INVESTIGATION OF MAXIMUM BELT SPEEDS OF IDLERS

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1 INTRODUCTION

The need to increase the operating belt speed for a number of conveyor belts is growing, with intentions to increase production. The low labour and energy requirements make it a favourable means for bulk materials transportation. The design of large conveyor belts can be very complex, with engineers having to take into account the dynamic operating factors and economics into the design. Determining the appropriate belt speed for the conveyor is a major design parameter the selection of idlers and its design limits are some of the important factors to consider when determining the operating belt speed.

At present SANS 1313 serves as the standard for conveyor belt idler design in South Africa, and the standard is used for belt speeds of up to 5 m/s. It is unknown why this belt speed is given, and whether idlers can be run to speeds greater than 5 m/s. The study aims to calculate the maximum belt speed for standardized idler sizes, with established formulae, to check if this belt speed limit of 5 m/s is valid.

Past research has shown that idler failures are most common in the least loaded idlers. D.R Watson[1] presented various reasons for idler failure and the source of the failures as excessive vibrations due to rotating mass unbalances in idlers. No analysis has been done on the effect of rotating mass unbalances on the failure of bearings, and this paper aims to investigate the impact of the added forces generated from a rotating mass unbalance.

The following paper aims to see if the forces arising from a rotating mass unbalance can reduce the bearing life of a ball bearing, and whether it could cause shaft deflection. It also aims to find the future course of research needed to establish standards for mass unbalance and Total Indicated Runout (TIR) for high speed belts.

2 STANDARDS

2.1 Anglo American Standards

Anglo American has its own standards that SANS 1313 do not cover. Idlers are designed for a life of 40,000 hours. This is because the accepted grease life for the bearings used is 35,000 hours when run continuously. It is interesting to note that in the past, Anglo American used design lives of 75,000 hours and 100,000 hours. This was possible because the bearings used in the past were re-greasable. Belt sag should not exceed 2% and the rotational speed limit is 750 rpm for a 127 mm diameter shell

2.2 SANS 1313

SANS 1313 has a comprehensive set of standards for the design and selection of idlers. It also has a number of tests described in detail that need to be performed on samples of idlers to meet the standards set out.

2.2.1 Idler series

Idlers are assigned into different series, based on the shaft diameter (in millimetres). The series are: 20, 25, 30, 35 and 40. These sizes are the respective shaft diameters in millimetres. Using this shaft diameter, Deep Groove Ball bearings are selected. For example, for the series 25 idler which has a 25 mm shaft diameter, the bearings shown in Table 1 can be chosen.

Table 1 - Idler Series 25 Deep Groove Ball Bearings

Bearing Code	Dynamic Load Rating (kN)	Static Load Rating (kN)
6205	14.8	7.8
6305	23.4	11.6
6405	35.8	19.3

Based on the loading of the idler, the appropriate bearing can be chosen ensuring that a bearing life of 40,000 hours is achieved as per the Anglo American standards[1].

2.2.2 Idler Selection

Idler selection is clearly defined for all types of idlers. However the basic methodology is as follows:

1. Once the belt is selected, the belt width is known and depending on the type of idler required, the idler series can be chosen as shown in Table 4 of SANS 1313[3]. For each standardized belt width, the design engineer can select idlers from a variety of idler series'; hence the correct idler series must be selected.
2. Depending on the load calculations, the appropriate idler series is chosen which is suited for the designed load on the idlers.
3. The appropriate bearing can be chosen that can give a life of 40000 hours and the correct idler series is selected.
4. Depending on the idler pitch required and other factors; the appropriate roll diameter would be selected. The procedure to select the appropriate roll diameter is not given in SANS 1313. A larger roll diameter allows for larger idler shafts and thus a larger idler pitch. Therefore the selection of the roll diameter is crucial in designing the conveyor belt to be most economical.

2.2.3 Testing on Idlers

In terms of operational performance, the idlers have three different quality checks set out by SANS 1313 and these are:

1. Total Indicated Runout (TIR)
2. Operating Rolling Resistance
3. Breakaway resistance

The important factor to consider is TIR. D.R Watson[1] did find the standards given in SANS 1313 as unacceptable for high speed idlers. The maximum Total Indicated Runout (T.I.R.) is determined by the formula:

$$\text{TIR} = (\text{Roll length} / 600) + 0{,}55$$

Although there is a test for the ovality of the idler and for the wall thickness, these tests cannot be used to check for the amount of unbalanced mass on the idler. The ovality test is described in SANS 657-3 and a similar test is specified in SANS 1313[3]. Both ovality tests require the Total Indicated Runout to be measured at only 3 points along the length of the idler.

The test specified in SANS 1313[3] for the difference in wall thickness requires measuring the wall thickness at 6 places and to avoid areas in the vicinity of welding. This test will not cannot control the amount of static or dynamic unbalance on the idlers.

3 HIGH SPEED CONVEYOR BELTS

The need for operating conveyor belts faster than ever due to production increases in mines, results in the thorough analysis of the conveyor belt at high speeds with a dynamic analysis as opposed to the traditional static analysis. Today, conveyor belts in Australia run at speeds between 5 m/s and 10m/s and it is dynamically feasible to run conveyor belts at even higher speeds. However it must also be economically viable. The conveyor belt speed is a crucial variable that has effects on the different elements of the conveyor belt. It is vital to understand how it affects the design of the conveyor belt before it can be determined.

The most violent and dangerous effect of operating the conveyor belt at high speeds is the effect of resonance. A major cause for resonance is the excitation of the belt by various elements such as the drive system, and the idlers. As the belt speeds and belt lengths increase, it is important to design, operate and maintain the conveyor belts for optimum efficiency, reliability and safety.

3.1 Belt Speed

The selection of the belt speed must take into account a number of practical considerations. Firstly, the material transported may not allow for a high belt speed especially if it has a very small lump size. Transporting such materials at high speeds could result in a large amount of dust formation and loss of material during transportation.
Another factor to consider is that the land may not permit a belt to run straight horizontally and vertically. Under such circumstances, a belt will be limited in its belt speed.
The belt speed effects the required belt tension, the idler spacing and the starting and stopping procedures and therefore understanding these interactions is crucial before a high operating belt speed is decided.

