CONVEYOR IDLER TROUGHING PROFILES

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1. INTRODUCTION

There is a modern tendency to trough conveyor idlers at what could be considered steep angles, exceeding the traditionally accepted 45°. This paper considers the perceived advantages and disadvantages of carry idlers troughed at steep angles. An attempt is made to provide industry with some guidance as to what is accepted as good practise, with consideration also being given to different types of belting and the effect on transition distances, material spillage, the life of the idler and roll loading.

Conveyor belting and idlers are often the biggest contributors to the cost of a conveyor, and not taking careful note of the idler configuration and the subsequent influence on the life of the belt may result in a situation where the client unknowingly accepts a system that has the potential to yield an improved 'Total-Cost-of-Ownership' ratio.

Furthermore, this paper considers various idler configurations, and intends to again, to provide industry guidance regarding what could be considered as an optimum configuration. Naturally in such a recommendation, the unit cost would be a significant factor influencing the final selection.

2. COST ANALYSIS OF A CONVEYOR

An analysis or comparison such as this has but one goal, and that is to save money for the client. However, it would be beneficial to quantify what it is one is trying to achieve. For that reason the author has selected four fictional conveyors, and determined how much it would cost to get such conveyors to become operational. The purpose of this exercise was to be able to attach a percentage cost to idlers and belting, since only then could the impact of a possible mistake in making the original selection be appreciated. The values reflected in Table 1 are based on conventional conveyor design and tension calculations. Idler selection was based on 1,5 m spacing, using conventional three roll trough, series 25, diameter 127 mm rolls in a standard frame. For the purpose of the comparison, the return idlers were spaced at 4,5 m crs. The expected wear life [1] was calculated using 6 mm top and 2 mm bottom covers, of grade N [2] and assumes that the carcass will not be damaged throughout this period (however likely or unlikely that may be).

Conv. No.	Length of conveyor (m)	Field cost of conveyor	Belting			Idlers		
			Cost of belting (Rand)	Percentage contribution to overall cost	Expected life (years)	Cost of idlers (Rand)	Percentage contribution to overall cost	Expected life (hours)
1	1 783	**R17 846 807**	R3 942 748	22,09%	41	R2 209 840	12,38%	40 000
2	121	**R 4 379 101**	R 244 445	5,58%	3,3	R 309 049	7,06%	40 000
3	87	**R 2 887 717**	R 172 834	5,98%	2,4	R 236 692	8,19%	40 000
4	392	**R 5 607 225**	R 772 162	13,77%	10,9	R 616 841	11,01%	40 000

