt and is given by

$$M_c = L(m_r' + 2m_b' + m_l') \tag{11}$$

The second key mass is the mass that comes from the inertia of the drive train

$$M_D = \frac{I_D}{r_D^2} \tag{12}$$

where I_d is the inertia of the drive train on the low speed side and r_d is the radius of the drive pulley. The total mass M_T then is equal to

$$M_T = M_C + M_D \tag{13}$$

The discussion of the development of transient belt forces begins with an investigation of a start-up of a belt conveyor. Assume that the belt must accelerate from 0 m/s to V m/s. In this example, the speed difference dV is equal to V. Suppose that the belt is accelerated in a time interval dt. With dV and dt the acceleration a of the belt can be calculated by

$$a(t) = \frac{dV}{dt} \tag{14}$$

Note that a *(t)* can be a function of time. If the belt speed is linearly increased from 0 to V in time interval dt, then the acceleration during the start-up and the acceleration force is constant. This however, leads to two "jerks" in the belt tension—right at the beginning of the start-up and upon reaching V after time interval dt. In order to prevent these "jerks", alternative start-up profiles, that specify the development of V over the time interval dt, have been proposed. An appropriate start-up profile has been proposed by Harrison (Harrison, 1983). The additional force F_T that is exerted on the belt during transient operation, in this case a start-up, using Newton's second law is then equal to

$$F_T = M_T a(t) \qquad 15$$

The force F_T leads to an additional belt tension ΔT_1. With F_T being given in Equation 15 and using Equation 8, the total maximum belt tension at the head pulley during start-up is given by

$$T_1^* = T_1 + \Delta T_1 = T_2 + F_d + F_T \qquad 16$$

There are three important aspects to consider.

The first aspect is that, looking back at Figure 2, the belt tension just before/on the drive pulley increases with ΔT_1 from T_1 to T_1* (Equation 16). The belt tension on other locations of the conveyor, for example at the tail pulley, also increases but less than ΔT_1. This is caused by the fact that the mass of the belt and its components is distributed along the conveyor and not all located in one spot like a flywheel. Leaving the mass of the gravity take-up out for a moment, the increase of the belt tension during start-up then varies from T_1* right at the drive pulley, to T_2 at the take-up pulley or just after the drive pulley. If the mass of the take-up is taken into account then the effect of the acceleration of that mass on T_2 should be taken into account.

The second aspect is that during a start-up, the belt tension around the conveyor is not immediately increased all around the conveyor. This is due to the dynamics of the belt conveyor. Since a conveyor belt is not a flywheel but an elastic band, it also behaves like an elastic body (Lodewijks, 1996). That means that it takes time for a tension wave, which is initiated by the increase of T_1 at the drive pulley, to travel around the conveyor. The wave speed c_1, or the velocity at which an increase or decrease of belt tension travels through an empty belt is given by Lodewijks. (Lodewijks, 1996)

$$c_1 = \sqrt{\frac{E}{\rho}} = \sqrt{\frac{EA}{m_b'}} \qquad 17$$

where E is the belt's Young's modulus, ρ the belt's density and A the belt's cross-sectional area. Typically c_1 varies around 750 m/s to 1 000 m/s for a fabric belt and around 1 250 m/s to 2 250 m/s for a steel cord belt.

The third and final aspect is that the acceleration of the belt conveyor can change over the start-up time interval, see also Equation 14. This implies that ΔT_1 also varies over time. Normally the maximum acceleration, and thus ΔT_1, is obtained halfway through the start-up sequence. This may have some consequences if an emergency stop is initiated at that point in time.

BELT DE-TENSIONING

In the previous paragraph the belt tensioning during start-up was discussed. In reality, two belt stops can be identified:

- *normal operational stop*; a normal operational stop is a stop where the belt is stopped in a planned manner. Planned in this context means that the belt tensions are controlled during the stop. The belt tensions can, for example, be controlled by using the drives to stop the conveyor.
- *emergency stop*; an emergency stop is a stop where the belt has to be stopped in a short period of time because an emergency condition occurs. During an emergency stop the drives cannot usually be used and other measures have to be taken to control the belt tensions.

Here, focus is on the emergency stop to describe the process of de-tensioning of the belt. Assume that a fully loaded belt is running at full speed. In that case the belt tensions are as shown in Figure 2. If an emergency stop occurs, then the drives stop supplying the drive force. In practice this means that the maximum belt tension T_1 tends to drop from $T_2 + F_d$ to T_2. This sudden decrease in belt tension travels as a tension wave with magnitude $-F_d$, in this case, therefore, a compression wave travels through the belt decreasing belt tension throughout the conveyor.

In theory, the tension at the head pulley may drop to T_2. At the drive pulley, with the gravity take-up in the vicinity, the belt tension will probably stay around T_2. In the carry strand however, that mechanism does not exist. Therefore, the further away that a belt section gets from the head towards the tail, the less controllable the belt tension is. This means that the level of the belt tension may actually drop below T_2. The belt tension may become less than T_2 since the magnitude of the compression wave is F_d. If the magnitude of F_d is substantially more than the magnitude of T_2 then the belt tension in the return strand may go to very low levels, sometimes becoming unacceptably low. One could argue that, in theory, the lowest tension might become T_2 minus F_d. That, however, would not happen in reality.

