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The Need for Unified Standards for the
SA Bulk Handling Industry

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THE NEED FOR UNIFIED STANDARDS FOR THE S.A. BULK HANDLING INDUSTRY

1. INTRODUCTION

Every second year everybody related to the bulk materials handling industry in our country, gather at this event, and papers are presented on dynamic conditions, tension waves, belt resonance etc. Delegates then go back, file their papers and promise themselves that some day, when they have an application, they will dig them up again.

If we look at the enquiries issued, and the technical quality of some tenders offered, it is clear that there is a great need in our industry to address design issues at a more basic level.

Buyers often do not realise the implications of their specified requirements, and contractors and suppliers have done very little to work together to produce design guidelines and national standards.

Due to the lack of clear specifications and available standards, for a simple conveyor system, there are invariably as many alternatives offered for belt tensions, drives etc. as there are tenderers.

Gearbox suppliers, for instance, are often required to prepare 10 different quotations for 10 different tenderers, sometimes, even with two or three alternatives for each tenderer. Often, each of these quotations have to be accompanied by completed technical schedules, often in excessive detail and outline drawings.

The industry, with both human and financial resources already very thinly spread, cannot afford this situation, and neither can the buyers and end users afford the risk of uncomparible tenders and over or under designed plants.

The aim of this paper is therefore to illustrate the influence of various and conflicting basic design criteria and specifications, and to indicate the cost and technical quality effects, with specific reference to belt conveyors. At the same time the paper stresses the importance of clearly defined and well researched criteria and specifications to ensure that all parties involved i.e. client, designer and supplier, are using the same basis and have the same understanding of what is required and what should finally be provided.

To reach this objective, it is proposed that a body such as the Institute of Mechanical or Materials Handling Engineers, or the Conveyor Manufacturers Association, or any other new body, is used to co-ordinate all different aspects and attempts to produce acceptable standards.

2. BASIC CONSIDERATIONS FOR BELT CONVEYOR DESIGN

2.1 Basic design calculation methods

There is a need to adopt one single proven calculation basis for the design of conveyors to establish tensions and powers. Methods currently available demonstrate varying degrees of suitability. For example, Table 1 compares typical results from the calculation of three different types of conveyors in actual installations using three calculation methods.

TABLE 1 - COMPARISON OF CALCULATION METHOD

CONVEYOR TYPE	ISO 5048	GOODYEAR	CEMA
Example 1 Inclined Conveyor (1200mm) Length = 1532m Lift = 412m Capacity = 2281 t/h Belt Speed = 4,75m/s	T_1 = 934kN T_2 = 311kN T_e = 623kN Pabs = 2959kW	T_1 = 945kN T_2 = 315kN T_e = 630kN Pabs = 2995kW	T_1 = 912kN T_2 = 309kN T_e = 608kN Pabs = 2888kW
Example 2 Overland Conveyor (1800mm) Length = 4000m Lift = -11m Capacity = 3580 t/h Belt Speed = 4,0m/s	T_1 = 316kN T_2 = 73kN T_e = 243kN Pabs = 975kW	T_1 = 314kN T_2 = 50kN T_e = 264kN Pabs = 1055kW	T_1 = 270kN T_2 = 43kN T_e = 227kN Pabs = 908kW
Example 3 In-plant Length = 196m Lift = 41m Capacity = 550 t/h Belt Speed = 1,54m/s	T_1 = 85kN T_2 = 35kN T_e = 50kN Pabs = 78kW	T_1 = 66kN T_2 = 18kN T_e = 48kN Pabs = 74kW	T_1 = 66kN T_2 = 18kN T_e = 48kN Pabs = 74kW

From the above, it can be seen that all three methods concur in absorbed power for the in-plant conveyor. For the inclined conveyor, where the easily definable lift portion of the calculation is of great significance, the variance in absorbed power is within ± 4%.

The belt tensions for the in-plant conveyors vary significantly because the Goodyear and Cema methods do not bring the start factor for acceleration torque into account. This start factor for in-plant conveyors using fluid couplings is around 1,4, and the take-up mass (T_2 tension) is increased by this amount to avoid slipping during start-up. The necessity of including this factor is clearly demonstrated in practice by the number of take-ups to which additional mass had to be added afterwards.

Many designers have modified the Goodyear and Cema methods to include for these starting factors, which then give better results for in-plant and inclined conveyors, as illustrated by the inclined conveyor example.