Table 1. Cost of conveyors

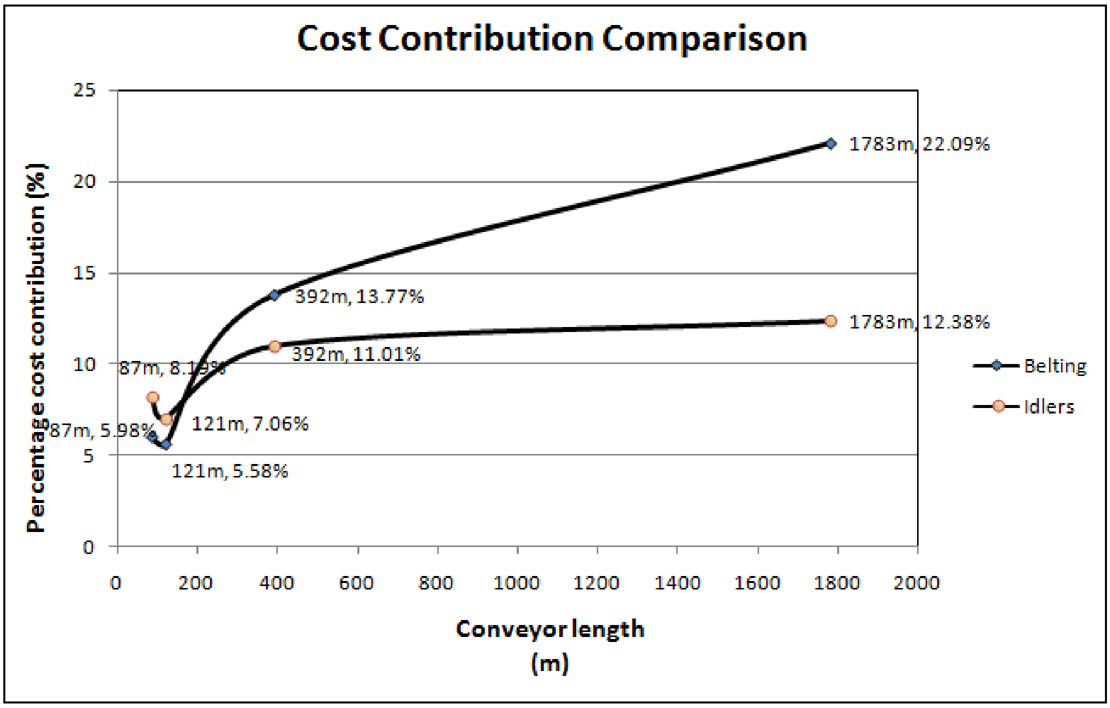

Figure 1. Graph showing difference in cost contribution
The important matter illustrated in Table 1 and Figure 1 is that as the conveyor becomes longer, the percentage contribution made by the cost of the belting becomes greater, given that all the other factors remain constant. It can be said that there is a point where the cost contribution of the belting levels out, but this point has not been determined since it would be different for various belt widths and classes and would be misleading to try to pinpoint.

Furthermore, the expected life of the belting was determined and assuming a life-of-mine of say 40 years, (240 000 hours) the cost of belting was calculated over this period, using current day prices as a basis of comparison.

The scenario now changes and if the cost of the belting is calculated over the life of the mine, the following is found:

Conv. No.	**Expected life (years)**	**Number of belts used over life of installation**	**Total cost of belting over life of installation**	**Cost of belting per metre of conveyor over life of the installation**
1	41	1	R 3 942 748	R 2 211
2	3,3	13	R 3 177 785	R 26 262
3	2,4	17	R 2 938 178	R 33 772
4	10,9	4	R 3 088 448	R 7 878

Table 2. Cost of belting over life of installation
The shortest conveyor, requiring the most belt changes is by far the most expensive over the life of the mine, having the highest Total Cost of Ownership (TCO), whereas the longest conveyor will theoretically last for the entire life of the mine. It is important to note that the above is based purely on the cover wear and does not take carcass fatigue and potential failure into account. It is a purely theoretical estimation of cost, given that the installation is perfectly installed and maintained.

The reader might wonder what all this has to do with deep idler troughing angles. The answer is that if the idler configuration [3] is not optimum, the belt carcass will rapidly fatigue. Should we assume that the belt deterioration is such that the carcass offers only 75% of the life of the covers, due to damage, the cost of the belting will increase resulting in the following. For the purpose of the calculation it was assumed that conveyor No.1 has a steel cord carcass and would be unaffected:

Conv. No.	Expected life (years)	Number of belts used over life of installation	Total cost of belting over life of installation	Cost of belting per metre of conveyor over life of the installation
1	41	1	R 3 942 748	R 2 211
2	3,3	16	R 3 911 120	R 32 323
3	2,4	22	R 3 802 348	R 43 705
4	10,9	5	R 3 860 560	R 9 848

