CORRECT APPROACH FOR DESIGNING CONVEYOR IDLER ROLLERS

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SYNOPSIS

The principle of designing conveyor idlers is reviewed. There are numerous specifications imposed on the supply of idlers to the industry and the applicability of these documents on the modern conveyor are discussed.

Detailed analysis is given on both the mechanical and structural design aspects of conveyor idlers. Specific attention is given to the design of bearings using the L_{10} bearing life approach. Focus is placed on the key failure mode of idlers.

Idler sealing arrangements and sealing efficiencies are reviewed. The various sealing arrangements and what constitutes an acceptable sealing arrangement are discussed.

The use of polymers in the construction of idler rollers is examined. The relevance of SANS 1313 in the design of idler rollers and bases is brought into the equation.

The subject of energy efficient rollers is touched on as well as the relevance of SANS 1313 on the way forward in the design of idlers.

Appropriate deductions are made with definitive proposals tabled as to expected future progress.

1. INTRODUCTION

The general design of conveyor idler rollers is well documented and freely available to the industry. There are specific items which seem to be ignored by the industry and the purpose of this paper is to bring these anomalies to the forefront.

The paper is structured with the view of first presenting the perceived view of the industry and against this background provide reminders of the not so familiar or forgotten design requirements.

The discussion section initially uses the point of departure as being the industry accepted theoretical approach to loading on the conveyor idler, followed by actual load tests for proving or disproving the methodology.

2. ASSUMPTIONS

It is assumed that the reader is familiar with SANS 1313 Parts 1 and 2 [1 & 2] as being the envelope requirements for supplying idler rollers to the South African industry and SANS 1313 Part 3 [3] being the performance specification in this regard.

3. DISCUSSION

The point of departure for ensuring that everyone is on the same wavelength is to confirm the principal approach to shaft design and bearing sizing for an idler application.

Even before starting with the design analysis, note that the design of an idler is rather peculiar and is not in accordance with the norm for designing shafts and bearings. The normal approach to shaft design is to make the shaft as short as possible in an effort to reduce the lever arm at the support point, thus minimising the bending moment in the application. The reduced bending moment correspondingly results in the bearing being subjected to a minimum angular misalignment under operating conditions. The norm is then a short and stubby shaft with support and reaction points being as close as possible for rigidity.

Compared to an idler application the criteria are the opposite. The shaft is relatively thin in relation to the length of the shaft. Typically on a 1 050 mm belt width three-roll configuration, the bearing would be 25 mm while the shaft length between supports is 390 mm. This ratio is in the region of 16 to 1. It is unlikely that a similar ratio would be selected for conventional shaft design. The slenderness ratio on the return idler is even higher for the same 1 050 mm belt width vee-return idler with a 25 mm shaft spanning 570 mm between supports has a ratio of 24 to 1.

Due to the small diameter shaft spanning long distances, the specifics of the application is one with deflection being a major consideration. Bearing manufacturers design bearings with specific internal clearances and specific ball cage designs.

The internal clearance in a bearing is primarily a temperature consideration. As the temperature requirement of the bearing application increases the internal clearance of the bearing increases accordingly. An idler application does not require the large heat clearance but makes use of the increased clearance to allow for the larger deflection in the application. Following the same consideration, the ball cage cannot be of a rigid construction and needs to be flexible. This has led to the introduction of polyamide cages. Suffice it to say that idlers should be designed using bearings with C4 type clearances and polyamide cages.

Another paper is being presented on the similar subject expanding and going into the detail of clearances and the effect it has on bearing life. Additional technical information is also available from the various bearing manufacturers for the reader to pursue should more detail and confirmation of the aforesaid be required. The narrative above is considered adequate in providing the reader with an understanding of the design criteria.

3.1 UNDERSTANDING BEARING DEFLECTIONS IN IDLER ROLLERS

The empirical formula used in designing the shaft is common knowledge and can be readily reviewed in SANS 1313 Part 3 [3]. For convenience the following information is extracted from the standard.

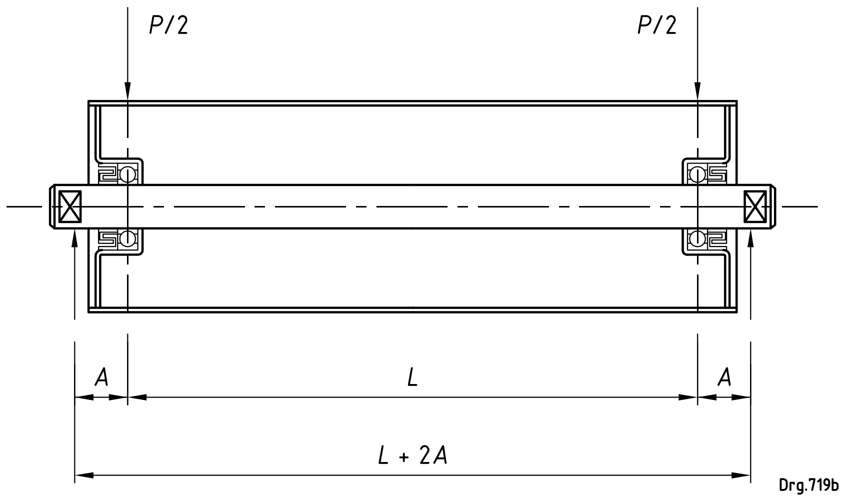

Figure 1. Idler cross-section

From SANS 1313 Part 3 [3]

4.3.4 Load induced shaft deflection at the bearing (see figure 1) can be calculated using the formula:

$$\theta = \frac{P \times A \times L \times 180 \times 60}{4 \times E \times I \times \pi}$$

where

ϑ is the angular deflection, in minutes (min);

P is the load per roll, in Newtons (N);

A is the distance from support point to centre line of bearing, in millimetres (mm);

L is the dimension between the two bearing centres, in millimetres (mm);

E is the modulus of elasticity of shaft material (210 000 N/mm^2); and

I is the moment of inertia of the shaft $\left(\frac{\pi \cdot d^4}{64}\right)$ (mm^4)

where

d is the shaft diameter (mm).

4.3.5 Bearing life calculations can be calculated using the basic rating life formula:

$$L_{10h} = \frac{1\ 000\ 000}{60 \times n} \times \left\{\frac{C}{P}\right\}^p$$

where

L_{10h} is the bearing life expressed in hours. (40 000 h in accordance with this part of SANS 1313);

n is the idler rotational speed in revolutions per minute. (Tables 1 to 9 are based on using a maximum speed of 750 revolutions per minute.);

C is the equivalent dynamic load rating of the bearing per the manufacturer in kilograms (kg).

P is the basic dynamic bearing load i.e. the actual loading applied to the roller at each bearing in kilograms (kg); and

p is 3 for ball bearings and 10/3 for roller bearings.