The real effect however of the differing design methods can however be seen for the overland example where the variance is ± 8% without any further consideration for profile or dynamic considerations (refer to section 3). Although this difference appears small, it can make a difference to belt classes and motor power selection, adding to the dilemma of evaluating between tenders using different methods.

To overcome this dilema, it is recommended that the ISO 5048 method be adopted, being not only proven numerous times in practice by European and South African designers, but also is extremely adaptable and readily incorporated into in-depth analysis calculations relating to the dynamic considerations. In this way, the situation becomes more streamlined, interpretation of data becomes easier, spreading of technology is facilitated, and adjudication by buyers on a common base becomes achievable.

2.2 Artificial Coefficients of Friction for ISO 5048

The problem facing designers using the ISO method, is the wide range from which friction factors can be selected, namely 0,016 to 0,03. Although certain guidelines are given in this specification, the designer must rely on the operating experience of the end users for guidance, as some of the factors determining the friction coefficients are related to upkeep and maintenance quality, as well as operating conditions, i.e. wet, dry, dirty or clean. If possible, the artificial coefficient of friction should be specified using the operating experience from similar existing plants. Otherwise the expected operating conditions should be stated as clearly as possible.

The sensitivity to the range of friction factors, as applied to the same three examples, is shown in Table 2.

TABLE 2 - COMPARISON OF DIFFERENT FRICTION FACTORS

FRICTION FACTOR	0,016	0,018	0,020	0,022	0,024	0,026	0,028	0,03
Example 1								
Inclined Conveyor								
Pabs (kW)	2886	2925	2965	3004	3043	3083	3122	3162
Pinst (kW)	3000	3100	3200	3200	3250	3300	3350	3370
T_1 (kN)	911	923	936	948	960	973	986	998
Belt Class (kN/m)	5100	5200	5250	5300	5400	5450	5500	5600
Example 2								
Overland Conveyor								
Pabs (kW)	812	925	1039	1153	1266	1380	1493	1607
Pinst (kW)	860	1000	1100	1200	1350	1500	1600	1700
T_1 (kN)	264	301	338	374	411	448	485	522
Belt Class (kN/m)	1000	1250	1250	1600	1600	2000	2000	2000
Example 3								
In-plant Conveyor								
Pabs (kW)	31	32	33	34	35	36	38	39
Pinst (kW)	37,5	37,5	37,5	37,5	37,5	45	45	45
T_1 (kN)	34	35	36	38	39	40	42	43
Belt Class (kN/m)	400	400	400	500	500	500	500	500

The influence of the friction factor is significant in all three examples, and as expected, much more pronounced for the overland conveyor.

In the absence of clear guidelines, and for clean operating conditions and well maintained plants, we have achieved successful results with the friction factors as shown in Table 3.

TABLE 3 - RECOMMENDED ARTIFICIAL COEFFICIENTS OF FRICTION

CONVEYOR TYPE	CARRY SIDE	RETURN SIDE
In-plant	0,022	0,02
Inclined	0,02	0,019
Shiftable	0,024	0,022
Overland*	0,019	0,018

* Tension distribution for overland conveyors must however be further analysed unless reasonably short or of insignificantly varying longitudinal profile. further analysis is also required on overland conveyors with horizontally curved sections. The artificial friction factor is increased by a factor directly calculated from the required training effect of each idler.

2.3 Design Capacity

2.3.1 Introduction

The interpretations of design capacities and volumetric design capacities differ widely between client organisations (buyers), designers, contractors and suppliers, resulting in buyers often receiving non-comparable quotations during tender stages.

It is obvious that if there are misinterpretations at this level, these problems will result in either additional time and money spent on tenders by all contractors, or alternatively in expensive and unnecessary scope changes after award. In the worst case, the client could end up with a plant not meeting the operating requirements, with catastrophic results for clients, designers, contractors and suppliers.

2.3.2 Design Capacity Specifications

Clients often specify the capacity of the conveying system in a variation of definitions, i.e.

i) Capacities

- An average metallurgical flow sheet capacity.
- A maximum flow sheet capacity.
- A normal capacity.
- Peak capacity.
- Maximum capacity.
- A design power capacity.
- Design capacity.
- A flooded belt capacity.

ii) Bulk Densities

In addition to the definition of "capacities" sometimes the bulk density is also varied between a minimum and a maximum.

iii) Moisture Content

Moisture content can vary from unspecified, dry or from a minimum to a maximum specified value.

iv) Design, production or future extension factor

On top of the above, certain clients specify a factor, varying from 1.1 to 1.25, to be used in addition to the above defined factors.