Table 3. Potential cost of belting over life of installation

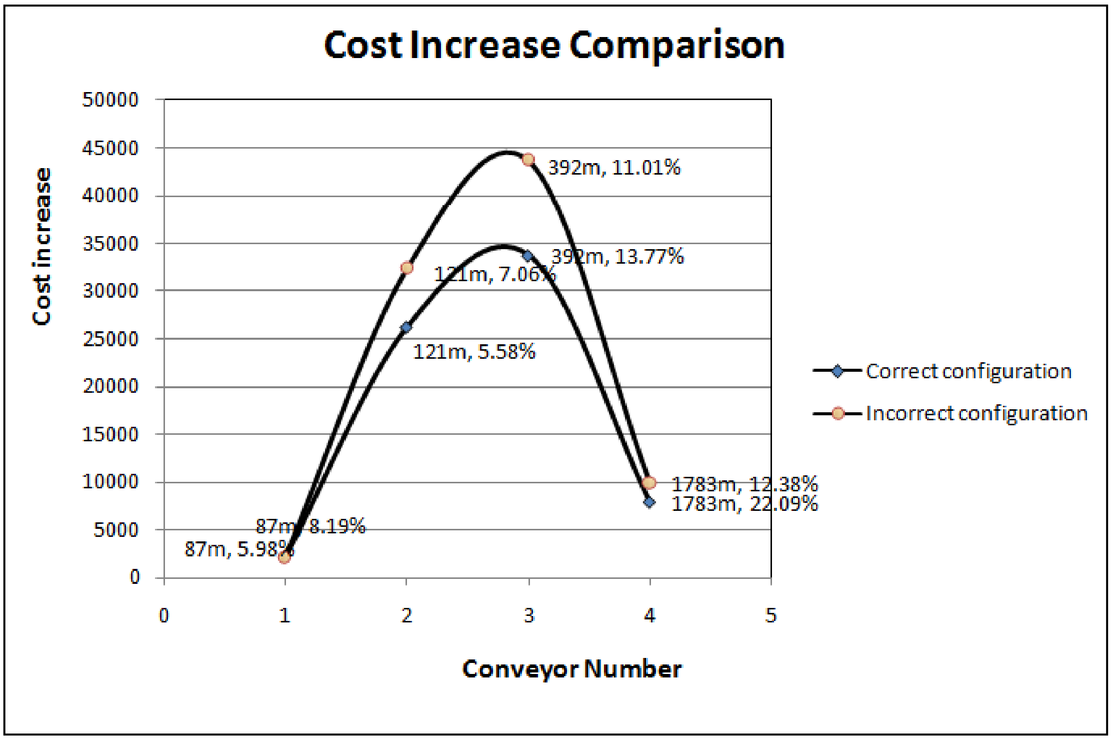

Figure 2. Graph showing belt cost increase

There are certain key areas that would require special attention to ensure that the belt is maintained and belting expenses are kept to a minimum.

The transition distance at the tail pulley – One has to keep in mind that the deep troughing angles as is being discusses is typically only applied at the loading points of the conveyors, therefore, for say the first five to ten metres of the conveyor. This is almost always a relatively low-tension area. There is also a school of thought that believes it appropriate to simply have the conveyor transition distance between the first deep troughed idler and the tail pulley at ten times the belt width. Therefore, a five roll idler set, troughed at 60° on a 1 800 mm wide conveyor, will require a transition distance of 18 m. In essence this would be an extremely safe solution, but the author is of the opinion that it may be somewhat of an overdesign.

- Idler spacing. Idler spacing determines natural belt sag~~,~~ and although the system may be designed for say 1,5% or 2% sag, the actual sag distance should not exceed 50% of the idler diameter. Therefore, should a dia. 152 mm roll be used, the actual sag distance should not exceed 76 mm, irrespective of the actual sag percentage.

- The transition distance at the head of the conveyor. This is probably the more important of the two transition areas, as this is a high tension area. Any transition distance at the head of the conveyor that is too short is likely to lead to overstressing of the belt edges. More about this later.
- Common sense maintenance. There is no substitute for implementing and executing a preventative maintenance plan, as this will allow the user to identify possible future problems and plan a course of action before the possible failure leads to a crisis and losses due to downtime.