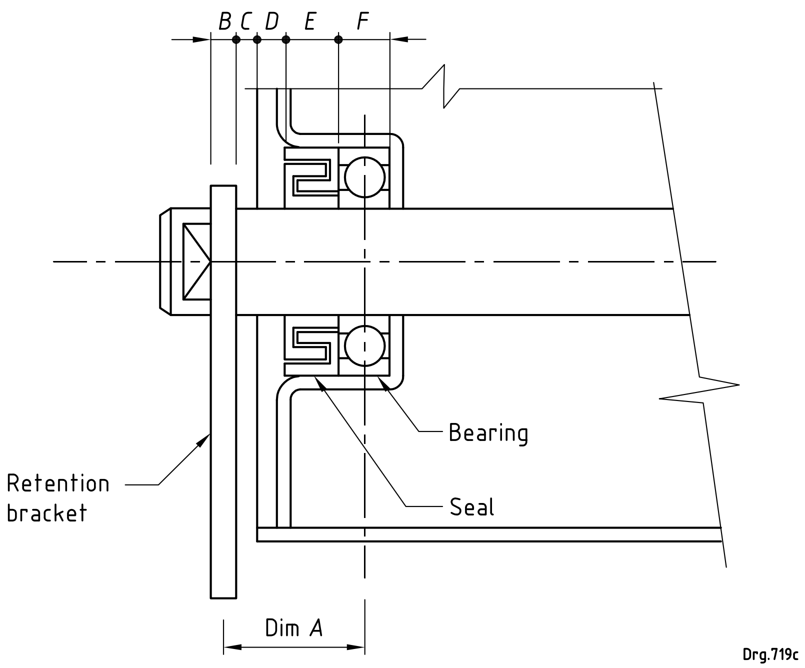

Figure 2. Cross-sectional bearing details

4.3.8 Dimension *A* (see figure 2), can be calculated using the formula:

$$\text{Dim } A = \frac{B}{2} + C + D + E + \frac{F}{2}$$

where

B is the minimum bracket thickness in millimetres in accordance with SANS 1313-1;

C is the clearance in accordance with SANS 1313-1 (inclusive of the 1 mm tolerance per side)

D is the clearance between the end of the roller and the end of the seal. (For these calculations this value is assumed as being equal to half the width of the bearing used. The actual dimension should be confirmed by the idler manufacturer);

E is the width of the seal. (For these calculations this value is assumed as being equal to the width of the bearing. The actual dimension should be confirmed by the idler manufacturer); and

F is the width of the bearing in accordance with SANS 445.

For completeness, the bending stress in the shaft needs to be reviewed. Load induced bending stress can be calculated by using the formula:

$$\sigma = \frac{M}{Z}$$

where

σ = bending stress developed in the shaft at the bearing in N/mm^2

M = Bending moment at the bearing

Z = Section modulus of the shafting which is $\frac{\pi \times d^3}{32}$

where

d is the shaft diameter (mm).

All of this is to bring one to the point where consideration must be given to what the acceptable norms are. The governing factors when designing an idler are the bearing life and shaft deflection. The bearing life consideration is calculated by using an L_{10} value of 40 000 hours.

The shaft deflection consideration is calculated using the maximum allowable deflection which the bearing can be subjected to. For this application bearings with large inner clearance are used, quantifying the allowable deflection as being 15 minutes. The criterion states that five minutes must be allowed for misalignment during the manufacturing process and the remaining balance of the deflection is used in the shaft deflection calculations.

This statement would be acceptable if the bearing housing was of a rigid construction. The cross-section shown through the idler clearly shows a formed bearing housing which will introduce additional deflection in the system localised to the bearing. Due to the fact that this deflection is difficult to quantify, the idler manufacturers tend to ignore the phenomenon.

The best way to quantify the existence of the deflection would be to conduct a finite element analysis of the system. At the outset, reference was made to a 1 050 mm belt width, and series 25 is the most commonly used bearing size in the South African industry. The loading on a typical three-roll idler roller can be readily extracted from one of the tables in SANS 1313 Part 3 [3].

1	2	3	4	5	6	7	8	9	10
		Series							
		20	**25**			**30**		**35**	**40**
		Bearing reference number[a]							
		420 204	**420 205**	**6 205**	**6 305**	**6 206**	**6 306**	**6 307**	**6 308**
Belt width mm	**Gauge** mm	**Allowable dynamic load on the centre roller[b]** kg							
400	180	174	213	235	377	327	471	556	687
450	200	174	213	235	377	327	471	556	687
500	210	174	213	235	377	327	471	556	687
600	250	174	213	235	377	327	471	556	687
750	300	174	213	235	377	327	471	556	687
900	350	164	213	235	370	327	471	556	687
1 050	400	139	213	235	312	327	471	556	687
1 200	460	119	213	235	263	327	471	556	687
1 350	510	105	213	235	232	327	441	556	687
1 500	570	93	213	221	204	327	385	556	687
1 650	620	85	207	201	185	327	349	556	687
1 800	670	78	190	184	169	327	319	553	687
2 000	750	69	168	163	149	315	281	468	687
2 100	775	66	162	157	144	304	270	468	687
2 200	810	63	154	149	137	289	257	445	687
2 400	880	58	141	136	125	264	234	405	637
		Allowable static load on the centre roller kg							
		1 019	2 243	1 427	2 365	2 283	3 262	3 670	4 995
NOTE 1 The above loads do not include the rotating mass of the roll.									
NOTE 2 The rotating mass of the roll should be subtracted from the above loads.									
[a] This bearing reference number does not constitute endorsement by the SABS Standards Division of any particular product.									
[b] Shafts are plain and not stepped.									

Table 1. Three-roll troughing and impact idlers

From this table and assuming a 6 305 bearing, the loading on the centre roller may not exceed 312 kg where the roller is rotating at 750 rpm. From the standard the gauge length of the rollers will be 400 mm and the thickness of the retention bracket 6 mm. The support centres will be 400 + 6 = 406 mm.

As per Figure 1: **dimension L + 2A = 406 mm**

Review the dimensional requirements from Figure 2.

$$\text{Dim } A = \frac{B}{2} + C + D + E + \frac{F}{2}$$

B is the minimum bracket thickness in millimetres which in this case is 6 mm;

C is the 5 mm clearance in accordance with SANS 1313-1

D is the clearance between the end of the roller and the end of the seal. (For these calculations, this value is assumed as being equal to half the width of the bearing used. The actual dimension should be confirmed by the idler manufacturer);

E is the width of the seal. (For these calculations this value is assumed as being equal to the width of the bearing. The actual dimension should be confirmed by the idler manufacturer); and

F is the width of the bearing in accordance with SANS 445. [5]

The width of the 6305 bearing is 17 mm. When applying the logic from SANS 1313 part 3 [3], dimension A is calculated as follows:

$$\text{Dim } A = \frac{B}{2} + C + D + E + \frac{F}{2}$$

B = 6 mm being the thickness of the retention bracket.

C = 5 mm being the clearance between the shell end and the gauge.

D = 8.5 mm being the distance from the end of the shell and the seal.

E = 17 mm being the thickness or width of the seal.