2.3.3 Illustrative Examples

To illustrate the effect and confusion resulting from introducing all these variables, a typical capacity specification often looks like this example from a recent enquiry.

Normal capacity	=	2 000 tph
Volumetric capacity	=	2 200 tph
Power capacity	=	2 500 tph
Bulk density	=	0,85 t/m^3
Moisture content	=	12% maximum
Surcharge angle	=	20°

In addition it is added that all conveyors shall be designed for starting with flooded belts, i.e. material overflowing the belt edge, with no free edge board, and that the contractor shall be responsible for all aspects of the design.

To illustrate the effects of these variations, design can now be based on the following:

Capacity 1	-	2 000 tph (normal - dry)
Capacity 2	-	2 500 tph (2500 tph at 0% moisture)
Capacity 3	-	2 841 tph (2500 tph at 12% moisture)
Capacity 4	-	4 047 (flooded with 12% moisture)

Examples

The results from these different capacities, as applied to a typical example as stated below, are shown in Table 4.

Belt width	-	1 800 mm
Belt speed	-	3,25 m/s
Length	-	500 m

Lift	-	60 m
Trough angle	-	35°/3 roll at 1,5m spacing
Return	-	10° V/2 roll at 3m spacing
Maximum inclination	-	18°
Bulk density	-	850 kg/m³
Convex radius - Tail end to tangent point	-	100 m

TABLE 4 - COMPARISON OF DIFFERENT DESIGN CAPACITIES

DESCRIPTION		Capacity Case 1	2	3	4
Design capacity	(tph)	2 000	2 500	2 841	4 047
Volumetric loading	(%)	77	96	96	137
Tensions: Te	(kN)	200	244	274	381
T_2	(kN)	127	156	176	176
T_1	(kN)	327	400	450	557
Steelcord belt class		1 200	1 600	1 800	1 800
Factor of safety during start-up for static tensions	(F = 1,4)	4,72	5,14	5,14	4,15
Absorbed power	(kW)	488	596	669	857
Minimum installed power	(kW)(x 1,1)	550	656	736	942
Single drive installed power	(kW)	600	700	750	1 000
Drive start-up factor		1,72	1,64	1,57	1,22
Start-up mass	(ton)	194	224	245	320
Start-up time	(secs)	5,5	5,2	5,01	8,57
Accelleration power	(kN)	80	98	110	84
Convex radii to prevent lift-off	(m)	943	1 095	1 253	1 282

2.3.4 Comments on Table 4

Please note from the above the following major differences (cases 2 to 5 only as compared with the normal operating situation - Case 1)

a) Equipment selection

Drastic variation in size of drives (i.e. motors, high speed couplings, reducers, low speed couplings, belting class and, not indicated, the obvious differences resulting from pulleys, plummerblocks and bearings, holdbacks, steelwork, etc.

b) Major differences in the layout resulting from the calculated curves.

c) Actual variations in dynamic conditions. For example, the client may specify a turbo soft start or dynamic refill fluid coupling, specified to be sized on installed power. For proper operating the coupling should be selected for case 2 absorbed power, i.e. 596 kW versus installed power of 700, 750 and 1 000 kW for cases 2 to 5 respectively.

2.4 Volumetric design Criteria

2.4.1 Effect of various Parameters

a) Volume based versus mass based design

The comparative example calculations in section 2.3.3 for different design criteria have pointed out the effects of specifying certain criteria. It is quite obvious that the normal capacity, as for Case 1 in the example, should at least be catered for. By paying careful attention to reasonable feed control, real possible variations in bulk density and moisture content etc., some if not most of the other design conditions could safely be eliminated.

It must be stressed that buyers must pay careful attention that their minimum requirements based on their operating exprience are satisfied but they should at the same time not include unnecessary conditions for "just in case", as the cost of these conditions can be horrendous as illustrated by the example.

b) Bulk density

Bulk density could easily vary by at least 10% between R.O.M. material and fine material. The question always arises whether the stated bulk density is the average, maximum or minimum.

c) Moisture content

Moisture content, within limits, affects mostly power design only and it must be clearly defined whether the moisture content is already included or not in the specified design capacity for design for power.

d) Surcharge angle

The surcharge angle may vary from 0° for very fine, very wet material to 25° for dry, coarse material and is also dependent on conveyor speed, inclination, loading, etc.

Table 5 shows variations in volumetric capacity for a 1 800mm wide conveyor using 35° troughing idlers, expressed in a percentage.