3. TRANSITION DISTANCES

The traditional approach for transition distances has always been to ensure that the belt is adequately supported, especially considering that the loading point is typically in a low tension area and that the belt is empty at this point. Following the CMA Conveyor Course and Handbook [4] it is clear that the transition distance, be it at the tail or at the head, is reliant on certain critical parameters:

- The troughing angle. It stands to reason that the greater the troughing angle, the longer the transition distance required
- The belt tension at the point of transition and the belt class. It is the opinion of the author that transition distance in the high tension area should be considered for the maximum tension the belt can handle for the particular class and width. Naturally the transition distance at the tail is likely to be shorter, as the operating tensions are likely to be lower
- The belt service factor and the manner in which it is spliced
- The belt speed.

Should one consider a 1 800 mm wide conveyor, class St1000, troughed at 45°, the transition distance should be calculated for the maximum possible tension in the system, which in this instance would be:

$$\text{Max T} = \frac{\text{Belt class}}{6{,}7} \times \text{W} = 268{,}6 \text{ kN}$$

Where T = tension (kN)
6,7 = belt service factor
W = belt width (mm)

Using the above, following trusted formulae, it can be said that the transition distance for this conveyor should be a minimum of 13,23 m at the head and slightly more conservative if considered in accordance with the CEMA method, at 14,4 m. It has to be noted that the above results are based on ‘full trough’ arrangements, where the pulley shell is in line with the belt line, and not raised to artificially shorten the transition distance. All of this provides an interesting problem, since if idler sets are spaced at three metre centres, it would imply that three transition sets would be required. As idler frames are typically supplied at standard troughing angles, one would be unable to simply divide the transition distance by five in this instance, and install standard frames. The preferred method would be to select the angle of the first transition set, at say 10°, and then calculate the distance it is to be from the centre line of the head- pulley. Then select the next standard frame and again calculate the distance is, trying not to exceed the original centres the idlers for the run of conveyor was selected at. Typically idlers are selected within a particular distance and it may be possible to slightly exceed the originally selected centres, as long as the client is made aware of this and the possibility that the transition idlers may not last the full life of the roller as originally selected.

Of critical importance is that the edges of the belt are not overstressed in the transition area. In the case of ply belts, overstressing the edges may lead to ply separation, which will allow carcass deterioration, reducing the belt life. In the case of steel cord belting, overstressing the edges may lead to outer cords breaking, reducing the overall belt strength and ultimately leading to a belt break.

A secondary indication that the edges of the belt are overstressed at the transition points is if the centre of the belt is showing signs of buckling. In effect, it is the edges of the belt wanting to follow the shortest path due to the tension it is subjected to which then displaces the centre of the belt resulting in the belt buckling upwards.

To conclude this section, the belt speed at which the conveyor operates may be the determining factor as to whether or not a belt will last. Naturally, for the same conveyor, a faster moving belt will perform more revolutions than a slower moving belt, which will determine how quickly the carcass will fatigue as a result of a poor transition distance. This bit of information should serve as nothing more than a motivation to ensure correct transition distances for particular belt speed and tension configurations.

It is furthermore the opinion of the author that transition distances and idler spacings should be standardised for particular belt widths and classes.

4. IDLER CARRYING CAPACITY

Conventional wisdom implies that idler sets are spaced to accommodate the load on each idler roll, the speed of the conveyor belt and also the required bearing design life. Often, for practical reasons, the idler sets are spaced to accommodate gantry dimensions or the span of stringers, which simplifies steel supply.

4.1 Idler Life

The idler design life is determined by the accepted lubrication life. It is commonly accepted that the grease used to lubricate idler bearings shall retain its lubricity and consistency for approximately 35 000 hours. Considering conveyors are typically designed for somewhere between five to seven thousand hours per annum, one could say that the idlers should last between five and seven years. This however, assumes that the idler rolls are uniformly loaded. As is common practice on a three-roll troughed conveyor, the centre roll is designed and the wing rolls are selected to be equal. Although this philosophy is sound and has proven to be reliable in the past, modern tendencies to trough a belt at 45° or even 60° require a different approach.