F = 17 mm being the width of the bearing.

$$\text{Dim A} = \frac{6}{2} + 5 + 8{,}5 + 17 + \frac{17}{2}$$

Dimension A = 42 mm

In Figure 1 the bearings centres is noted as dimension L.

If **dimension L + 2A = 406 mm** and **dimension A = 42 mm**

Then

L = 406 – (2 x 42)

Thus

L = 322 mm

In the finite element analysis, a shell diameter of 127 mm was used with a shell thickness of 3.5 mm. The thickness of the bearing housing was 3 mm.

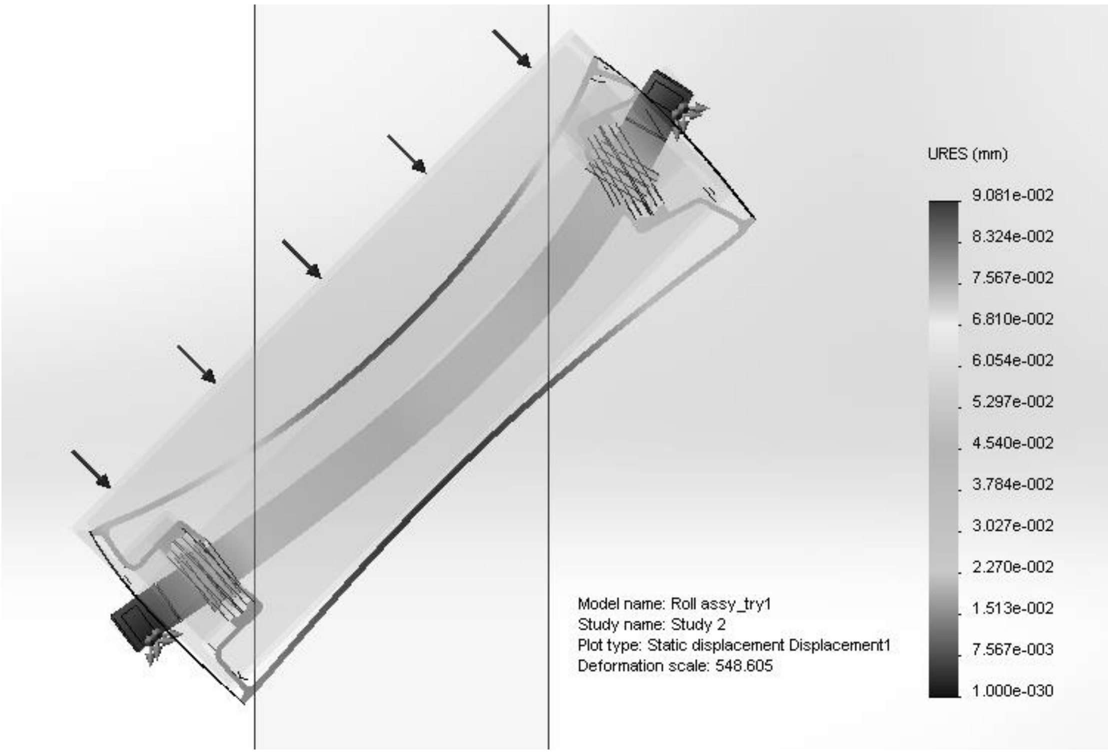

Figure 3. Finite element analysis illustrating deflection of the bearing housing

From the results of the finite element analysis, it is evident that the deflection of the bearing housing is a factor requiring consideration in idler life calculations.

3.2 UNDERSTANDING BEARING FAILURE MODE IN IDLER ROLLERS

One of the statements made relating to idler bearing failure is that the failure mode of idler bearings is not fatigue, but contamination. How is this possible if sealed bearings are being used and care is taken during the manufacturing process ensuring that there is no contamination taking place? The answer to this lies in the construction of the idler.

Using Figure 1 as the point of departure, the idler comprises a steel shell outer casing with a solid shaft where the shell has support from the shaft by means of the two bearings, thus enabling the shell to be rotated. The inside of the shell is filled with air and the bigger the roller the greater this volume of air.

The natural phenomenon is that air expands and contracts as temperature increases and decreases. Following this analogy, the air inside the roller must increase in volume and decrease in volume with variance in temperature. If this then is the case, the only place where the air can flow is through the bearings. This is not possible as sealed bearings are being used – or is it?

A simple experiment was conducted to illustrate this point. A typical roller, suitable for a 1 050 mm belt width was used in the application. A small plastic pipe connected the volume of air inside the roller to a homemade manometer.

The purpose of the experiment was to heat the air inside the roller and observe the corresponding movement in the manometer. To this end, a length of element tape was wound around the roller and subsequently heated. The temperature was increased to 40 degrees Celsius above ambient temperature and the fluid in the manometer monitored.