First, the hysteresis in the belt conveyor would ensure that the magnitude of the compression wave decreases when traveling on its way through the conveyor belt, thus limiting the problem. Secondly, the inertia of the belt conveyor slows down the decrease of the belt speed, thus delaying the propagation of the compression wave (Lodewijks, 1996). This is where flywheels can help to limit the problem. Another measure to counteract the tension drop is to apply a brake in the take-up system, also called a capstan. Whether or not the application of a capstan is successful depends on the length and geometry of the belt conveyor.

Transient Belt Tension in Dips

Figure 3 shows a typical belt conveyor with a valley or dip. In order to keep things simple only one dip is considered and the overall change of elevation is 0.

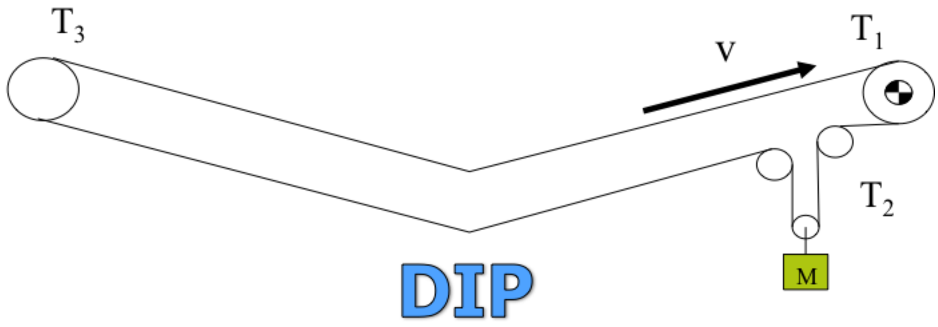

Figure 3. Typical layout of a belt conveyor with one dip

The belt tension in the dip is calculated as below. For the return strand the belt tension in the dip is

$$F_{dip,R} = T_2 + Cfg\frac{L}{2}\left(m'_{rr} + m'_b cos(-\delta)\right) \qquad 18$$

Note that the belt tension in the return side in the dip is independent of the magnitude of the dip (H in this case). This also means that the value of T_3 is independent of the magnitude of the dip. For the carry strand the belt tension in the dip is

$$F_{dip,C} = T_3 + Cfg\frac{L}{2}(m'_{rc} + (m'_b + m'_l)cos\delta) - m'_l gH \qquad 19$$

where H here is the depth of the dip. Note that the belt tension in the carry side of the dip decreases with an increase in depth of the dip.

In essence, the start-up and stop of this belt conveyor is identical to that described in the previous paragraphs. There is however, one important difference that makes a belt conveyor, especially one with a dip, responsible for the development of areas with very low tensions.

Consider a truck driving over a road in the mountains. If the truck drives downhill then one probably needs to brake in order to maintain a certain velocity. However, if one goes uphill, then extra power is required to maintain that velocity. This is because a truck driving downhill with mass W experiences an additional driving force -W.sinθ, and a truck driving uphill experiences an extra resistance force -W.sinθ. (Figure 4). For a belt conveyor, the same observation holds true with the important difference that the conveyor belt is not a rigid mass like a truck but an elastic body.

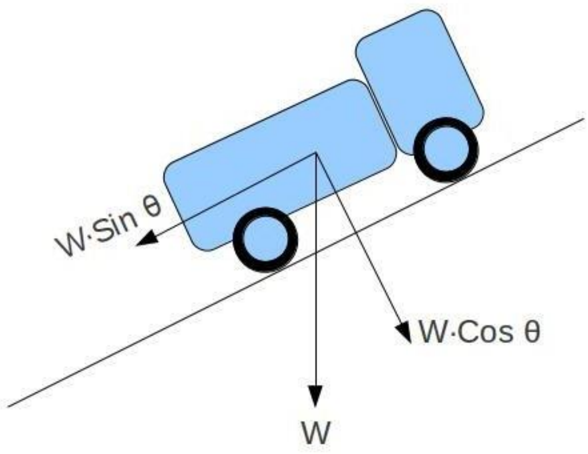

Figure 4. Truck going uphill experiencing backward force W.sinθ
(courtesy engerandu.wordpress.com)

Figure 5 shows a typical model of a belt section as used in finite element models of belt conveyors. The interaction with rolls has been left off for simplification (Lodewijks, 1996. Lodewijks, 2002). The springs and dashpot represent the viscoelastic behaviour of the belt and the masses M_i represent the mass of the belt and bulk solid material on the belt in section i of the belt conveyor. If the belt runs at steady-state velocity, then the deviation between the elements in terms of spring behaviour will be small, depending on the length of the belt section one finite element represents. However, when the belt experiences an emergency stop this situation changes significantly. In this case the belt shows some resemblance to the situation of a truck driving through a dip.

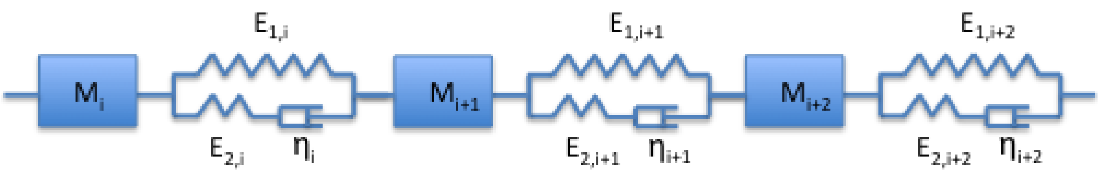

Figure 5. Typical representation of a part of the belt in a finite element model

The previous paragraph described the belt de-tensioning during an emergency stop. It was mentioned that the belt, particularly on the carry side, was responsible for the development of areas where the belt tension is low. When the layout includes a dip, this situation worsens. The reason for this is that the part of the belt going downhill experiences an extra (natural) driving force, like the truck, which results in a decrease of the belt tension. A decrease in belt tension will decrease the belt's elongation. The part of the belt going uphill needs an extra drive force to maintain the belt speed, again like the truck, which results in an increase in the belt tension. An increase in belt tension increases the elongation of the conveyor belt. Figure 6 shows the result of this using the finite element representation. When the effect of the de-tensioning during an emergency stop is combined with the earlier phenomenon described, it is easy to understand why having a dip in the belt conveyor layout increases the likelihood of low-tension areas.