The modern tendency of overland conveyors is to operate at high speeds. The reason for this is that if the very same conveyor was operating at a lower speed, the tensions in the system would be vastly increased, due to the mass of material loaded on the belt, also known as the ‘Z’ value. Therefore, the slower a conveyor operates, the higher the Z value, at the same capacity per hour. This will typically result in higher tensions in the belt and the pulley shafts. For argument sake, should the speed of the system double, the Z value will be halved. Provided that the capacity is controlled to this value, it does not imply tensions will be halved, but it would improve the tensions in the system, typically resulting in smaller bearings and lower belt class requirements, which in turn will not only save a significant amount of capital during the design and installation of the conveyor but also during operation, as replacement equipment will be significantly less expensive. However, the idlers, especially the centre roll, are often forgotten in this optimisation process.

A conveyor troughed at 35°, loaded to its limit in accordance with ISO 5048 (100% of belt loading), is assumed to have around 67% of the load on the centre roll, as a worst case scenario. However, a conveyor operating at increased speed, resulting in reduced Z value, could easily have 100% of the load on the centre roll. This implies that if the system was originally designed for say 40 000 hours, and idler centres were based thereon, it would actually be under-designed if the percentage load values in the design were not adapted accordingly.

For the reason given, it is critical to the life of the idler roll to ensure that it was designed for the correct loading, taking the belt speed and load profile into account. On the other hand, having a much reduced load on the wing rolls would qualify it to be of a lesser specification, depending on the speed the conveyor is operating.

For the purpose of the paper, a 1 200 mm wide belt is considered, troughed at 45°. Following the Funke method for area calculation, load was determined and approximate values are as follows:

Total load on the conveyor / % capacity	Load on centre roll (%)	Load on each wing roll (%)
100	67,4	16,325
80	73,6	13,2
60	79,0	10,5
50	85,8	7,1
40	92,4	3,8
21	100	0