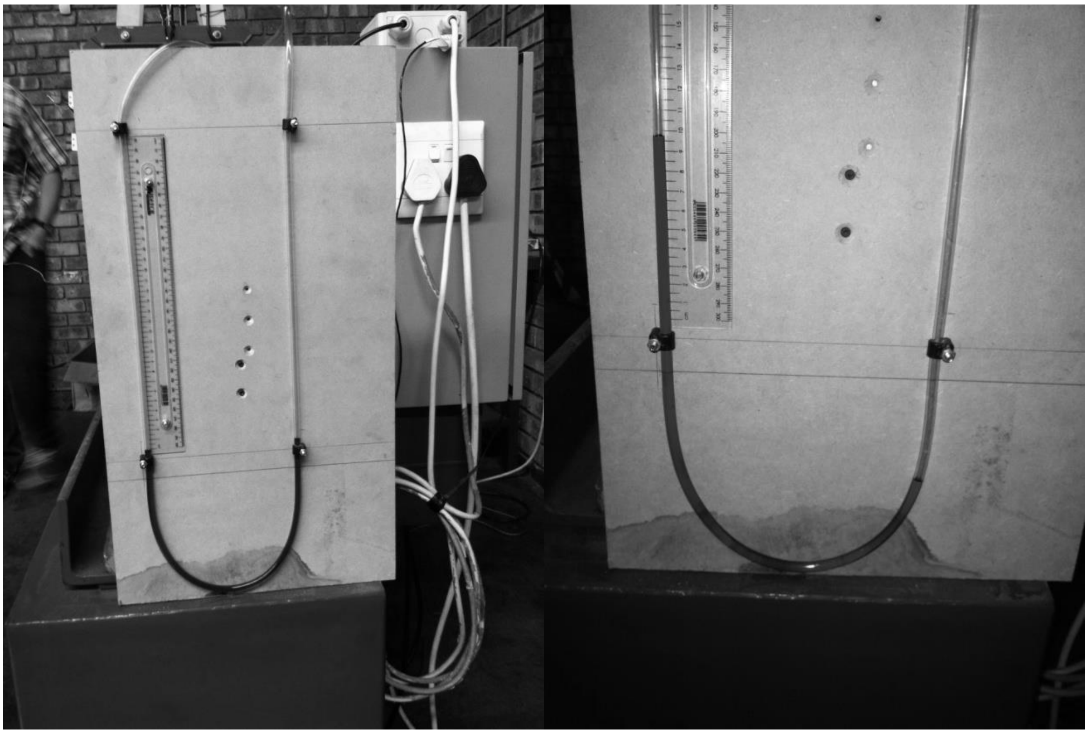

Figure 4. Homemade manometer

Figure 5. Tubing connected to the shell

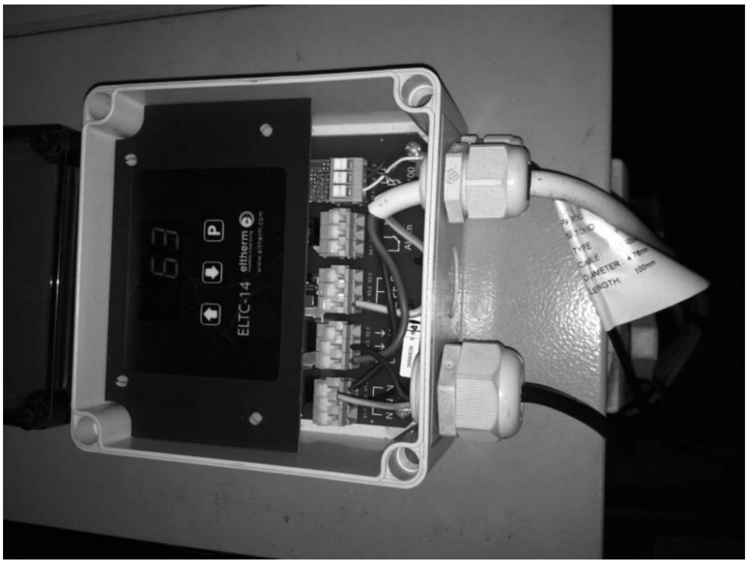

Figure 6. Temperature controller

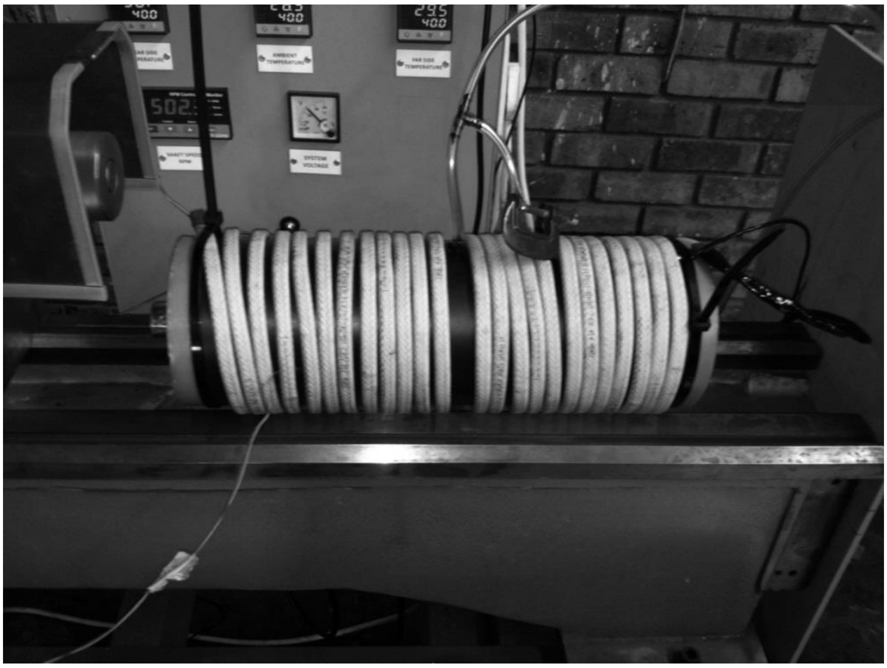

Figure 7. Idler wrapped with a heating element

Figure 8. Epoxy sealed bearing

During the first attempt there was no movement in the manometer. The conclusion was that if there was air movement, it must have leaked past the sealed bearings.

The experiment was then reset with the bearing seals being sealed with epoxy and the experiment was redone. This time around there was movement of the fluid in the manometer. The conclusion was that the air in the roller was moving freely in and out through the bearing seals due to the change in temperature. The purpose of this paper is not to quantify the exact volumes of air flowing through the bearings but to conclusively illustrate the point that it is taking place.

When rollers operate in dusty environments and a typical shower of rain is experienced, there is a definite change in the air temperature causing the cooled air to flow from outside of the roller into the roller cavity through the bearing. Depending on the level of contamination in the air, it is the breathing action of the rolls through the bearings which causes these impurities to eventually bypass the seals and become trapped in the lubrication of bearings. The contamination typically comprises moisture and/or small particles of dust.

The same applies to the variation in temperature between day and night. In South Africa the day and night temperatures easily vary by up to 20 degrees Celsius. This is more than enough to result in flow of air through the bearings as illustrated.

This phenomenon is the key conclusion for the bearing suppliers to state that bearing failure in idlers is as a result of contamination and not due to bearing fatigue. A case in point is to examine rollers which have been in service and have stopped turning. On opening them and cleaning the bearings, the units become free turning again and the failed bearing mode ceased to exist. This has certainly been the writer’s experience as well on numerous occasions.

The lesson to be learnt from this is that there is merit in the creation of a roller that breathes through an alternative path and not through the bearings.

3.3 SEALING EFFICIENCY

The industry specifies various parameters when it comes to seals. One of the requirements is that the rollers must be free running and specific values are noted in the National Standard which must be complied with in order to carry the South African Bureau of Standards (SABS) mark. This refers to both running and break-away friction.

The industry prefers the use of a labyrinth seal as being the best balance between least resistance to rotation and acceptable sealing efficiency. The other end of the scale would be to fit a mechanical seal which will let nothing through, but at the same time requires considerably more energy for rotating the idler roller by itself. This results in a compromise situation. The better the sealing efficiency, the worse the corresponding rotational resistance becomes.

Even in the application of a labyrinth seal in isolation there is also an anomaly. In order for a labyrinth to operate successfully it requires a quantity of grease in the labyrinth path to capture and contain the particles of debris entering the seal. Should the seal have no or very little grease, the rotational resistance from the seal will be minimal. Should the labyrinth be overfilled with grease, the same lubricant will offer great resistance to turning and result in an exponential increase in the rotational resistance.

In some instances the suppliers of idlers neglect to review the basics of their labyrinth seal design. One of these issues is that insufficient consideration is given to the self-cleaning aspects of the labyrinth path. The consideration given to the entrance of water into the labyrinth is the best example. Over the years this has been referred to as establishing where the flood line of a seal is located.

The logic behind this is that when a roller is stationary, water will enter the labyrinth generally at the top and naturally run out at the lowest point in the labyrinth seal. This is best illustrated pictorially and reference is made to the following sketch.