3.2 Economics

The economics of conveyor belts favours the use of narrow faster belts compared to slower wider belts. The study also says that longer high speed belts are more efficient and appropriate than shorter high speed belts.
The power consumption of the whole conveyor belt increases with an increase in the belt speed. Therefore the capital savings made from a narrower belt must be greater than the increase in the cost from the increase in the power consumption.
Another interesting study showed the cost relationship with idler spacing. It was found that by increasing the idler spacing, the power consumption also increased. This was due to an increase in sag and running resistances. However, the capital costs decrease with an increased idler spacing.

Overall, the requirements for High Belt Speeds are as follows:

- low TIR and mass unbalance
- good starting and stopping procedures
- optimal idler spacing, belt tension and sag control (for controlling power consumption)

3.3 Belting

The conveyor belt can be made from a number of different types of material. A major producer of conveyor belts is Dunlop, which produces a variety of belts. Three commonly made types of belting are:

1. Multiply carcass construction
2. Solid woven carcass construction
3. Steel Cord construction

Each type of belt is suitable for particular materials and conditions. For example, multiply construction conveyor belting with rubber covers is best for conveying hard ores and other hard abrasive materials when the conveyor length is less than 1000 meters and the belt tension up to 125 kN/m. For long conveyors and where the belt tension is greater than 125 kN/m the steel cord belts prove to be the most economical.

The belts also do come in a variation of classes. The different classes of belts differ in; density, wear rate and stiffness. There are standard belt widths manufactured which are given in the CEMA handbook. A commonly used belt is the Steel Cord Class 1000 belt, with a width of 1200 mm.
Each type of belt has its own properties, and one property of the belt that must be considered in the design of high-speed belts is its natural frequency otherwise known as its resonant frequency.

4 IDLER SPACING

The idler spacing affects the resonant frequency of the belt, the required belt tension, and the overall cost of the conveyor belt. The shorter the idler spacing, a greater number of idler sets will be needed to support a set conveyor length and hence there will be higher capital costs.
The idler spacing is typically chosen in the following manner, with consideration to the Idler Diameter.

The idler spacing can be calculated as follows:

Idler Spacing (a) = Load Rating of Idler / Actual Load

Where: Actual Load = Idler Load x K_c
K_c = K1 x K2 x K3 x K4
Load Rating of Idler = Load rating of the bearing x 2

The economic idler spacing can also be found, but this economic idler spacing can only be determined if the costs of the different elements for a particular conveyor system are known. Therefore this investigational project will not consider using the economic idler spacing, but the idler spacing based on the Load Rating of the Idlers.

4.1 Belt Sag

The belt sag is a measure of the dip in the belt between two idler sets. The greater the amount of sag, the greater will be the total resistance of the system. The belt sag can be controlled by increasing the belt tension, until a desirable level is obtained.

A study by J.M.A.M. Brands[9] showed that belt conveyors designed for belt sag of 1.5% or more are not suitable for high speeds. For belt speeds greater than 6m/s, it was found that there is a significant increase in energy loss due to an increase in frictional resistance of over 3%. This results in a higher consumption of power. It is advised to have a low as possible belt sag in order to decrease the power loss. But decreasing the belt sag increases the tension required and this increases the lowest natural frequency of the belt which can be dangerous.

The formula to calculate belt sag is given as:

$$f = \frac{q.a^2}{8T}$$

Where:
f = belt sag
a = idler spacing
q = distance load which is the belt mass plus the mass transported for the idler spacing
T = the tension in the belt

4.2 Idlers

Idler designs in South Africa are standardized by SANS 1313[3]. The standards provide the different design configurations and the quality standards that the idlers need to adhere to.

4.2.1 Failure of Idlers

Anglo American has found that return idlers are the most commonly failing idlers. Return idlers are most susceptible to damage compared to carrying idlers because of the increased damping provided by the material transported[1] for carrying idlers. If it can be proved that return idlers can operate at higher speeds, then carrying idlers should be able to operate at greater speeds to.

D.R Watson[1] conducted an investigation into the failure of idlers and this revealed some interesting points. The main factors contributing to the failure of idlers are given in Table 2.

Table 2 - Factors Contributing to Failure of Idlers

Problem	Comment	Importance
1.	End cap Deflection	Limited Importance
2	Back Seal Design and Bearing Grease.	Small contributory factor to failures.
3	Conveyor Support Structures.	Could be a major contributory factor to failures. If structures are not stiff enough.
4	T.I.R. and unbalance of Rolls.	Major contributing factor to all failures.
5	Looseness of Bolts Holding down Idler Bases.	Contributory factor to failures.

The above list indicates that three main factors to be concerned about are:

1. TIR and Mass unbalance vibrations
   The investigation found that the most significant factor to the failure of the idler rolls was due to the rotating mass unbalances. The findings show that the highest failure rate took place on the least loaded rolls. These rolls that had the least amount of loading were being subjected to the highest T.I.R./out-of-balance forces, and this caused the bearing failures.
2. End cap rigidity
   This is the effect of damage caused to the idler from poor handling. Dropping the idlers and receiving blows to the end cap results in significant damage to the actual idler and thus making the idler more prone to failure.
3. Loose base holding down bolts
   This is a major problem that should be checked periodically with an operating conveyor. D. R Watson et al[21] found that there were excessive vibrations on the base. These vibrations needed a source, which mainly came from the out of balance forces.

Interestingly M. Stewart-Lord[4] found that fatigue is never encountered on an Idler roll bearing. The primary mode of failure of bearings is that the seals collapse, debris enters the rolling surfaces and severe wear, or seizure of the bearing has takes place. According to M. Stewart-Lord[22], there is little use for the ISO formulae for determining the life of the bearings. It is the contamination of the bearing with the ingress of particles that is said to damage the bearings the most. The paper however doesn't address how the seals can fail easily. One possible reason could be the high level of vibrations from the rotating unbalances.

4.3 Bearings

The Bearings and their selection play a vital role in the idlers and the reliability of the idlers. Seize resistant bearings are considered to be most suitable for mining operations. However, the cost of these bearings in comparison to standard ball bearings makes these bearings less favoured. Standard ball bearings can provide good reliability and by analyzing the causes of failures, the reliability of the bearings can sometimes be improved at a lower cost to that if seize resistant bearings were used.
It is important to know what the reasons for the failures of the idlers are before the maximum speed can be calculated. Since the majority of the failures take place in return idlers due to bearing damage[1], the ways in which the bearings can be damaged needs to be determined.