Table 4. Roll load percentages at varying belt load percentages

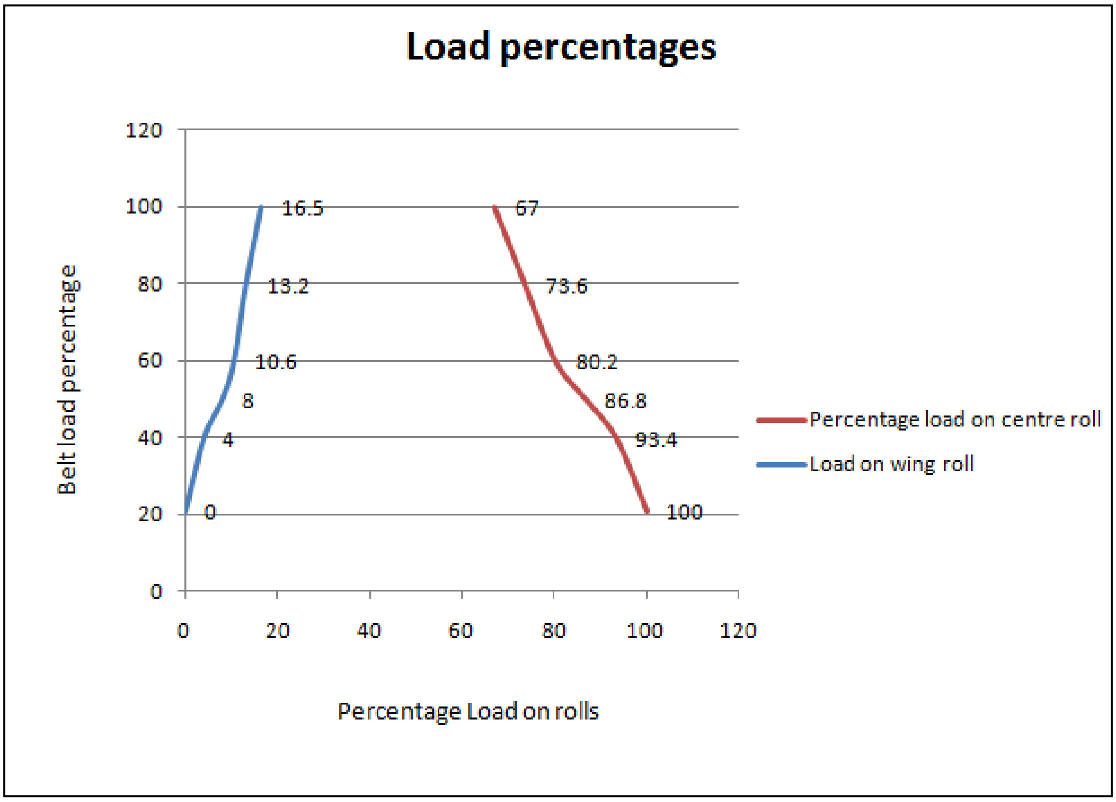

Figure 3. Graph showing roll load percentages

From the above it can be seen that for a 45° troughed idler set, as a worst case scenario, 67% of the load, consisting of the material mass and the belt mass, can be on the centre roll. This is expressed as a percentage of a fully loaded belt and should be the condition for which the centre roll is designed. As the percentage load on the conveyor is reduced, although the load percentage of the overall load is centred more on the centre roll, the resultant load on the centre roll reduces almost linearly. This principle applies whether the belt is troughed at 60° or at 20°.

The most important fact to keep in mind is that if the belt is indeed troughed at 60°, the percentage load on the centre roll will be greater than if the belt was troughed at 35°, and that idlers should be designed accordingly.

5. LOAD PROFILES

Conveyor load profiles should be selected such that they suit the material. Using a shallow trough for a spillage-prone conveyor and material would not be suitable. Using a very deep trough for large run-of-mine lumps may also not be suitable. The following table considers cross sectional areas for three-roll troughed conveyors and could serve as an indication of how different troughing angles will affect the conveyor's load carrying ability. For the purpose of the comparison a three equal roll 35° trough was used as a benchmark. Therefore, the capacity at 35° is expressed as '1'. Other values for other angles are expressed as a percentage thereof. Although calculations were made for a 1 200 mm wide belt, the relationship for other belt widths would be the same:

35° Trough Benchmark	Alternative troughing angle (degrees)	Increase / Decrease in capacity
1	20	-27,8%
1	35	0,00%
1	45	11,8%
1	60	18,3%
1	75	12,2%
1	90*	-3,40%

Table 5. Troughing angle selection

* It is understood that a 90° troughed three-roll unit is not commonly used, although they do exist, but the value was used to illustrate the point.

Figure 4 below illustrates the increase in carrying capacity as the troughing angle increases. It shows that there is a turn-around point at about 60° where the carrying envelope of a troughing angle higher than 60° decreases carrying capacity. Such angles are to be avoided, since they really add no value in terms of belt carrying capability. It puts the wing roll bearings at risk, since it is no longer operating in a plane where it can function optimally as it has to cater for some thrust loading, and it also puts the belt at risk, since there is not a lot of research with belting operating at such steep troughing angles. It can be said that there are many belts operating successfully at a troughing angle of 60°, but 75° would be an additional 25% increase on 60°, and the author would be reluctant to allow such an operating angle without the permission and agreement of the belting manufacturer.

There is a modern tendency in long overland conveyors to decrease the length of the centre roll and increase the length of the wing rolls to distribute the load more equally across the idlers in order to optimise the system. This however, produces a new set of issues to consider and was not considered in this paper [4].