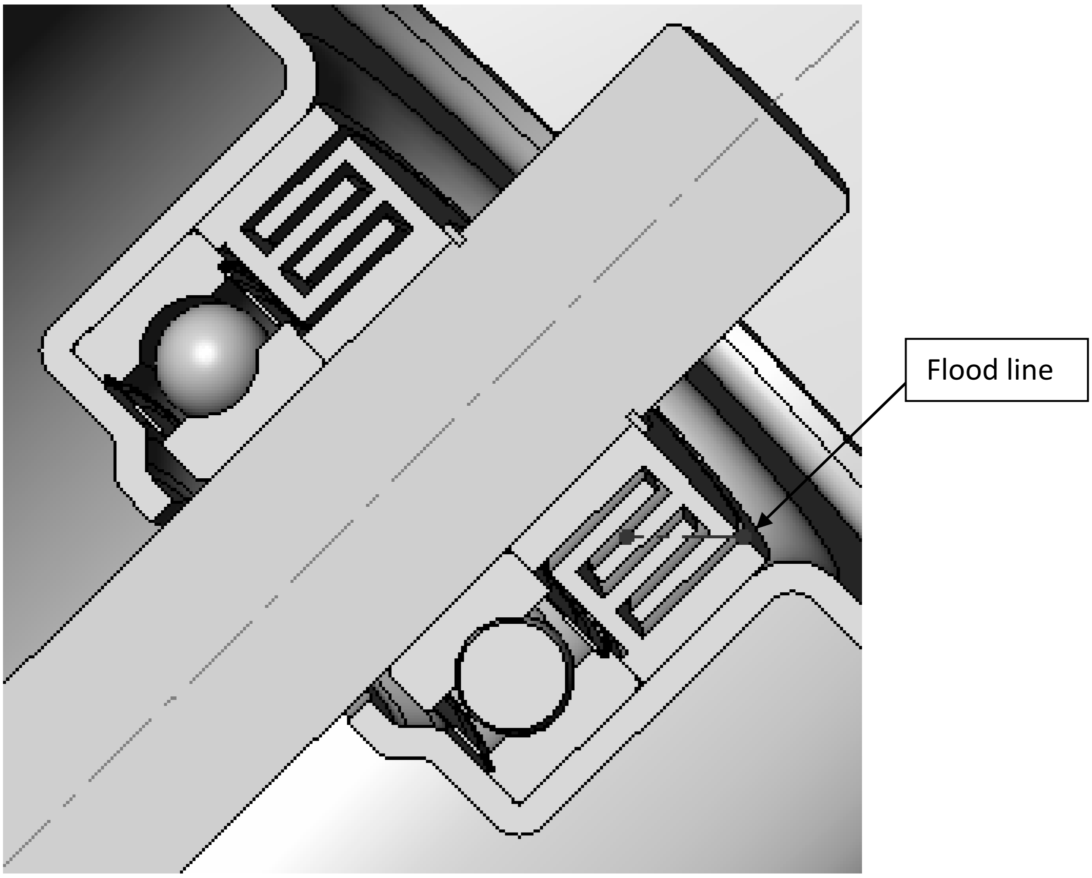

Figure 9. Section illustrating flood line

Note the position of the flood line. In theory, the water enters the labyrinth on the upper level somewhere and flows downward. The lower lip of the labyrinth is where the water drains back out into the atmosphere. The position in Figure 9 shows a stationary roller.

Should the roller start rotating, the water will be redistributed inside the labyrinth. If there is no restriction on the outside, the tendency will be for the water to be washed out. This is referred to as being something like a self-cleaning action.

Should the labyrinth have some sort of a lip or restriction along the outlet path the water will be prevented from being washed out. The implication of this is that once the water has entered the labyrinth it becomes trapped and will not dispel. In this case the water will eventually find its way towards the bearing with disastrous consequences.

The other end of the scale is when the labyrinth path is located nearest to the shaft. In this application there is no evidence of a self-cleaning action of the labyrinth path. Any debris or water entering the labyrinth is unlikely to be dispelled and may impede the reliability of the product.

3.4 LUBRICANT RETENTION

Lubricant retention forms part of the sealing efficiency of the bearing and must be mentioned to reaffirm requirements in this regard. The seal of an idler roller must be designed and checked so that it will adequately retain the lubricant in the bearing over an extended period of time.

This becomes a very specific requirement when considering that the function of the sealing arrangement is one of sealed for life. The bearing is only lubricated once during manufacture and then never again over the full life expectancy of the idler roller. According to the National Standard, the calculated bearing life is 40 000 hours which equates to an uninterrupted application period of 4.5 years. In reality this period becomes longer when rationalised with respect to actual utilisation of the idler roller and the periods of time that the conveyor belt installation will not be operational. A system operating eight hours per day will require the unit to remain effective for a period of 13.5 years.

Lubricant retention is best achieved by using a contact type seal. In an idler application the severity of the contact sealing pressure of the seal must be balanced with the requirement of having minimal rotational resistance. Very good results are achieved by making use of sealed bearings supplied complete with either lip or labyrinth type seals.

Open bearings are also used in the manufacture of idler rollers. In this instance the end user is cautioned to only use products from reputable suppliers. The sealing arrangement of the supplier must be properly evaluated for compliance to lubricant retention and substantiated with documented evidence with respect to reliability under operating conditions in the field.

3.5 CROSS LOADING OF BEARINGS

The way that most and if not all idler rollers are constructed is that the bearings are retained and located on a common shaft with retaining rings on the outside. Here again the design of an idler roller is such that it is not in accordance with sound mechanical engineering practices. The correct way would be to have the one bearing fully located and the other bearing floating.

The point is raised to highlight the fact that bearings can become cross loaded very easily through the accumulation of acceptable manufacturing and production tolerances. When dealing with synthetic rollers with steel shafts the problem is exacerbated. The differential linear expansion of dissimilar materials results in problems developing very quickly.

According to SANS 1313, the maximum permissible end float which may be present in a roller is 0.7 mm. Unless special equipment is used, it is difficult for the longer distance to be accurately measured during the manufacture of idler roller components. Tape measures should measure accurately to the nearest mm. Conveyor idler shafts require to be accurately manufactured to fractions of a mm, thus prohibiting the use of tape measures as being unsuitable for the application.

Unfortunately industry overlooks this critical requirement and incorrectly makes use of tape measurements for this application. This invariably leads to inaccuracy of manufacture and consequently increases the risk and possibility of bearings becoming cross loaded.

In the case of polymer shells the expansion coefficient of shell material is 0.00012 mm/mm/°C and that of the steel shaft is 0.000012 mm/mm/°C. Being generous and allowing the 0.7 mm end float under all conditions, once the temperature variance exceeds 20 degrees Celsius, the bearings start binding.

On applying the sound engineering approach of having one bearing fixed and the other one floating, the problem is largely solved. On an all steel idler application the problem goes away altogether while on a polymer type idler the problem is relieved and only manifests itself at greater temperature variances.