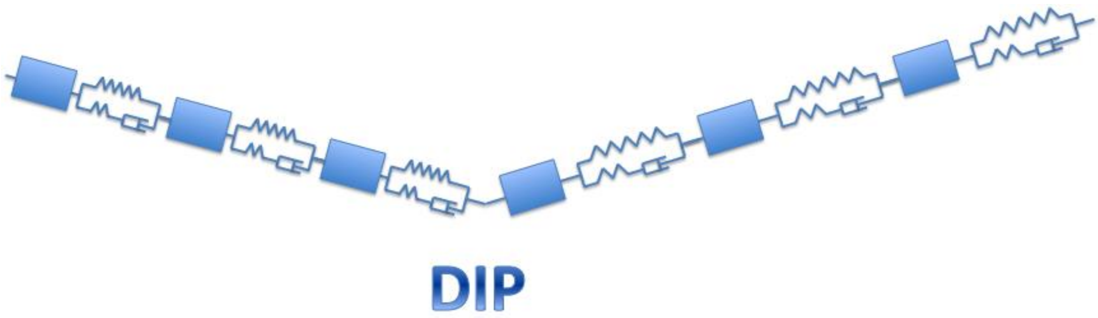

Figure 6. Typical compression in the downhill section of a belt in a dip

Summarising: a dip in a belt conveyor layout makes that area predisposed to the development of low-tension areas particularly during an emergency stop because:

— The belt tension decreases with an increase in the magnitude of the dip. (Equation 19).

— The compression wave that develops during an emergency stop decreases the tension considerably. The magnitude depends on the hysteresis in the conveyor and the location of the dip.

— The extra driving force in the downhill section of the belt caused by the weight of the belt and bulk solid material decreases the belt tension of the *elastic* belt going into the dip.

SOLUTIONS

Before looking at a practical example of the tension development in a dip of the layout of a belt conveyor, a list can be given with practical measures that can be taken to prevent the development of low-tension areas in a dip of the profile of a belt conveyor.

— Use the drives to slow down the belt conveyor and bring it to rest in a controlled manner. Of course this is only possible when the drives are available which normally is not the case in an emergency stop. By using the drives the development of a compression wave is prevented.

— Increase the belt pre-tension. If the belt's pre-tension T_2 is seriously increased then the belt tension in the dip may be acceptable. However, if this is the only measure taken then the pre-tension may have to be raised to unpractical levels. The impracticality comes from the required raise in belt rating, which would be quite expensive.

— Slow down the decrease in belt de-tensioning by the application of flywheels. Flywheels increase the inertia of the drive train that contributes substantially to the total inertia of the system. This increases the stopping time and decreases the propagation of the compression wave. The hysteresis further reduces the magnitude of the compression wave.

—Counteract the decrease of the belt tension by locally increasing the belt tension, for example by installing a brake that activates on initiation of an emergency stop. Typically, a brake for this purpose is located at the tail of the belt conveyor, but that sometimes leads to low tension regions in the return strand. Therefore an alternative is to locate the brake after the take-up. Be careful if a belt turnover is installed because braking over the belt turnover is not recommended. Another option is to install a capstan brake in the take-up. Whether or not the application of a capstan is successful depends on the length and geometry of the belt conveyor.

—Decrease the idler pitch in the dip. This does not help to prevent low belt tensions but it will limit the belt sag. This option is not used very often.

CASE STUDY

To gain a better understanding of the effects of the combination between a dip in the layout of a belt conveyor and the belt conveyor dynamics, the tension in the carry side of the belt in the dip was studied. The overall length of the belt conveyor under consideration is 1 km. The elevation H of the belt conveyor is 0 m in the basic arrangement. However, four other arrangements were also analysed. The first and basic arrangement was a fully horizontal conveyor.

In the second arrangement the belt conveyor has a decline with a length of 500 m towards a dip of -10 m, relevant to the level of the head and tail pulleys, and then an incline with a length of 500 m bringing the conveyor line back to the level of the head and tail drive. (Figure 3). In arrangements three, four and five, the dip is -20 m, -30 m and -40 m deep respectively. The conveyor has an 800 mm wide ST 800 belt and runs at 5.75 m/s. The carry idler pitch is 4.5 m and the return idler pitch is 1.5 m. The system is powered by a 315 kW drive and uses 157 kW when fully loaded. The system's power consumption is the same for all arrangements when the belt is fully loaded since the total overall elevation H is zero. The take-up tension is 25 kN and the take-up is located immediately after the head drive. The conveyor has a capacity of 1 000 tonnes of coal per hour.