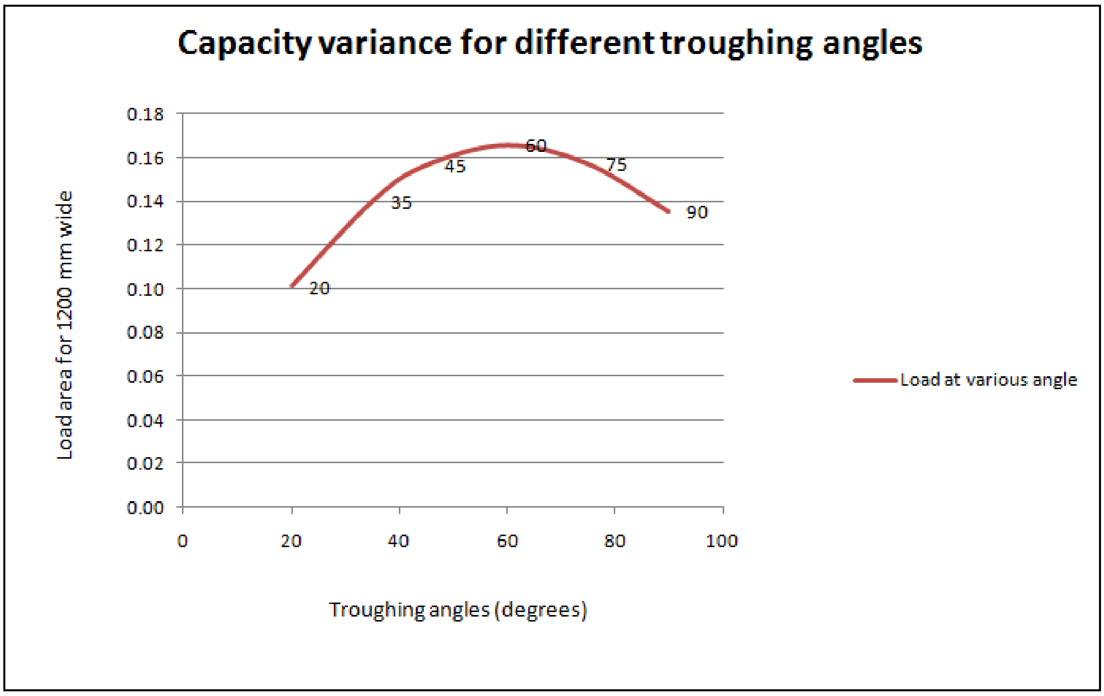

Figure 4. Graph showing load values for various troughing angles
Considering the above, the load on the centre roll will vary accordingly:

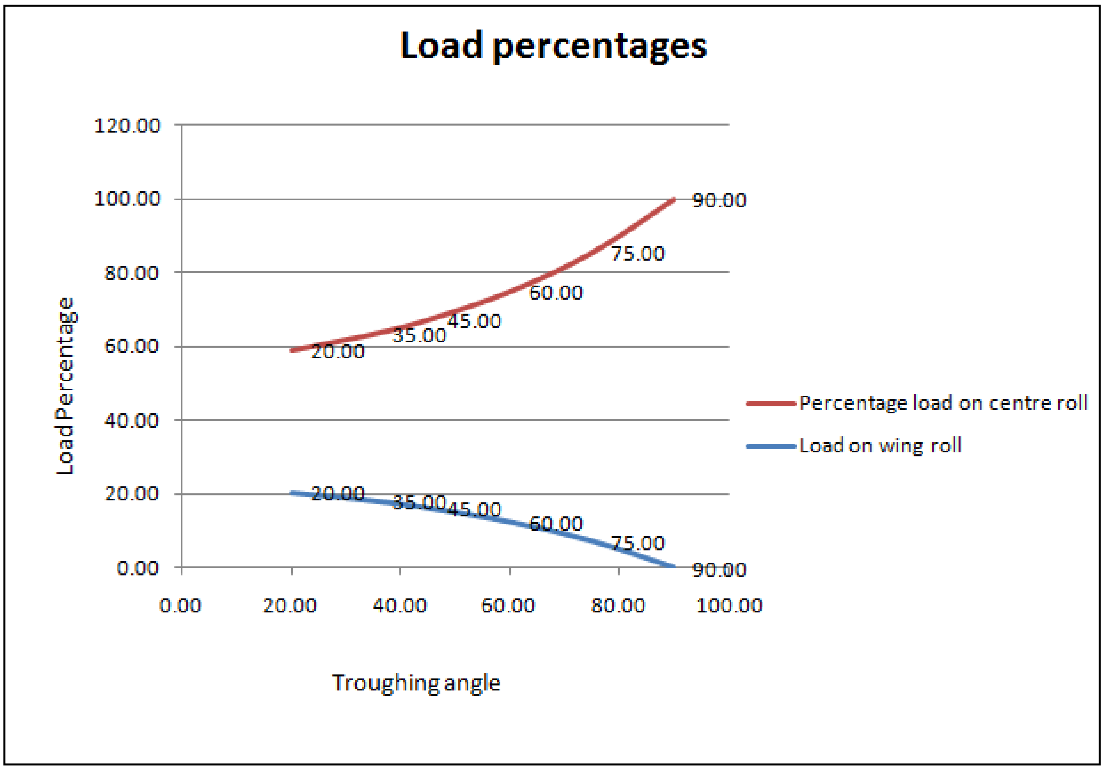

Figure 5. Graph showing roll load percentages at different troughing angles

The graph in Figure 5 indicates that as the troughing angle increases, the load on the centre roll increases accordingly and the load on the wing rolls decrease to the point where the entire load will be on the centre roll. One can therefore conclude that when designing a conveyor, troughed at say 35° for the normal run of the conveyor, idlers at the loading point with a deeper trough should be designed separately and cannot be at the same spacing as the rest of the frames if the same life is expected from the rollers.

6. CONCLUSION

Voices in industry have expressed concern about the modern day trend to trough conveyors at 60° or deeper in loading areas. As the belt is the most expensive piece of mechanical equipment on the conveyor, it is only natural to be concerned about the manner in which it is likely to react to such a deep troughing angle.

Theoretically, provided the transition distance at the tail is adequately designed, and also that the transition distance from 60° to the run-of-conveyor troughing angle is adequate, the belt carcass will not fatigue at an accelerated rate. This view is strongly supported by belting suppliers. So far, under operational conditions, neither the user nor the belting manufacturer has expressed any concern about the method of loading. There have been no signs of excessive belt wear either.

At the other end of the debate stands the idler manufacturer, cautioning users to ensure that idlers are designed not according to conventional methods, but to actual loads. It is again worth reiterating that when designing idlers, especially in the loading area, such idlers should be designed for a 100% of belt load, and the appropriate load proportion should be used for the design and idlers spaced accordingly.

Idler spacing, taking sag into consideration, may have a significant impact on power consumed by the conveyor. Furthermore, indentation rolling resistance of the bottom cover of the belt will certainly impact on the power requirement as well as the tensions in the system. It is therefore important to take a holistic view of the conveyor, and not design items in isolation, hoping that in the end it will fit together like a puzzle.

ACKNOWLEDGEMENTS

Mr Dave Pitcher, Fenner SA, for assisting the author to understand belt carcass fatigue due to too short transition distances

REFERENCES

[1] Life calculation was performed in accordance with CMA Conveyor Course notes.
[2] Grade N – Grade of cover in accordance with South African National standard - SANS 1173
[3] Idler configuration refers to the number of idlers per set, the idler roll and shaft diameter for a specific set and the spacing of such an idler set.
[4] CMA Diploma course in the design and operation of belt conveyors. Chapter 11. Transition distances
[5] Wheeler, C. Bulk solid flexure resistance. Beltcon 13, Johannesburg.

ABOUT THE AUTHORS

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Paul Nel is employed by Bateman Engineered Technologies. He spent 16 years with Anglo American Corporation where he was responsible for materials handling system and equipment design and the execution of various projects within the Anglo group. A year ago he accepted the challenge of becoming part of the 'Bulk' team of BET, also responsible for the design of material handling systems and equipment for the preparation of tender documents. Paul is a member of the South African Institute of Materials Handling, the Conveyor Manufacturers Association of South Africa and member (past chairman) of the International Materials Handling Conference committee responsible for the Beltcon series of conferences. Bateman's is a member of the CMA, represented by Paul Nel, who is an active member of the CMA sub-committees and working groups

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