The extent of the relief is relative to the differential expansion coefficients of the material. The differential expansion is tabled below and is expressed in terms of 10 degree Celsius increments. Still using the 1 050 mm belt width example as the norm, the shell length of the troughing roller is 390 mm and the flat return roller 1 146 mm.

DIFFERENTIAL EXPANSION OF SHAFT AND SHELL LENGTH (in mm)						
Temp variance in degrees Celsius	**Increase in length of the 390 lg trough roller**			**Increase in length of the 1 146 lg return roller**		
	Polymer shell	**Steel shaft**	**Variance**	**Polymer shell**	**Steel shaft**	**Variance**
0	0.00	0.00	**0.00**	0.00	0.00	**0.00**
10	0.39	0.04	**0.35**	1.15	0.11	**1.03**
20	0.78	0.08	**0.70**	2.29	0.23	**2.06**
30	1.17	0.12	**1.05**	3.44	0.34	**3.09**
40	1.56	0.16	**1.40**	4.58	0.46	**4.13**
50	1.95	0.20	**1.76**	5.73	0.57	**5.16**
60	2.34	0.23	**2.11**	6.88	0.69	**6.19**
Co-efficient of expansion for steel: 0.00001						
Co-efficient of expansion for polymer: 0.0001						

Table 2. Differential expansion of shaft and shell length

A variance in length of up to 2 mm should not present a problem from an application perspective when using the SANS 1313 envelope sizes. This standard calls for a maximum end float of 0.7 mm and the clearance to the retention brackets is 5 mm. This implies that the troughing roller is acceptable over a temperature range of 50 degrees and the return roller only 20 degrees. The typical temperature variance in South Africa would peak in the 50 degrees mark. This makes the troughing roller acceptable for the application but not the return roller. Special attention is required for deviating from the standard in order to allow for the use of the return roller for this application.

When considering the extent of linear expansion over the temperature range, the conventional construction of pure polymer shell rollers with external circlips is questioned. For this application, the fixed and floating bearing arrangement becomes a minimum requirement.

Up to this point the loading and calculations have been considered for the roller in isolation only. The next step is to consider the actual loading being induced on the idler under normal operating conditions.

3.6 PERCEIVED IDLER LOADING

The convention for determining the loading on an idler is typically to consider the material on the belt as well as the mass of the belting itself. The self-weight of the rotating masses of the idler roller is also included. This approach is similar to the loading conditions for beam design and is reflected in the following diagram.

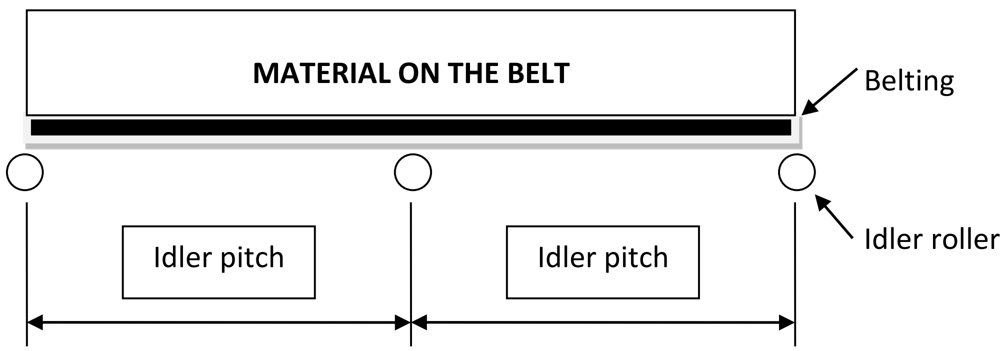

Figure 10. Conventional idler loading condition

The actual application differs from this approach in that it is not a typical rigid beam design construction. The application is similar to that of a suspended rope application which is the same methodology that is used for determining the sag in a conveyor belt.

Presenting this technology, the following applies in accordance with ISO 5048 [5]

$$T_s = \frac{l\,(q_B + q_G)g}{8\,(\frac{h}{l})}$$

T_s = Sag tension

q_B = Unit length of belting in kg/m

q_G = Unit length of material in kg/m

g = Gravitational force kg/mm

l = Idler pitch in m

h = Sag distance in the belt in m

By making use of an example the following becomes apparent:

Still using a 1 050 mm belt width with the following loading

Belt speed of 4.987 m/s

Material loading 1 700 tons per hour or 94.69 kg/m

Loading from belting 19.3 kg/m.

Rotating mass per roller 4.9 kg each = 4.9 x 3 = 14.7 kg per idler set

Idlers pitched at 1 370 mm centres.

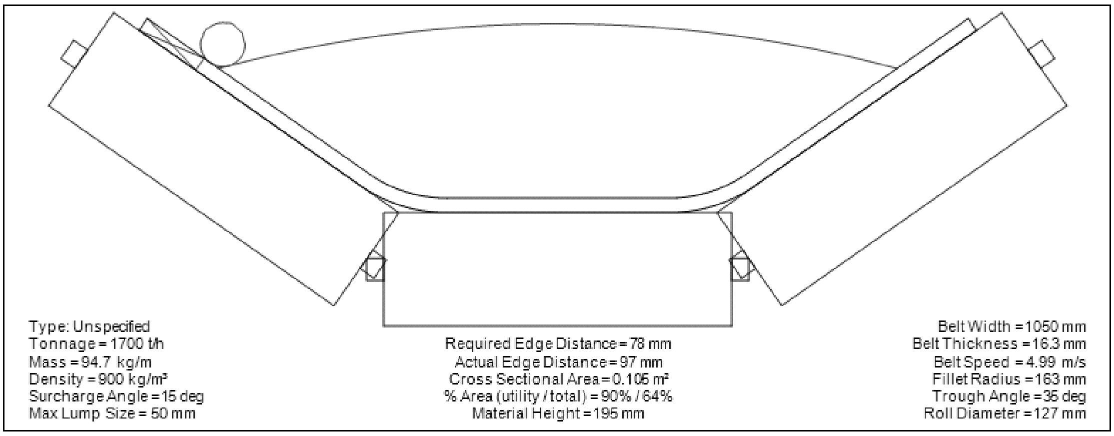

Figure 11. Extract from equivalent conveyor calculation

Loading on the idler set using the conventional approach:

Total loading = material load + belt load + idler self-weight

T_l = 94.69 kg/m x 1.37 m + 19.3 kg/m x 1.27 m + 14.7 kg

T_l = 129.73 kg + 26.4 kg + 14.7 kg

T_l = 170.83 kg

Considering a typical sag of 2%, the sag tension for this application will be

$$T_s = \frac{l\,(q_B + q_G)g}{8\,(\frac{h}{l})}$$

l = 1.37 m

q_B =

q_G =

g = 9.81 m/s^2

$\frac{h}{l}$ = 2% = 0.02

$$T_s = \frac{1{,}37\,(19{,}3 + 94{,}69)9{,}81}{8\,(0{,}02)}$$

T_s = 9574,95 N = 9,57 kN

Sag tension in this instance is 9.57 kN

The 2% sag over the distance of 1 370 mm idler equates to 0.02 x 1 370 = 27.4 mm.