4.3.1 Sealing

The first way in which bearings can be damaged is from the initial failure of the sealing, which leads to contamination of the bearing. Although seals for the idler rolls are of many varying designs, they can be placed into two categories; rubbing seals and labyrinth seals. Rubbing seals have good dirt exclusion properties but unfortunately produce very high friction. When used in a contaminated environment, rubbing seals can be worn rather rapidly, reducing their effective life and efficiency. Labyrinth seals have an interesting operating principle in that

there is no rubbing contact and hence little friction, and the sealing effect occurs from the flow of air in the clearance. Under harsh conditions, there will eventually be ingress of contamination into the seals and the bearings, causing higher friction and damage to the bearings. Due to the inability of seals to be effective especially in the harsh operating conditions of mines, standard bearings in conveyor idlers are prone to failure at a rate not indicated by the L_{10} life.
According to M.Stewart Lord[4], a 1 to 10 relationship exists between the percentage of seized idlers and the increase in power consumption. For example, a 10% rate of seized or near seized Idlers, would demand an additional 100% of power consumption. This was the reason for SKF to develop the seize resistant bearing which can handle a large amount of contamination in the bearing in comparison to standard ball bearings.

4.3.2 Misalignment

One of the major reasons for the failure of idlers is that the shaft of the idler deflects at an angle greater than what can be tolerated by the bearing. When this happens, the bearing typically starts to wear out its inner races and soon the bearings seize. In many bearing applications misalignment can be avoided entirely, or when it is present, bearings can be chosen which can accommodate misalignment such as spherical roller bearings or self-aligning ball bearings.

Idler roll bearings should be able to accommodate a misalignment of up to 0,004 radians. This takes into account shaft deflection under load and manufacturing inaccuracies that cannot be avoided. The formula for deflection is given below.

$$Deflection = \frac{R_m A(L-2A)}{4EI}$$

$$I = \frac{\pi d^4}{4}$$

I = Moment of inertia (mm^4)
R_m = Total Load on both bearings (kN)
L = Length between the outer support points (mm)
A = Bearing support to the edge of Idler support points (mm)
E = 210 000 N/mm^2 for steel.
This constant must be changed depending on the material used for the shaft.

Most bearings do tolerate a deflection of up to 0.004 radians, but when this limit is crossed, the ball bearings fail more quickly. The failure rate can be calculated using de-rating curves for the bearing life, for tapered-roller bearings.

4.4 Lubrication

In South Africa, idler rolls are generally greased for life. There are three reasons for this. The sheer number of Idler rolls in a conveyor installation makes a re-lubrication routine impractical, and very expensive. It is difficult to enforce the necessary safety measures during maintenance. It is also extremely difficult to pump grease through one grease nipple to three or more rollers. The pressures involved are extremely high and, if not carried out properly, this could result in the seals in the outer or first roller being subject to maximum pressure causing the seals to be blown out. Ball bearings are generally preferred to tapered-roller bearings because ball bearings do not need to be lubricated again for a longer period. Hence, the lubrication life of ball bearings is greater than that for tapered-roller bearings.

4.5 Bearing Life Calculation

The bearing life can be calculated as shown by the equation below for a constant rotation speed.

$$L_{hours} = \frac{100000}{60n}\left(\frac{C}{P}\right)^3$$

L = life in hours
N = revolutions per minute
C = Dynamic Load Rating of the bearing
P = Actual Load on the bearing
The bearing ratings for Deep Groove Ball Bearings are given below in Table 3.

Table 3 - Deep Groove Ball Bearing Ratings[10]

Bearing Code	Dynamic Load Rating (kN)	Static Load Rating (kN)
6205	14.8	7.8
6305	23.4	11.6
6306	29.0	16.3

5 MASS UNBALANCE

Unbalance is caused by the displacement of the mass centre-line from the idler's rotation axis by an uneven distribution of the idler mass. Balancing is the correction of this uneven mass distribution by the removal or addition of mass. There are two general forms of balancing: Static and Dynamic. Static balancing involves installing the component into a balancing machine and measuring the heaviest point of the idler in relation to the centre-line, while the part is rotating. If the required balance correction is at a single axial point on the rotor the balance is said to be Single-Plane Unbalanced. Single plane balancing (Figure 1) is adequate for idlers that are short in length.

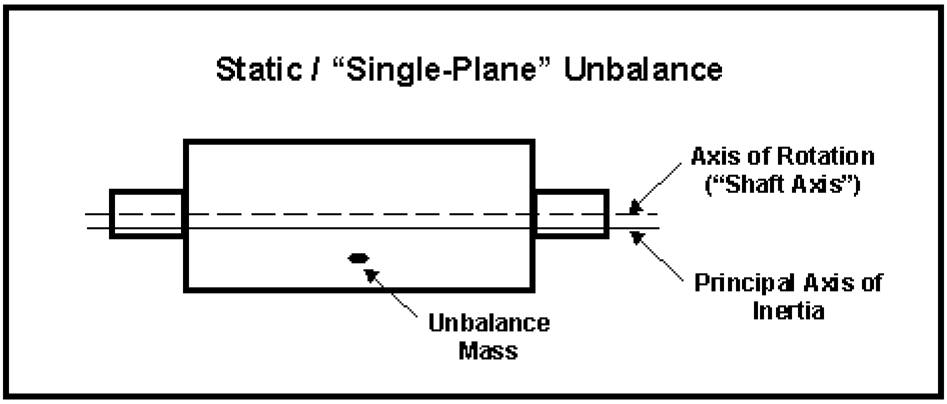

Figure 1 - Static Single Plane Balancing

Dynamic or Two-Plane balancing as shown in Figure 2, is required for idlers of significant length. Idlers with a larger length can have two heavy points at opposing ends, acting independently on the mass centre-line. In order to balance the component, both planes must be corrected for centre-line error. Dynamic balancing is usually done for small and large high-speed components and for large heavy rotating components.