The dynamic analysis considered is an emergency stop. For this conveyor the stopping time is around 24 seconds. Therefore the belt can drift to rest, it does not need a brake to stop on time. Also, for the basic arrangement, the belt tensions are sufficient to allow a drift stop without any further help.

Figure 7 shows the belt speed as a function of time on the location of the dip. The belt is fully loaded and drifts to rest in about 24 seconds. As can be seen in Figure 7, the drift time depends very little on the arrangement and the presence of a dip. The drift time is reduced from 24.2 seconds for a fully flat conveyor to 23.9 seconds for a conveyor with a dip of -40 m. This was to be expected. Since the inertia M_T and the total friction force, equal to the drive force F_D, remain the same for all arrangements, the stopping time should also remain the same. The only effect that decreases the drift time slightly is the fact that the uphill section of the belt tends to stop a bit faster than the downhill section, causing a slightly reduced stopping time in the dip.

It is interesting that the development of the belt speed shows no severe signs of dynamic effects. That is to be expected when the take-up tension is more than sufficient for power transfer and belt sag requirements. In addition, the inertia is distributed throughout the conveyor with no sites of substantial inertia. Finally, F_d for this conveyor is 27.3 kN, in the same range as the take-up tension of 25 kN.

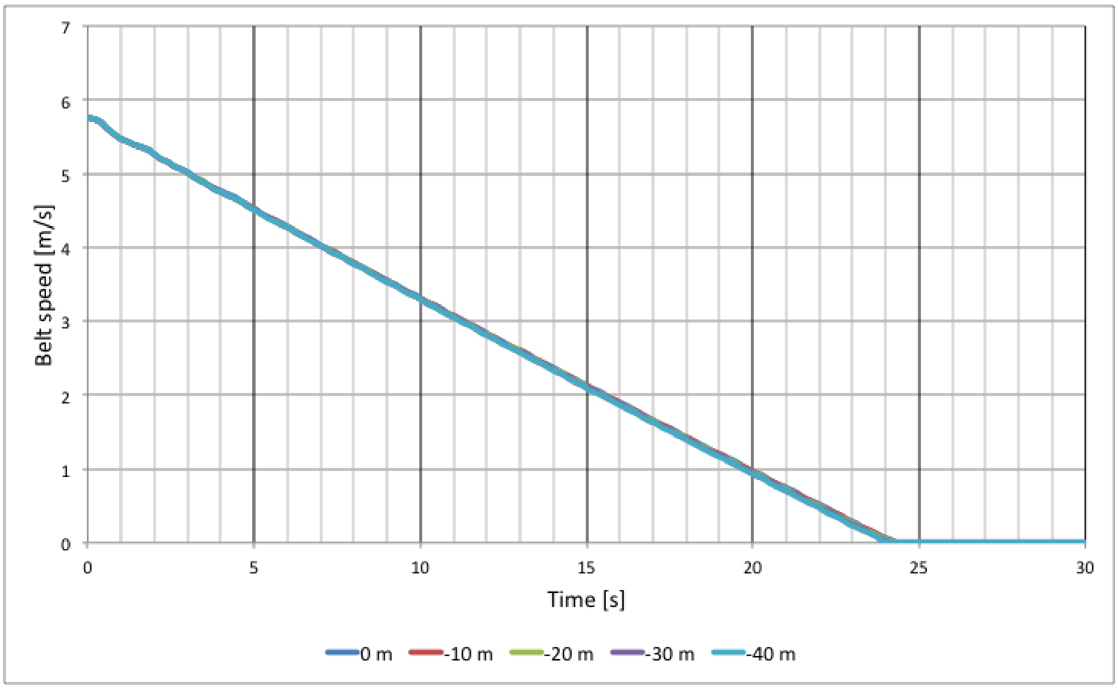

Figure 7. Belt velocity as a function of time during a stop

Figure 8 shows the belt tension in the dip during the same emergency stop. A couple of observations can be made.

First of all, the effect of the dip on the tension in the carry side of the belt in the dip can be observed. Increased magnitude of the dip, varying from 0 m to -40 m, causes the belt tension to decrease. However, this effect is predicted by Equation 19 where it only depends on the term ($m'_l gH$). This can be explained by the effect mentioned earlier about the elasticity of the belt. (Figure 6). The order of the difference is maximum 30%, which is significant.

Secondly, it is interesting to see that the difference between the tension before stop and the minimum tension during stopping is about 17.3 kN. This implies that 500 metres from the head pulley, the theoretical compression wave tension of ΔT_1 of 27.3 kN (=P_d/V) has been reduced to 17.3 kN. This decrease depends purely on the hysteresis in the conveyor system and is therefore independent of the magnitude of the dip. This is confirmed in Figure 8. Note that ΔT_1 in this case is equal to F_D. If a start is aborted, ΔT_1 would be significantly more!

The last observation is that with a dip of -40 m, the originally acceptable basic design needs adjustment because the belt tension becomes too low during the emergency stop. This can be done by each of, or a combination of, the measure(s) mentioned under *Solutions*. With a belt sag of 1% the minimum belt tension is 35 kN. The three per cent belt sag level for this conveyor is therefore 11.6 kN. If this criterion is used, then the design of belt conveyors with a dip of -20m and -30 m should be adjusted since the minimum belt tension drops below the 10 kN.