Constructing the force diagram of the forces in play it looks as follows:

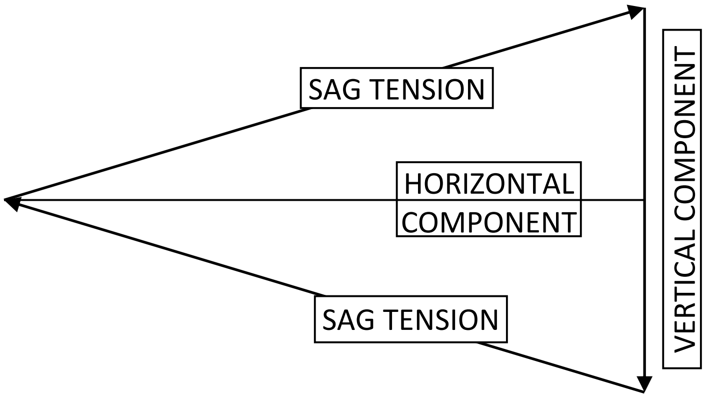

Figure 12. Sag tension force diagram

From this it is apparent that the vertical component of the sag tension needs to be supported by the roller. This is an additional load. The additional load of the idler is calculated by solving the force diagram.

The angle of the sag force is calculated by the ratio of the sag distance relative to the idler pitch. The 2% sag value occurs midway between the idlers. The corresponding angle is

2% sag on a 1 370 mm idler pitch = 0.02 x 1 370 = 27.4 mm

Half the idler pitch = 1 370 mm / 2 = 685 mm

From the above, the angle of the force relative to the horizontal is the tangent ratio of a 27.4 rise on a base of 685 mm.

The arc tan angle of the value 27.4 ÷ 685 = 0.04 = 2.29 degrees

The total additional force exerted on the roller is double that of the sag tension which equates to

Additional force = 2 x sin (2.29 deg.) x 9.57 kN sag tension

= 0.764 kN

Expressing this value in terms of kg it is

0.764 kN x 1 000 ÷ 9.81 = 77.88 kg

This value needs to be added to the loading already being exerted on the idler by the material, belting and rotating masses as calculated in the previous section.

For completeness the loading value now stands at:

Total loading = 170.83 kg + 77.88 kg = 248.71 kg.

But still, this is not the end.

SENSITIVITY ANALYSIS OF THE SAG TENSION COMPONENT ON AN INSTALLATION

The tension values on a conveyor installation are dynamic and alter continually as the loading on the system varies. Similarly, the tension typically increases incrementally from the loading section to the point where the material is discharged at the head end of the conveyor.

Any variation in tension changes the actual sag tension on the system. The aforementioned sag tension is a minimum value to prevent material spillage. The actual sag experienced along the length of the conveyor varies according to the actual tension values present in the system at that particular point. It thus makes sense to consider the sensitivity of this variance in sag relative to the force being induced on the idler system.

To illustrate the principle, consider the tension diagram of a conveyor where the sag tension is not the criteria for minimum tension in the system, and drive slip is the consideration.

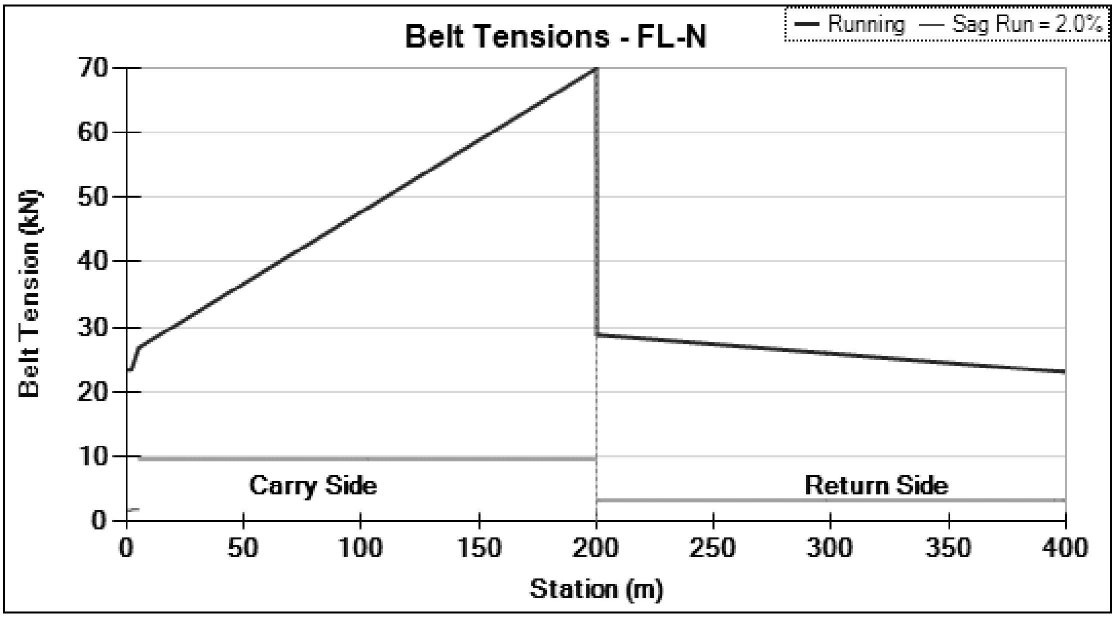

Figure 13. Typical tension graph

From the graph, the lowest tension is in the region of 25 kN with the highest at 70 kN. These values have been entered on a spreadsheet and the results tabled accordingly.

SENSITIVITY ANALYSIS OF SAG TENSION												
Idler pitch:	**1370**	1370	1370	1370	1370	1370	1370	1370	1370	1370	1370	mm
Tension in belt:	9.57	25	30	35	40	45	50	55	60	65	70	kN
Material loading:	**129.73**	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	kg
Belt loading:	**26.4**	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	kg
Idler rot mass:	**14.7**	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	kg
Load excl sag:	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	kg
Percentage sag:	0.0200	0.0077	0.0064	0.0055	0.0048	0.0043	0.0038	0.0035	0.0032	0.0029	0.0027	%
Force angle:	2.29	0.8775	0.7313	0.6268	0.5485	0.4875	0.4388	0.3989	0.3656	0.3375	0.3134	deg
Additional force:	**77.88**	**78.06**	**78.06**	**78.06**	**78.06**	**78.06**	**78.06**	**78.06**	**78.06**	**78.06**	**78.06**	kg
Total force:	**248.7**	**248.9**	**248.9**	**248.9**	**248.9**	**248.9**	**248.9**	**248.9**	**248.9**	**248.9**	**248.9**	kg

Table 3. Sensitivity analysis of sag tension

The calculation conducted in the first column reflects the calculation in accordance with the example. On comparing these values, the variance seems to be minimal. The essence is that once the allowance has been made, the effect along the length of the conveyor seems minimal.