TABLE 5 - EFFECT OF SURCHARGE ANGLE ON CAPACITY	
Surcharge Angle	**% Variation**
25°	100%
20°	92,6%
15°	85,4%
10°	78,4%
5°	71,5%
0°	64,3%

The selection of the surcharge angle is mostly left to the discretion of the tenderer or designer. Different selections lead to large variations in tender data, and if not agreed and checked with practical conditions, and properly specified, conveyors will ultimately be over or under-designed.

e) <u>Material size</u>

Material size affects the volumetric loading of a conveyor directly. For the same surcharge angle, 1mm sized material will provide the possibility of loading the conveyors used in example 2.3.3 to 137% for the conveyor loaded to the belt edges, but in practice the loading of the same belt will be less than 100% for material with a top size of 400mm.

f) <u>Inclination</u>

Inclination of a conveyor reduces the effective cross sectional area capacity of a conveyor belt. The effect is illustrated by Table 6 for a 25° surcharge angle:

TABLE 6 - EFFECT OF CONVEYOR INCLINATION	
Inclination Angle	**Reduction in Cross Area**
0°	1,0
5°	0,99
10°	0,96
15°	0,90
20°	0,82
25°	0,72

In most cases clients, tenderers and designers tend to ignore the above factor, and buyers should ensure these factors are included.

2.4.2 Effects of Loading Methods and Upstream Equipment

The influence of loading methods and upstream equipment is one of the factors most often ignored in the preparation of enquiries, and sometimes even by the designers.

Certain plants or machines do not load the belt conveyors smoothly and continuously. The discharge from gyratory crushers, for instance, is always associated with volumetric surges. The width and speed of belt conveyors must therefore be sized to accommodate these surges without spillage. Often enquiries for conveyor systems are issued as separate packages, not stating how the systems are loaded.

The effect of these volumetric inefficiencies is illustrated by the factors by which the volumetric capacity of a system must be increased. Examples of these factors as specified by certain major clients are:

Apron Feeder discharge	= 1,6
Crusher discharge	= 1,6
Bucket Wheel Reclaimer	= 1,25 - 1,5
Vibrating Feeders	= 1,25

The discharge system from the belt can also have an influence as in the case of tripper cars, where this factor is around 1,4.

This increase in volumetric capacity of tripper cars is mainly as a result of increased capacity of the tripper when running against the stream resulting in increased capacity, the effect of spillage on normally a reduced concave curve, and the inclination factor as discussed in section 2.4.1(f) above.

The major portion of buyers often ignore the above influences. If a tender is based on these allowances the tenderer is penalised (increased cost) and if finally designed without considering these factors, the design will not meet the client's operational requirements resulting in either a reduced system capacity or excessive spillage.

2.5 Pre-tension of Belt Conveyors

2.5.1 Introduction

The three basic factors determining the pre-tension (T_2) on drives are often not clearly specified, or sometimes specified without the necessary research in the conditions expected, or without fully realising the effects of certain criteria.

Some of the basic criteria, and the effects on conveyor design, are discussed in this section.

2.5.2 Start-up factor

This is the factor by which the calculated running pre-tension is multiplied to provide for sufficient pre-tension during starting.

As seen in section 2.1, these factors are not only ignored by buyers and designers, but also by certain widely accepted design methods.

The effects of applying a factor of 1,4 (as used for certain fluid couplings) to absorbed and installed power conditions, are illustrated in Table 7.

TABLE 7 - COMPARISON OF DIFFERENT PRE-TENSION FACTORS

		UNIT	BASED ON ABSORBED POWER		BASED ON ABSORBED POWER X START-UP FACTOR 1,4		BASED ON INSTALLED POWER		BASED ON INSTALLED POWER X START-UP FACTOR 1,4	
			$\mu = 0,3$	$\mu = 0,25$	$\mu = 0,3$	$\mu = 0,25$	$\mu = 0,3$	$\mu = 0,25$	$\mu = 0,3$	$\mu = 0,25$
a) 2% Static Sag Allowable										
Sag tension	T_{o2}	kN	28	28	28	28	28	28	28	28
Pre-tension	T_2	kN	35	47	50	66	46	62	65	86
Tail tension	T_t	kN	43	55	58	74	54	70	73	94
Max tension	T_1	kN	177	189	191	208	188	203	206	228
Steelcord Belt class required		kN/m	659	704	712	774	700	757	768	848
b) 1% Static Sag Allowable										
Sag tension	T_{o1}	kN	56	56	56	56	56	56	56	56
Pre-tension	T_2	kN	48	48	50	66	48	62	65	86
Max tension	T_1	kN	190	190	191	208	190	203	206	228
Steelcord Belt class required		kN/m	707	707	712	774	707	757	768	848