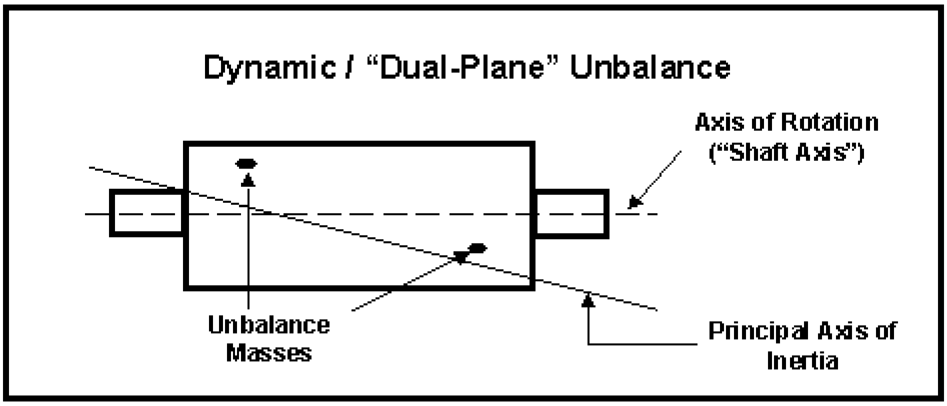

Figure 2 - Dynamic Balancing

5.1 Mass Unbalance Correction Procedures

Balance correction is typically done to specific standards set by ISO 1940/1-1986(E). Balancing is typically done with sophisticated balancing machines. The higher the specification for balancing, the more complex the balancing machine must be. The balancing machine only calculates the amount of unbalance and how it should be corrected. This correction amount is stated as a mass and at an angle on the specific balancing plane to which the mass should be added. There are different ways in which the correction can be made and they are as follows:

- Abrasive Material Removal - Grinding or air powered sanding equipment is used to remove mass at the appropriate location specified.
- Drilling or Milling Material Removal - Drills or end-mills are used to plunge to specified depths at the appropriate location.
- Mass Addition - Mass is added by the addition of epoxy, welding metal strips, or by adding mechanical hardware (set screws, washers) if provisions are included in the design.

The calculation for the force generated from the mass unbalance is:

$$F = m\omega^2 r$$

F = Force
M = mass unbalance
W = rotation speed (radians/sec)
R = radius of the shell
The ISO 1940/1 standards are used to calculate the permissible unbalance of a rotating idler. The importance of balancing the idlers cannot be stressed enough, and if it's not possible to have the idlers balanced, then the idlers should be kept to within G6.3 or G16 standards set out by ISO 1940/1.
The equations to measure the maximum tolerable unbalance and the resulting force is given below. An example is given where a G Rating of 6.3 is chosen, for an idler with an outer diameter of 127 mm, of mass 10 kg, rotating at a speed of 750 rpm and with a face length of 1298 mm.
First the angular velocity is calculated in radians per second.

$$\omega = \frac{2\pi N}{60} = \frac{2x\pi x750}{60} = 78.54 \text{ (rad/s)}$$

The residual specific unbalance (e) is found for a specific G-Rating. This is the amount of mass unbalance that is allowed per mass kilogram for this rating. The residual specific unbalance (e) is calculated below, where N is revolutions per minute (rpm) and G is the rating value.

$$e = \frac{30000G}{N} = \frac{30000x6.3}{750} = 252 \text{ (g.mm/kg)}$$

Next, the permissible residual unbalance is calculated for the idler. This is allowable unbalance for the G6.3 Rating. The permissible residual unbalance (U) is calculated below, e is the residual specific unbalance and M is the mass of the idler.

$$U = eM = 252x10 = 2520 \text{ (g.mm)}$$

Then the force generated for each bearing is calculated in Newtons as follows:

$$F = \frac{U}{1,000,000}\omega^2 = \frac{2520x78.54^2}{1,000,000} = 15.54 \text{ (N)}$$

By selecting a G-Rating for an idler, one can then know the maximum force that can be generated by a rotating mass unbalance, if all the idlers are balanced to the specific G-Rating.

5.2 Equivalent Loading on Bearings

At SKF, a method of adding the normal static force and a force from a rotating mass unbalance is given below.

$$F_m = f_m (F_1 + F_2)$$

Where:
F_m = equivalent Total Force
F_1 = Normal Static Force
F_2 = Force from Rotating Mass unbalance
$f_{m.}$ = equivalence factor found from using Figure 3 below.

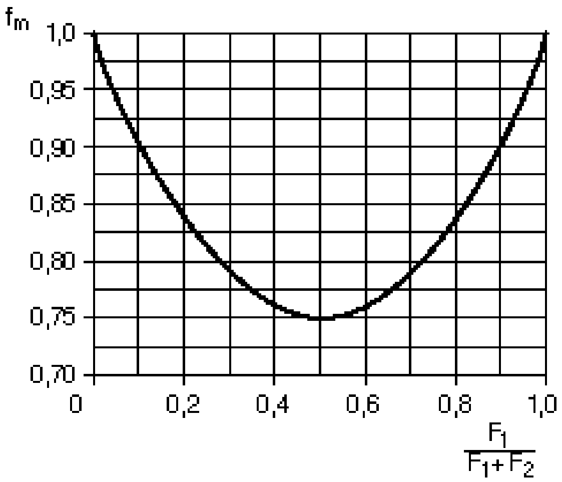

Figure 3 - SKF Equivalence Factor

The forces can now be added by using the factor presented in Figure 8 and by knowing the relevant forces.

6 RESULTS & ANALYSIS

The analysis was done on all the idlers for the following idler types:

1. 3 – Roll Carry
2. 5 – Roll Carry
3. 2 –V Return
4. Flat Return

Table 4 - Operating Parameters for Analysis

Belt Class	1000	Material	
Belt Thickness (mm)	11.2	Density (kg/m^3)	750
Belt Mass per unit length (kg/m)	19.6	Surcharge Angle (degrees)	25
Belt Density (kg/m^3)	1458.333	Lumpsize (mm)	50
Belt Length (m)	1000	Lumpsize factor	0.009
Adjustment Factor	60		
		Idlers Data	
Drive		G6.3 Rating Specification	6.3
Coefficient of friction	0.022	G16 Rating Specification	16
Drive Factor	0.38	Return Shaft Strength (N/mm^2)	210000
Sag %	2	Carry Shaft Strength (N/mm^2)	210000
Trough Angle (degrees)	20		

For each idler, the maximum belt speed, the maximum idler spacing and the deflection at the maximum belt speed was calculated. The maximum belt speed was calculated for idler spacing settings of 1, 2, 3, 4 and 5 meters. The analysis was conducted on Microsoft Excel using the Goal seek function, to limit the bearing life to 40000 hours, and the deflection to 0.004 radians.