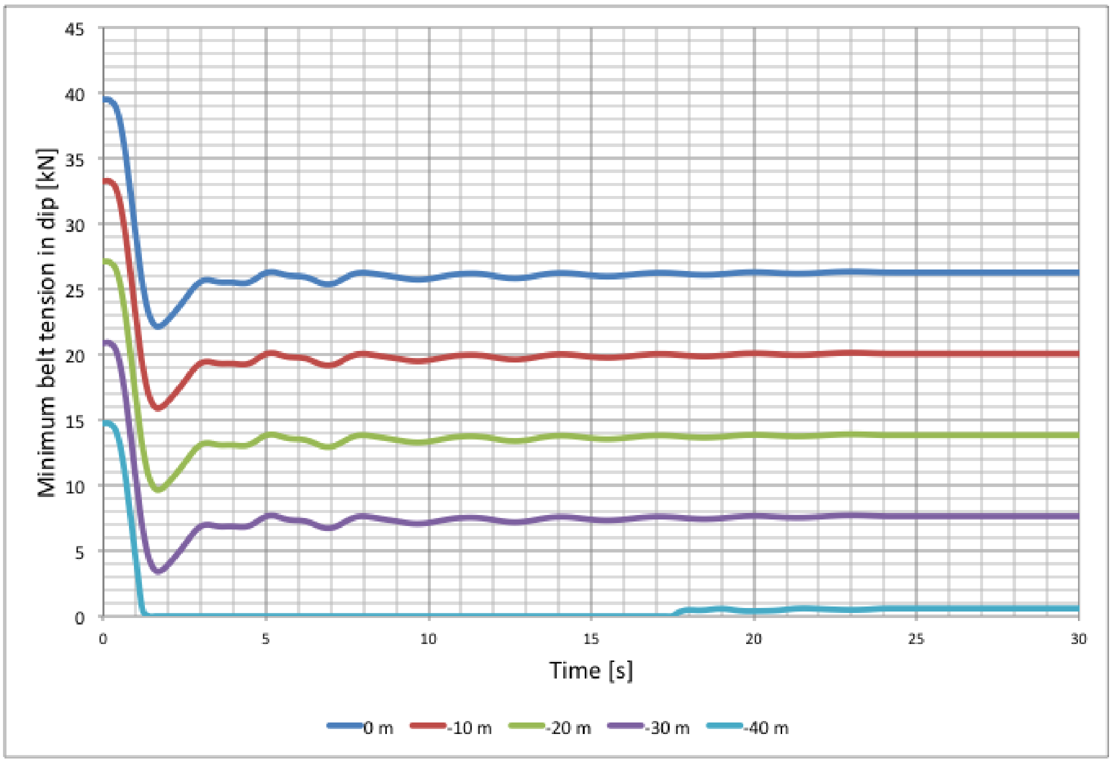

Figure 8. Belt tension in the dip as a function of time during a stop

Examination of Case Four (dip of 30 metres). The minimum tension here is 3.43 kN (Figure 7), whereas the minimum required belt tension to limit the belt sag to 3% is 11.6 kN. Therefore, considering the solutions proposed earlier, the following practical measures could be taken:

— *Use the drive to slow down the belt conveyor and bring it to rest in a controlled manner.* Application of the drive to stop the belt conveyor prevents the sudden drop in belt tension and therefore the compression tension wave. The natural drift time of this conveyor is 24 seconds. Therefore, using the drive to stop the conveyor with an operational (motor) stopping time of less than 24 seconds means that the drive starts to work as a brake, assuming that they allow for regenerative operation. If the drive cannot work under regenerative operation, than stopping in less than 24 seconds simply means that the drive cannot control the belt conveyor and the system effectively drifts to rest. Figure 9 shows the minimum tension in the dip during an operational stop (using a drive that allows for regenerative operation) and a drift stop. The minimum belt tension in the dip during a drift stop was 3.43 kN (Figure 8). As can be seen in Figure 9, the braking action of the drives cause a lower belt tension in the dip compared to the drift stop if the stopping time is less than 24 seconds. Considering 11.6 kN as a minimum tension level to limit the belt sag to 3%, a proper operational stopping time for this conveyor is 41 seconds. (Figure 9).