MISALIGNMENT ALSO HAS AN INFLUENCE

The accuracy of installation is referred to in this section. In this instance, it is the height difference between any three idlers in succession. If the acceptable accuracy of installation is 3 mm, the implication is that the in between idler is 3 mm higher than the adjacent two idlers.

Using the same application as before in the example, one idler projects 3 mm above the adjacent idlers on the 1 370 mm idler pitch. In this instance the similar sensitivity analysis is conducted, but due to the constant off-set of 3 mm, the force angle does not change as can be seen in the following table. The 3 mm misalignment equals a percentage sag of 0.22% or a sag factor of 0.0022 as noted in the table.

SENSITIVITY ANALYSIS OF MISALIGNMENT												
Idler pitch:	**1370**	1370	1370	1370	1370	1370	1370	1370	1370	1370	1370	mm
Tension in belt:	9.57	25	30	35	40	45	50	55	60	65	70	kN
Material loading:	**129.73**	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	kg
Belt loading:	**26.4**	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	kg
Idler rot mass:	**14.7**	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	kg
Load excl sag:	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	kg
Percentage sag:	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	%
Force angle:	0.2509	0.2509	0.2509	0.2509	0.2509	0.2509	0.2509	0.2509	0.2509	0.2509	0.2509	deg
Additional force:	**8.54**	**22.32**	**26.79**	**31.25**	**35.71**	**40.18**	**44.64**	**49.11**	**53.57**	**58.04**	**62.50**	kg
Total force:	**179.4**	**193.2**	**197.6**	**202.1**	**206.5**	**211.0**	**215.5**	**219.9**	**224.4**	**228.9**	**233.3**	kg

Table 4. Sensitivity analysis of misalignment

The sag tension consideration was fairly constant but the misalignment force escalates depending on the tension in the belt.

SAG ALLOWANCE AND MISALIGNMENT COMBINED

Due to the fact that the methodology of calculation of the one is identical to the other, it is understandable that the final force can be calculated for both in a single table. The combined table is as below.

SENSITIVITY ANALYSIS OF SAG AND MISALIGNMENT COMBINED												
Idler pitch:	**1370**	1370	1370	1370	1370	1370	1370	1370	1370	1370	1370	mm
Tension in belt:	9.57	25	30	35	40	45	50	55	60	65	70	kN
Material loading:	**129.73**	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	129.73	kg
Belt loading:	**26.4**	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	26.4	kg
Idler rot mass:	**14.7**	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	kg
Material and belt load:	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	**170.83**	kg
Percentage sag:	0.02	0.0077	0.0064	0.0055	0.0048	0.0043	0.0038	0.0035	0.0032	0.0029	0.0027	%
Misalignment factor:	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	0.0022	%
Sag and Alignment:	**0.0222**	**0.0098**	**0.0086**	**0.0077**	**0.0070**	**0.0064**	**0.0060**	**0.0057**	**0.0054**	**0.0051**	**0.0049**	%
Force angle:	2.5411	1.1283	0.9821	0.8777	0.7994	0.7384	0.6897	0.6498	0.6166	0.5884	0.5643	deg
Additional force:	**86.50**	**100.37**	**104.84**	**109.30**	**113.77**	**118.23**	**122.70**	**127.17**	**131.63**	**136.09**	**140.56**	kg
Total force:	**257.3**	**271.2**	**275.7**	**280.1**	**284.6**	**289.1**	**293.5**	**298.0**	**302.5**	**306.9**	**311.4**	kg

Table 5. Sensitivity analysis of sag and misalignment combined

ALLOWANCE FOR DYNAMIC LOADING

In the final analysis the idlers are subject to dynamic loading. This typically occurs when the lumps in the material pass over the idler. This is a difficult phenomenon to quantify accurately and the norm being used in the industry is as noted in Table 6.

In this approach, the fixing of the idlers are considered and an impact value added accordingly. This approach makes an allowance, but it does not distinguish on the actual size and distribution of the lumps. It is the author's opinion that the subject is an ideal area requiring further research with the aim of quantifying the influence on the idler life.

For now it is considered adequate to use the current values as is depicted in the following table as being the norm in the industry.

MATERIAL CONDITION	LUMP FACTOR	
	Fixed Idler set	Garland Idler set
Fine Grained	0	0
Individual small chips	0.005	0
Coarse chips on layer of cushioning material	0.009	0.005
Coarse chips without layer of cushioning material	0.014	0.009
Exclusively coarse lumps	0.05	0.02

Table 6. Current values used by industry

The values in the table are used as a multiplier for increasing the loading on the idler to compensate for the impact loading.

4. CONCLUSIONS

- The correct approach in the design of a conveyor idler is far more complex than it seems.
- The deflection of bearing housings must be brought into consideration where applicable.
- Optimum seal design is of cardinal importance in designing a cost efficient idler.
- Sufficient attention must be paid to ensuring adequate lubricant retention combined with sealing efficiency for preventing the ingress of moisture and contaminants.
- Rollers 'breathe' air in and out through the bearings and seals.
- The fixed and floating bearing principle must be considered when designing an idler.
- Sag tension and the level of tensions must be factored into the design of idler rollers.
- Additional tension due to misalignment under construction conditions must be factored into the design of idler rollers.

5. RECOMMENDATIONS

- A holistic approach must be adopted in the design of idler rollers.
- Current conventions must be challenged in the approach to designing idler rollers and the necessary modifications implemented accordingly.
- Further research is required to quantify the impact of lump size and distribution on idler life.

Although not covered in the discussion of this presentation, research is required to confirm the actual loading distribution on all the configuration idler support rollers used in the industry. This includes single roll, two-roll, three-roll and five-roll idler roller arrangements.

REFERENCES

1 SANS 1313 Part 1 – Troughed belt conveyor idlers (metallic and non-metallic) for idler roller rotational speeds of up to 750 revolutions per minute

2 SANS 1313 Part 2 – Link suspended idlers and fixed form suspended idlers (metallic and non-metallic) for idler roller rotational speeds of up to 750 revolutions per minute

3 SANS 1313 Part 3 - Performance specifications for troughed belt conveyor idlers (metallic and non-metallic) for idler roller rotational speeds of up to 750 revolutions per minute

4 Conveyor Manufacturers Association (CMA) conveyor design course notes

5 ISO 5048 – Continuous mechanical handling equipment – Belt conveyors with carrying idlers – Calculating of operating power and tensile forces.

6 CEMA 6 "6th edition of Belt Conveyors for Bulk Materials"

7 SANS 445 Rolling bearings - Radial bearings - Boundary dimensions, general plan

ABOUT THE AUTHOR

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Simon Curry is the project development manager at Flexco. He studied mechanical engineering and has 34 years of experience designing conveyors specifically for the mining industry. As a conveyor design specialist, he has designed one of the longest PVC booster conveyors in the world at Arnot Colliery (7 300 m) powered by one drive station at the head and two intermediate drives. Other noteworthy achievements are the design of the first 300 metre, 6 lap belt storage and the first driven trippers installed underground in South Africa. The author has thus far registered four patents during his working career.

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