The operating conditions are given in Table 4 above. It was assumed that factors such as the belt mass per unit length, the drive factors and the idler shaft's Young's Modulus were constant through the calculations for all the idlers.

6.1 3 – Roll and 5 – Roll Trough Carrying Idlers

The data for series 25, 3-Roll Trough idlers can be seen in Figure 4 below. One can see a comparison of the maximum belt speeds for idler diameters 127 mm and 152 mm. The results show there is little difference in the maximum speeds since there is only a small difference in the total load, which is the weight of the idlers. However a major difference can be seen between the different G-Ratings. The use of the G16 rating reduces the maximum belt speed by half. This shows that the G-rating chosen for the idlers to comply to is of great importance.

A convergence point of all the belt speeds can be noticed on the graph at a belt width of about 1200 mm. This means that using a 127 mm idler would not be much different to using a 152 mm diameter idler. Here, the different G-Ratings also don't matter much more. Therefore to select the most economical idler, the 127 mm idler, assuming it would be the cheapest, should be selected. Unless a larger idler shaft is used with the larger idler diameter, one shouldn't just select a larger idler diameter. The use of the G16 rating would be all right for idlers beyond this point, since there is not much difference in the maximum belt speed. This however should be taken with caution. It is unknown what damage can be done to the idlers by using a higher G-Rating and this requires further research.

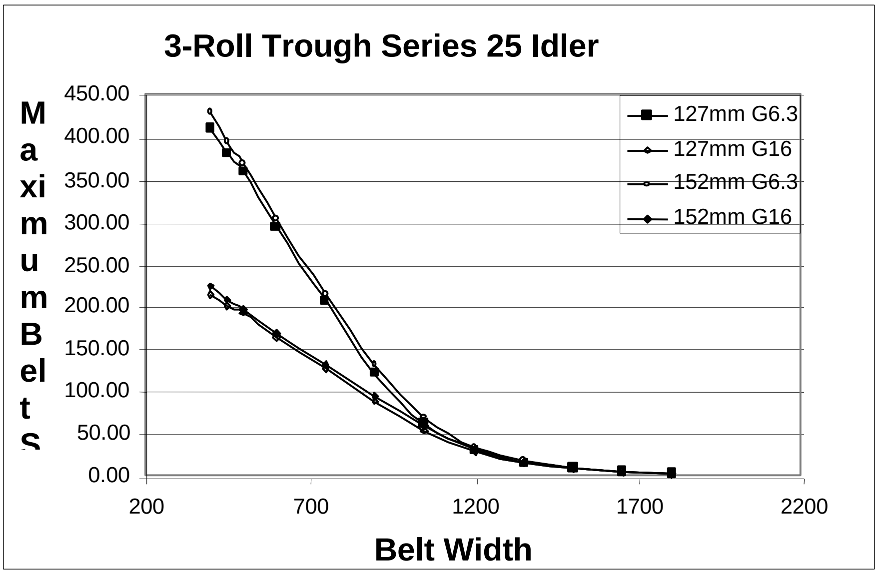

Figure 4 – Comparison of Maximum Belt Speeds for 3 – Roll Series 25 Idlers

The larger the idler shaft, the larger the bearing and hence the higher bearing rating. It is expected therefore that the idlers with a larger shaft will have a higher maximum speed. Comparison of the results for the different idler shafts can be seen in Figure 5.

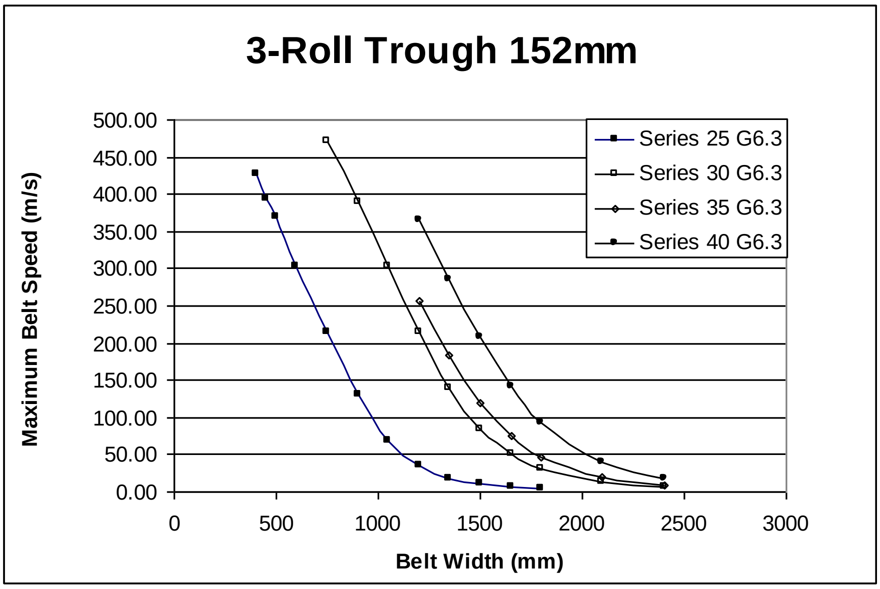

Figure 5 – Comparison of Maximum Belt Speeds for 3 – Roll 152mm Diameter Idlers

The difference in maximum belt speed can be seen for the four curves presented in Figure 5. The smallest idler shaft has the lowest maximum belt speeds as expected. The Series 35 and 40 shaft sizes are not made for belt widths of less than 1200 mm. This is because these shaft sizes would not be economical for smaller loads. The larger sized shaft diameters such as the 35 mm shaft diameter would be ideal for wider belts.