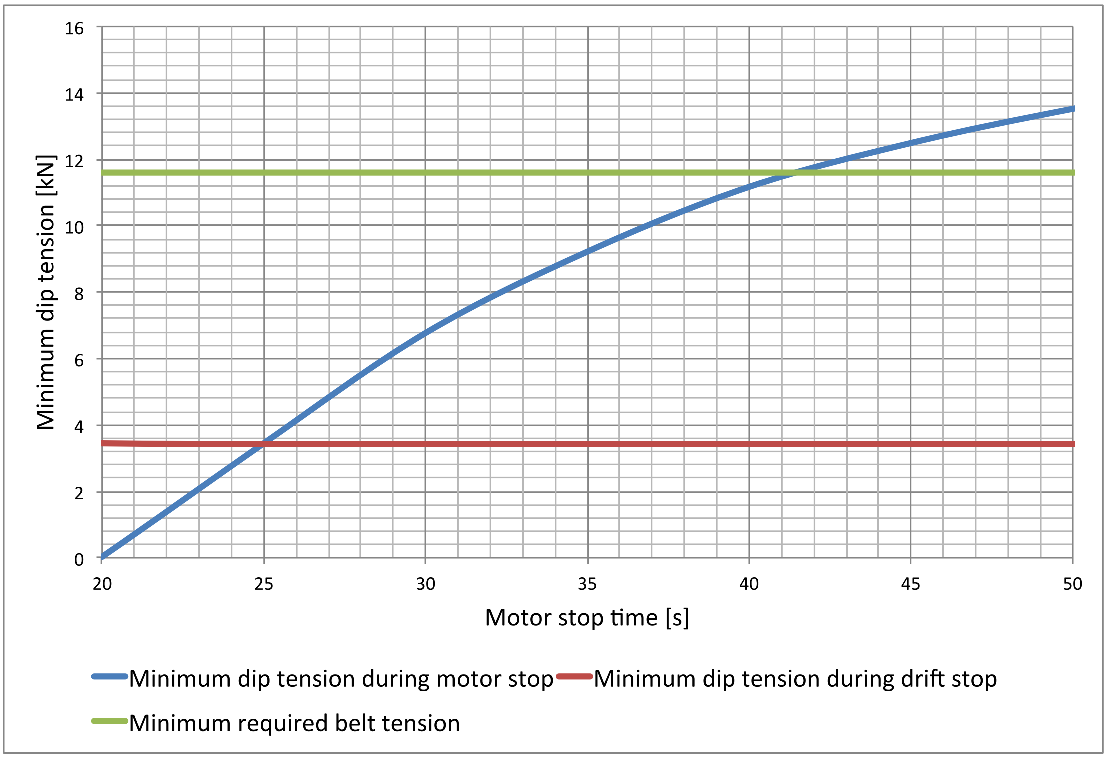

Figure 9. Minimum belt tension in the dip during a stop as a function of the stopping time

—*Increase the belt pre-tension.* Here the solution is pretty straightforward. The minimum belt tension during the drift stop is 3.43 kN whereas 11.6 kN is required. Increasing the take-up tension from 25 kN with 8.2 kN to 33.1 kN suffices (Figure 10). As a safety precaution it is recommended to increase the take-up tension to 35 kN.

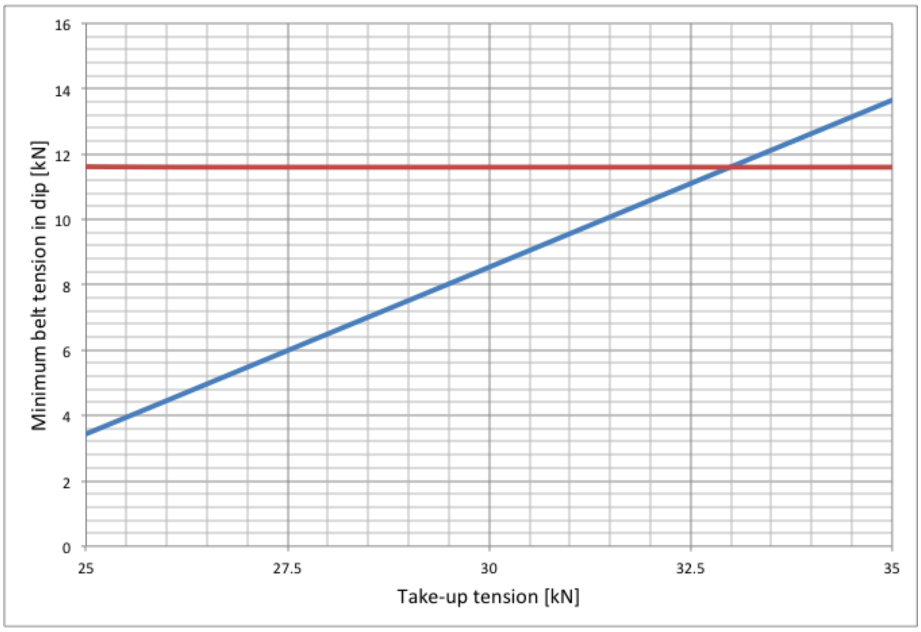

Figure 10. Minimum belt tension in the dip during a drift stop as a function of the take-up tension

—*Slow down the decrease in belt de-tensioning by the application of flywheels.* Flywheels increase the inertia of the drive train that contributes substantially to the total inertia of the system. This increase will increase the stopping time and will decrease the propagation of the compression wave. The drive used in the example has an inertia of 36.5 kg.m^2. If this is increased to about 135 kg.m^2 using a flywheel, then the minimum belt tension exceeds the required 11.6 kN. (Figure 11). By then the drift stopping time is about 39 seconds. If this is too long then a combination between a flywheel and a tail brake is required.