The same patterns can be noticed in 5 – roll trough carry idlers. It was noticed that the impact of the G.ratings is significant for smaller belt widths, but not as significant for the larger belt widths. The convergence point occurs at the point of 1500 mm belt width.
The results also showed that there was a greater advantage in using narrower belts, since these have a higher maximum belt speed. The 5 Roll carry idlers have another advantage in

that there are 127 mm idler shell diameters available even in series 40, whereas for the 3-roll idlers, the 127 mm idler shell diameter is only available up to series 30. This could mean that the 5-Roll idlers could be more economical, since with a higher bearing rating available for the higher series, a larger idler spacing could be used where fewer idler sets are used for a set conveyor length.
The analysis for 5 – Roll and 3 – Roll idlers does show that the G-rating does impact the maximum belt speed, but at unimaginable belt speeds. At belt speeds between 0 and 20 m/s, the G-rating does not affect the maximum belt speed significantly. Therefore the mass unbalance forces cannot be causing idler failures when considering bearing life.

The required speed was calculated by using the design capacity of the conveyor belt. When this was compared to the maximum belt speed for various idler spacings, it was found that this comparison could be used to deduce which idlers, and which idler spacing would be most suitable for selection. Figure 6 shows the comparison and it can be seen that the data lines for 3 m, 4 m and 5 m idler spacings intersect the required speed data line at different points. As expected the larger the idler spacing, the further to the left will be the intersection.

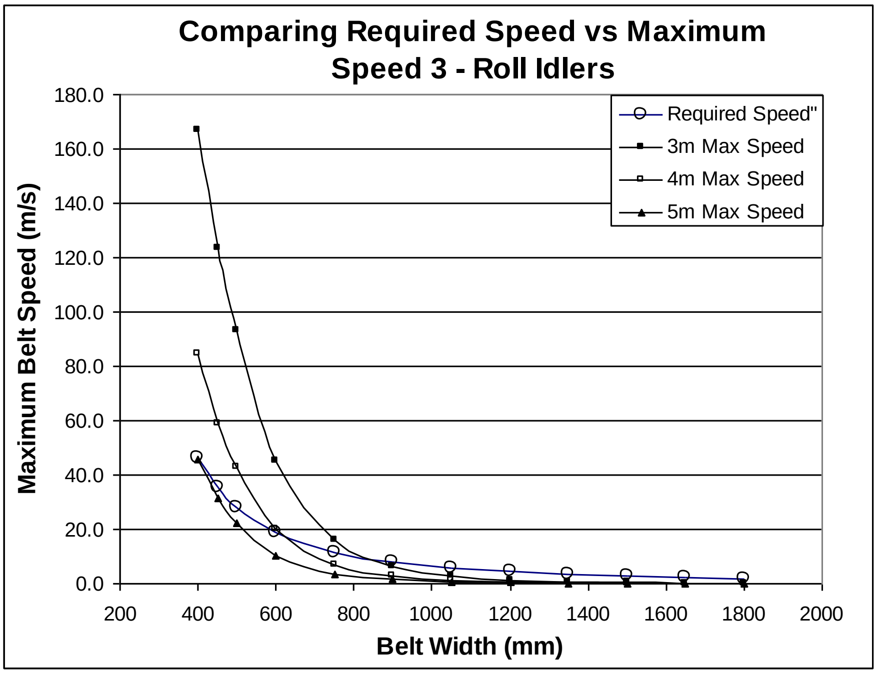

Figure 6 – Required Speed compared to the maximum belt speed

The most important point to take note is that for all the idler spacings, it could be seen that the maximum belt speeds do go below 5 m/s. Therefore the 5 m/s limit used in SANS 1313 does have some validity. But at the design phase of a conveyor belt, an analysis should be done with all idlers available and at different idler spacings, where the maximum belt speeds are calculated, so that the optimal idler is selected. It is clear that the increased load from an increased idler spacing impacts the bearing life a lot more than the forces arising from a rotating mass unbalance.

6.2 Flat Return and 2-Roll Vee Return Idlers

In Figures 7 and 8 the results can be seen for the Flat Return idlers. These idlers are only available in series 25 and series 30. The G6.3 rating does achieve a higher maximum speed as expected as shown in Figure 7. However, there is also a sharp difference in the maximum speeds of the two idler series'. This is evident in Figure 8 where the series 30 idlers can achieve a higher maximum speed.

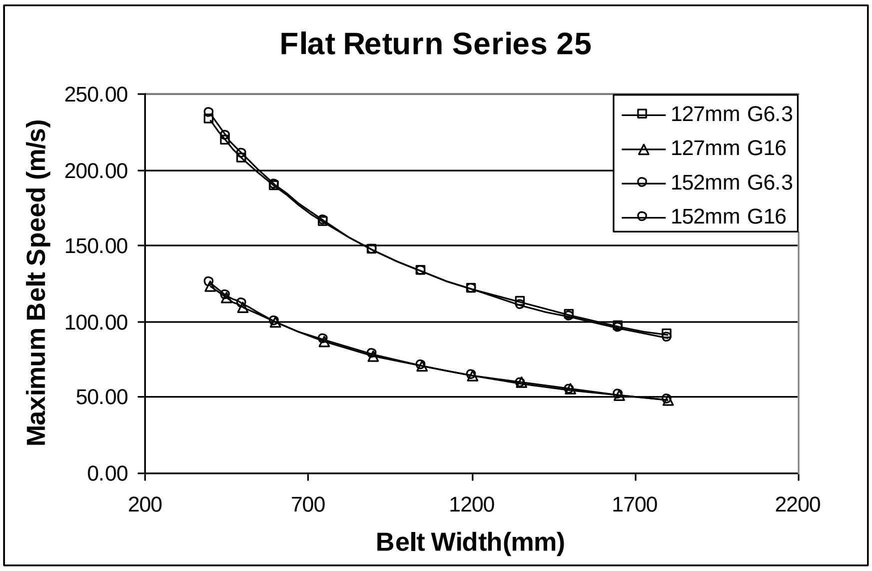

Figure 7 Flat Return Idler, Series 25

In Figure 8 when comparing the 127 mm Series 25 idler to the 127 mm Series 30 idler, it can be seen that be just using a larger shaft and a slightly larger bearing, the maximum belt speed increases significantly. There is little difference in the results when comparing the 127 mm diameter return idler to the 152 mm idler.