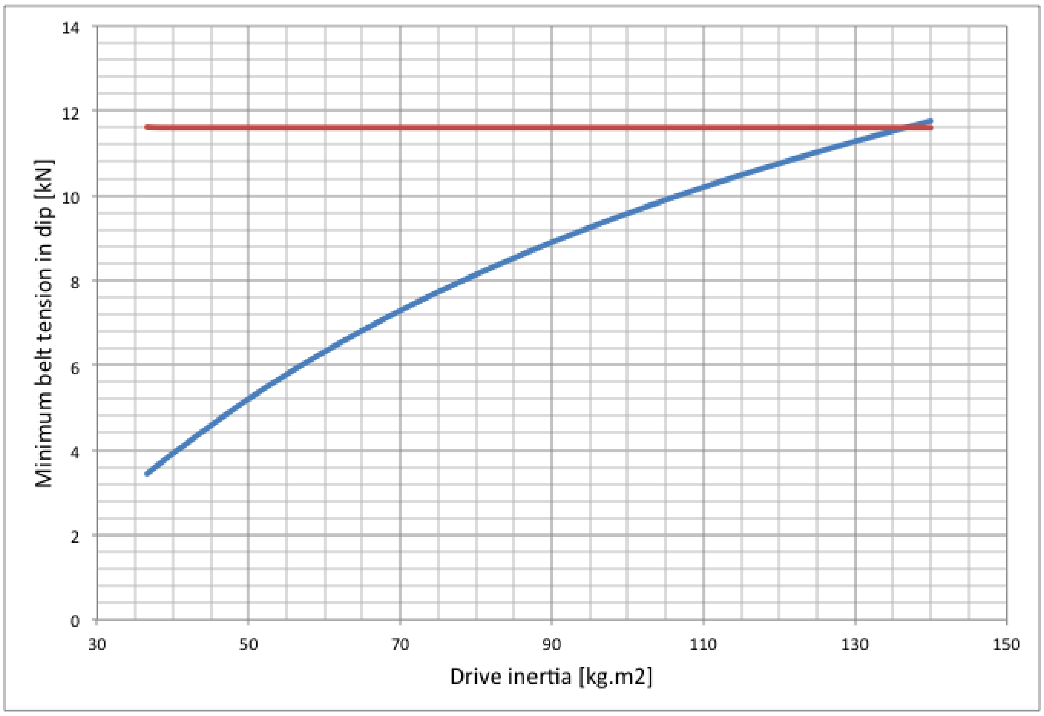

Figure 11. Minimum belt tension in the dip during a drift stop as a function of the drive inertia

—*Counteract the decrease of the belt tension by locally increasing the belt tension, for example by installing a brake that applies on initiation of an emergency stop*. Figure 12 shows what happens if a small brake is installed on the tail pulley. The figure shows the increase of the minimum belt tension as a function of the installed brake rating. As can be seen, a brake sized 6 kNm is required to increase the belt tension over the specified minim of 11.6 kN. The stopping time reduced from 24 seconds to 15 seconds because of the application of the brake. Figure 13 shows the effect of installing a capstan brake in the take-up that holds the take-up pulley during a drift stop. Comparing Figures 8 and 13 shows that Case Two with a dip of -20 m now becomes acceptable. Notice that with the application of a capstan brake the stopping time remains unchanged.

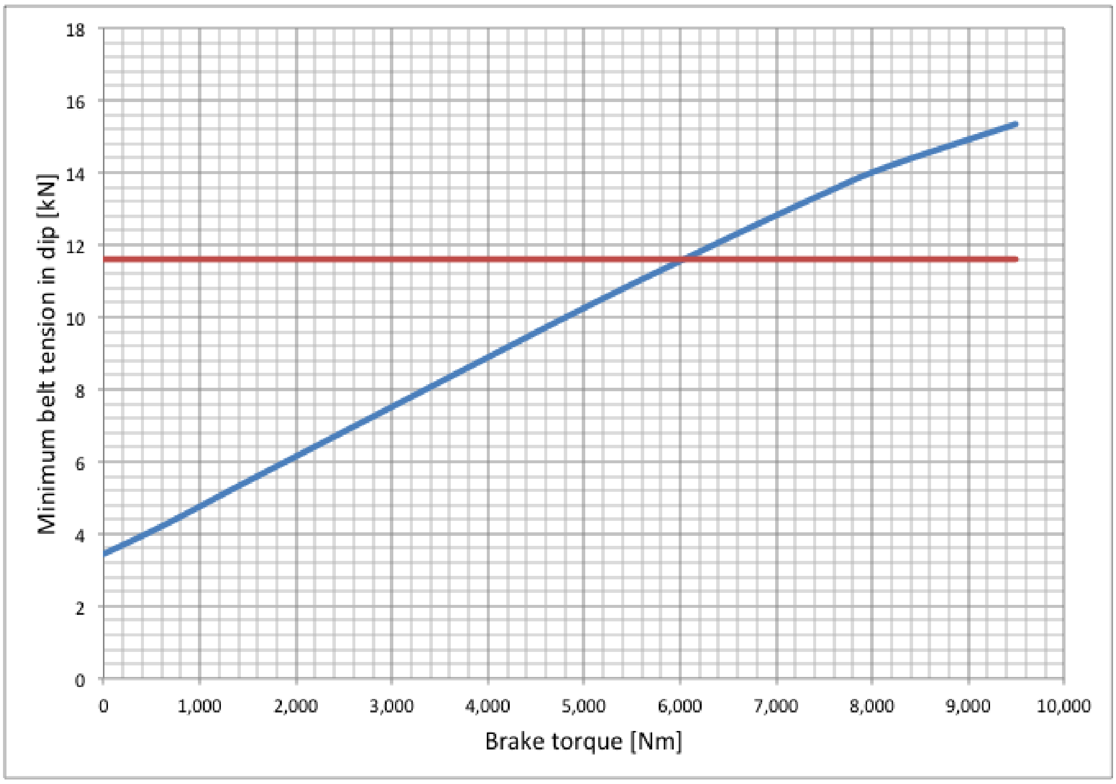

Figure 12. Minimum belt tension in the dip during a brake stop as a function of the brake torque