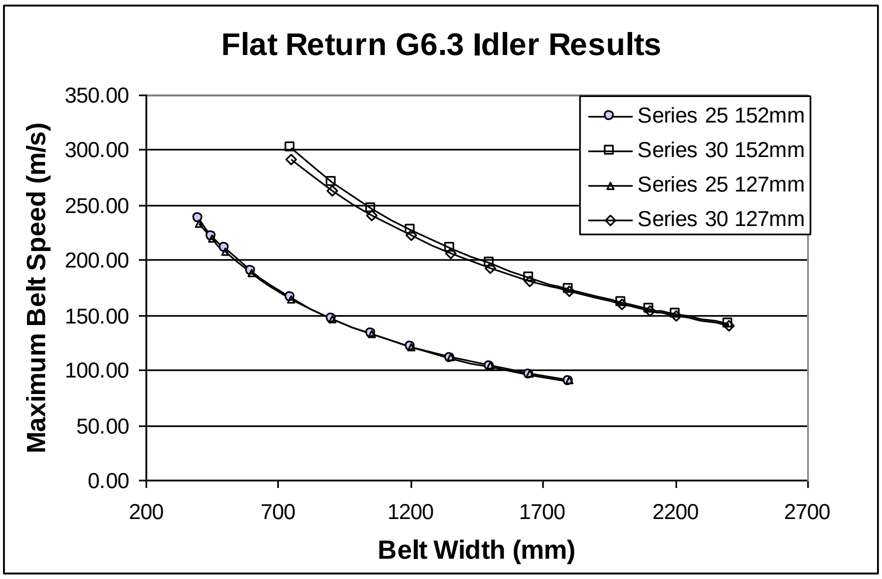

Figure 8 – Flat Return Idler, G6.3 Ratings

The return idlers also show the same patterns as the Flat Return idlers. As expected the 2-roll Vee return idler can operate at faster speeds than the flat return idlers. The G6.3 rating 2-roll Vee return idlers can reach up to speeds of above 250 m/s, whereas the Flat Return idlers can't reach up to 250 m/s. This is as expected since the 2-roll Vee return idlers have the same loading but have four bearings to support the belt compared to the two bearings in the Flat Return idlers.

The results for the 2-roll Vee return idlers also showed that simply increasing the idler diameter does not in fact increase the maximum belt speed significantly. Only the use of a larger shaft diameter allows for a significantly higher maximum belt speed.

Overall, the results for return idlers show that rotating mass unbalance can affect the maximum belt speed of idlers but at impractical speeds. The explanation for why return idlers fail the most cannot be derived from these results. One plausible reason is that there are only two bearings for a flat return idler as opposed to 6 or 10 bearings for 3 Roll and 5 Roll idler sets. Therefore the load is distributed less with the flat return bearings and these bearings are more prone to failure. The idler shells on the return flat idlers are also subjected to a greater loading than the idler shells for the 3 roll and the 5 roll idler sets and therefore the idler shells are also more prone to failure. The solution to this would be to use 2-roll Vee return idlers that can support a greater load because the maximum belt speed is greater for the 2-roll Vee return idlers when compared to the Flat Return idlers.

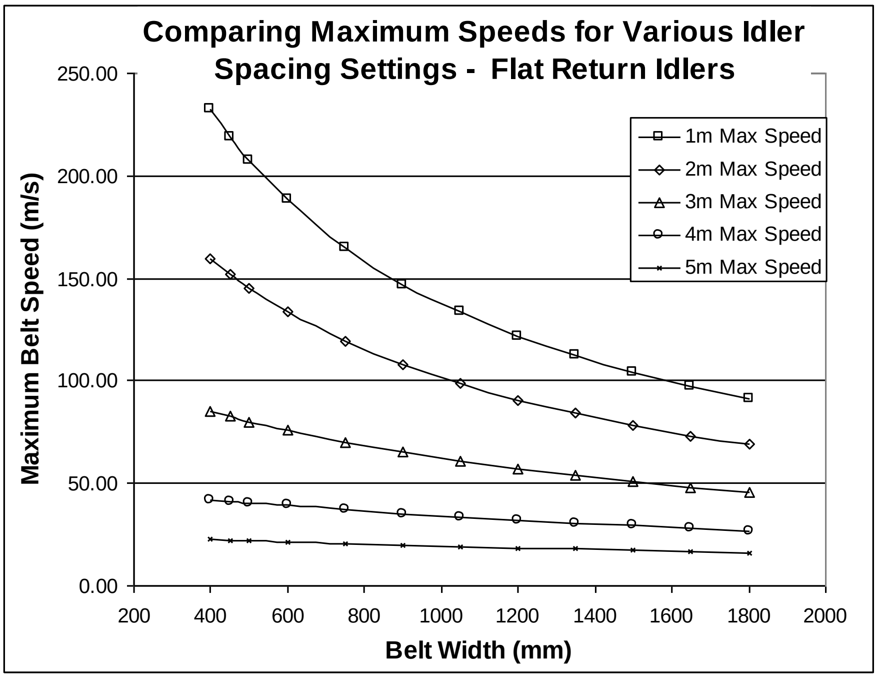

Figure 9 – Flat Return Idlers Maximum Speeds

Figure 9 compares the results for maximum belt speed for different idler spacing settings. It can be seen that for the given design conditions, the belt speed does not go below 5 m/s even for an idler spacing of 5 meters. This shows that there are other reasons for failure of return idlers which cannot be derived from this analysis considering it is known that idlers operating with the same design parameters do experience a high amount of failures.

Overall, comparing the maximum belt speeds for the idlers shows that even when the forces arising from mass unbalance are taken into account, the maximum belt speeds can be run to speeds greater than 5 m/s and the SANS 1313 limit of 5 m/s is not valid.

6.3 Deflection Analysis

The shaft deflection analysis however did yield interesting results. It did show that for certain idler spacing settings, the deflection did cross the 0.004 radians mark. This meant that the idlers could be failing from excessive shaft deflection. The results were the same across all the idler types, using both G6.3 and G16 ratings. It’s important to note that the G-Ratings don’t impact the shaft deflection since the shaft deflection calculation is not dependent on belt speed. The forces arising from a rotating mass unbalance rise exponentially with belt speed hence why this cannot be the reason for idler failures.

The deflection analysis for the Flat Return idler does show idlers for belt widths of 1200 mm and above can cross the 0.004 deflection limit, when an idler spacing of 4m or 5m is used. However, research has shown that return idlers have fail in large numbers even though smaller idler spacing settings are used. Therefore this analysis cannot explain why return idlers fail, even with the forces from a rotating mass unbalance are added.