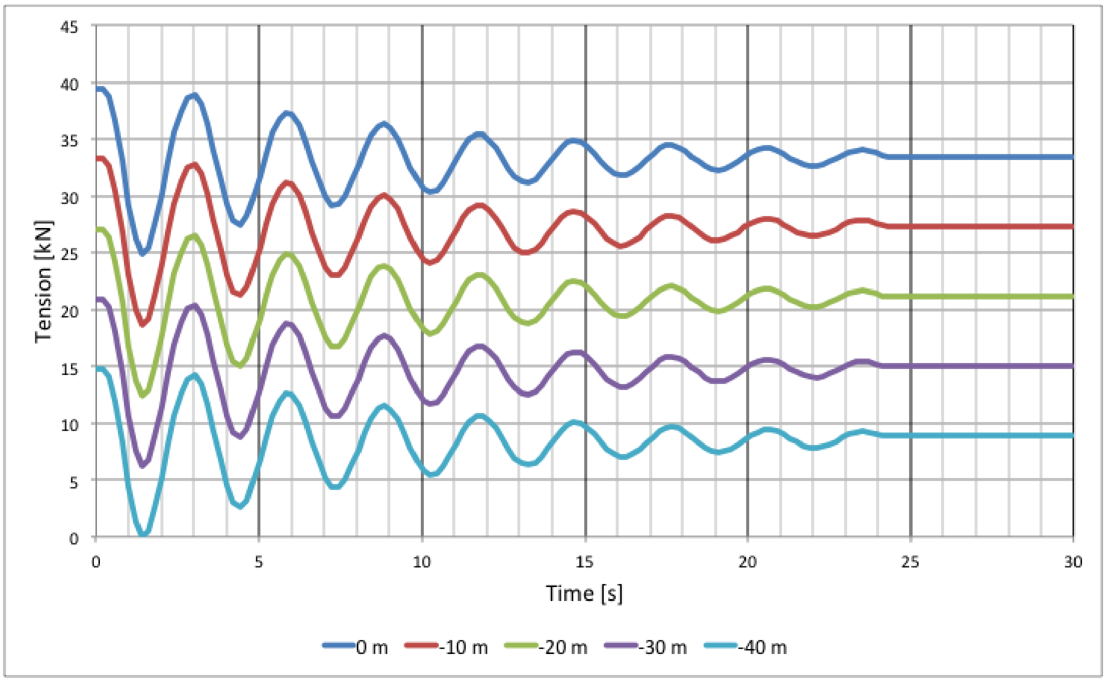

Figure 13. Belt tension in the dip as a function of time during a stop with capstan in the take-up

—*Decrease the idler pitch in the dip*. This is also a straightforward action. From Equation 2 it can be learned that the belt sag is linearly proportional with the inverse of the belt tension and proportional with the quadratic of the idler pitch. Knowing that it is necessary to decrease the maximum belt sag by a factor 3.38 (11.6 kN/3.43 kN), the idler pitch needs to be decreased by a factor 1.84 ($\sqrt{3.38}$). Therefore, by reducing the idler pitch from the original 4.5 m to 2.45 m or less also solves the problem. Knowing that the return idler pitch is 1.5 m, one might opt to reduce the carry idler pitch to 1.5 m.

Obviously, besides using the measures as explained above, combinations are also possible. The discussion of this is, however, outside the scope of this paper.

SUMMARY

This paper showed the effect of a dip in the layout of a belt conveyor on the belt tensions in the dip. It first introduced the reasons behind the requirements of having a minimum tension in the belt. Then it described steady state and transient belt tensions and the background of belt de-tensioning. It further focused on belt de-tensioning in a dip of the belt conveyor. Solutions were presented to prevent unacceptably low belt tensions in the dip. The paper concluded with a practical illustration of the development of a low-tension area in the dip of a belt conveyor and measures to counteract that.

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ABOUT THE AUTHORS

PROF.DR.IR GABRIEL LODEWIJKS

Prof Lodewijks studied mechanical engineering at Twente University and Delft University of Technology, The Netherlands. He obtained a master's degree in 1992 and a PhD on the dynamics of belt systems in 1996. He is president of Conveyor Experts BV, which he established in 1999. In 2000 he was appointed full professor in the department of Transport Engineering and Logistics at the Faculty of Mechanical, Maritime and Materials Engineering. In 2002 he was appointed as chairman of the department, and in 2011 became the deputy dean. His main interest is in belt conveyor technology, automation of transport systems, material engineering and dynamics.

Prof.dr.ir.Gabriel Lodewijks
Delft University of Technology
Faculty of Mechanical, Maritime and Materials Engineering
Department of Marine and Transport Technology
Mekelweg 2
2628 CD, Delft
The Netherlands
Phone: +31 15 278 8793
Fax: +31 15 278 1397
e-mail: g.lodewijks@tudelft.nl or g.lodewijks@conveyor-experts.com

DR.IR YUSONG PANG

Dr Pang graduated from Taiyuan University of Technology in China with a master's degree in electrical engineering in automatic measurement and control in 1999. In 2007, after his PhD research of belt conveyor monitoring and control at the department of Transport Engineering and Logistics, the Faculty of Mechanical, Maritime and Material Engineering, Delft University of Technology, the Netherlands, he worked in Royal Haskoning B.V. as a material handling expert. Currently, he is an assistant professor of the department from which he obtained his PhD. His main interest is in the monitoring, automation and control of material handling systems.

Dr.ir. Yusong Pang
Delft University of Technology
Faculty of Mechanical, Maritime and Materials Engineering
Department of Marine and Transport Technology
Mekelweg 2
2628 CD, Delft
The Netherlands
Phone: +31 15 278 8685
Fax: +31 15 278 1397
e-mail: y.pang@tudelft.nl
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