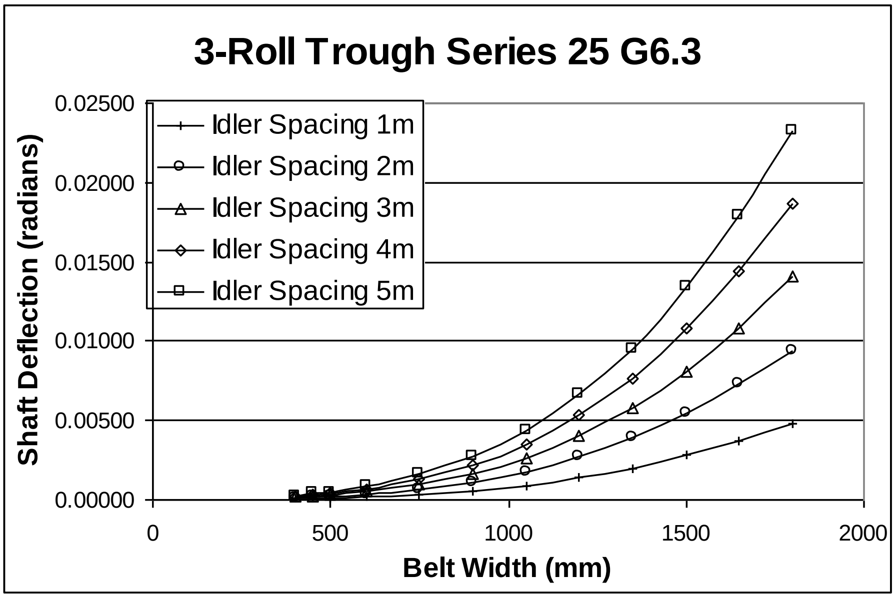

Figure 14 – Deflection Analysis of 3-Roll Trough Series 25 Idlers

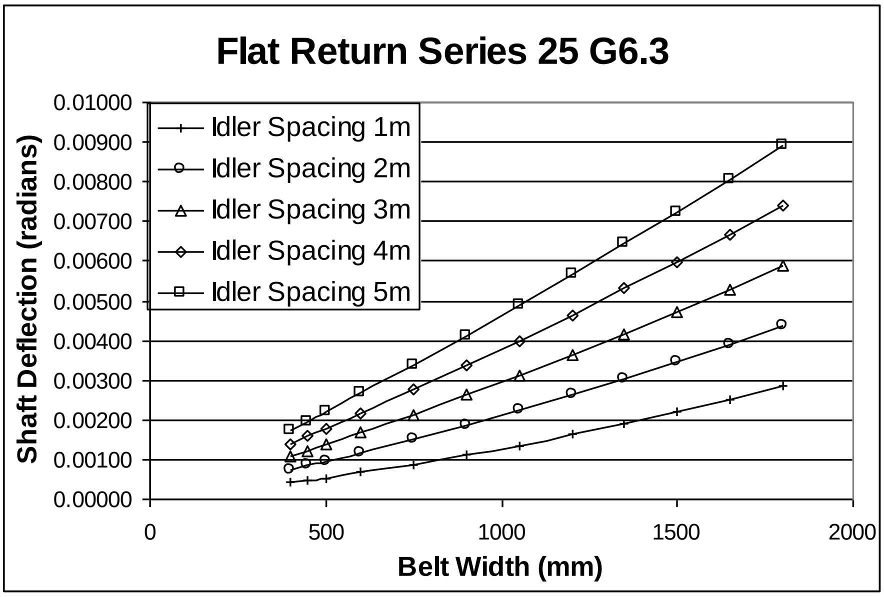

Figure 15 – Deflection Analysis of Flat Return Series 25 Idlers

7 IDLER SELECTION PROCEDURE

A basic idler selection procedure was can be used for selecting appropriate idlers to run at speeds greater than 5m/s using the analysis technique in this study.

1. Calculate the maximum belt speeds for the idlers. Calculate the maximum belt speed for 1, 2, 3, 4, and 5 meter idler spacing settings.
2. Eliminate idlers that have a maximum belt speed less than the required belt speed.
3. Calculate the maximum idler spacing for the idler with the design parameters. Eliminate idler spacing choices for each idler which is greater than the maximum idler spacing allowed.
4. Select the smallest idler possible with the largest idler spacing from the remainder of the list as the best idler.

8 CONCLUSION

Idlers can be run to speeds greater than 5 m/s. However caution must be taken in the design phase because for certain instances carrying idlers cannot go faster than 5 m/s. Return idlers however did show that they can operate at very high speeds for the loading parameters used in this study. The maximum belt speed that idlers can run to clearly depend on the loading parameters, which will vary for each conveyor belt.

It was expected that with the forces arising from a rotating mass unbalance, that return idlers would be expected to show failure due to either shaft deflection or bearing life. The results show that neither are the causes of failure. Therefore the failures of the return idlers cannot be explained by the forces arising from a rotating mass unbalance. The forces arising from the rotating mass unbalance does affect the maximum belt speed achievable, but on at unrealistic belt speeds.

It is advised that for operating a belt at speeds greater than 5 m/s, the ISO Rating of G6.3 be used as a standard. This is the suggested rating for operating idlers between 5 m/s and 10 m/s. Adhering to this standard will require idler manufacturers to improve the quality of their idlers in terms of mass unbalance. Although it would be impossible to balance every idler manufactured, manufacturers can perform quality checks at each stage of the manufacturing process and take steps to reduce mass unbalance at each step.

D R.Watson[1] identified rotating mass unbalances as the main cause of idler failures. The analysis in this research showed that the forces arising from a rotating mass unbalance cannot cause idler failures. The vibrations arising from the mass unbalance however are causing damage to idler components in a manner that cannot be quantified by a static force analysis. Further research is needed to determine the following:

1. How the vibrations damage the idler components?
2. Is the rate of damage to the idler components related to the belt speed?
3. What level of mass unbalance can be considered acceptable as a standard for high speed belts?

The further research will have to look at a vibrations analysis of the system and not a static analysis as done by this study.

The overall technique used in this study can be useful for future selection of idlers. It is a simple way of narrowing the selection of idlers, from the wide choice of idlers available to conveyor belt designers. The key to the selection of the optimal idler does depend on an economic analysis, to find the optimal idler operating with the optimal idler spacing.

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Authors’ CVs

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