Refer to Appendix 1 for the calculations backing up these figures

From the table it is clear that for this example, the 40% increase in pre-tension, results in a 10% increase in maximum tension and a 30% increase in tail tension. This could mean that certain tenderers can offer lower belt classes because this factor was ignored. To cover himself, the designer then makes provision to increase the counterweight mass during commissioning to avoid slippage. Thus decreasing belt safety factors without concern because belt safety factors are very large anyway. The effect on low tension pulleys however, are usually ignored, while these pulleys are subjected to excessive deflection and overstressing, resulting in largely reduced component lives.

2.5.3 Drive Friction Factors

The influence of two different drive friction factors is also shown in Table 7.

Although the drive friction factor of 0,3 is generally accepted in the industry as a reasonable criteria using lagged pulleys with groove lagging and with reasonably soft rubber, it cannot be accepted for all applications as the correct co-efficient of friction to to the following reasons:

a) The active angle of wrap varies in practice due to final alignment of pulleys.

b) It assumes a 100% torque sharing between the two drives which in practice is impossible.

c) The instantaneous load (T_2) may vary due to friction losses in the take-up or due to dynamic variations from the take-up mass.

d) No difference is generally made for wet and dry conditions.

e) No allowance is made for a wet, dirty belting, especially where a slippery type of material is handled.

f) Clients specify fluid couplings to be installed based on installed power, in many cases in excess of what is required for reasons of drive standardisation, resulting in start-up factors in excess of the normal.

2.5.4 Sag limitations

The influence of sag is also illustrated in Table 7. This table is based on static state (running) conditions, and in the cases where the start factors are used, sag limitation has little if any, influence on running tensions.

Clients and designers sometimes prefer and specify sag tension allowances in excess of 2% based on static conditions. (Some designers and actual installations use up to 5% sag or even more).

Conveyors were seen where the conveyor cannot start-up, under normal loading conditions due to excessive sag between the idlers.

Clients' immediate reaction to this situation is to decrease the percentage sag specified in their tender enquiries without further considerations. This reaction is typical in our industry, i.e. to handle the symptoms and not the real problem.

Reducing the sag tension to 1% will have a minimal (negligible) affect on the power consumption, but will increase tensions, tension and classes, equipment loading, vertical curves, etc. and will not necessarily solve the problem envisaged and experienced by the clients.

The main problem in designing is that clients mostly do not calculate conveyor systems for dynamic start-up, coasting or braking conditions.

This is also applicable to relative short, in-plant, conveyors, especially where stopping time is critical from the safety point of view. Installing large size brakes on conveyors will result in that, under braking conditions, excessive sag will occur resulting in spillage and possible start-up problems and if designed correctly, a potential increase in belt class.

It is of the utmost importance that clients/designers understand the actual problems experienced and that proper dynamic calculation and analysis of conveyors be done and to set criteria which are reasonable for conveyor designs.

2.5.5 General Comments

The above are only a few of the factors influencing the T_2 tension finally selected. It is evident that if these factors are ignored together, the results could be catastrophic.

Our experience in the industry is that usually the T_2 tension is under-designed because of the following reasons:

- Lack of information on the expected operating conditions.

- Tenderers/contractors assuming the best conditions to submit the lowest possible tender price.

To cover the designer/contractor, provision is thus made in the design to increase the counterweight mass (by an oversize box) so that T_2 can be increased during commissioning to prevent excessive creep or slippage.

Such an attitude is not cost effective as ultimately the life of belting and conveyor components is reduced due to increased loads on the conveyor components.

2.6 Equipment Selection Criteria

2.6.1 Introduction

Specifications and requirements for equipment components vary from one extreme to the other, and are in some cases irrelevant to what designers and suppliers usually work to, or what they can cater for.

We broadly touch on the following examples:

2.6.2 Idler Rolls

The most frequently used method of idler design specification is that of hours of bearing life, sometimes with a maximum shaft deflection.

The conditions, however, for which these calculations must be done, are often not given. The theoretical statistical value of 100000 hours bearing life, is often specified without consideration of:

- The load conditions with regard to convex curves, roll misalgnment, stringer deflection etc.
- Shell thickness, wear and shell life
- Derating of the bearings due to the bearing and mounting type and shaft deflection.
- Distribution of load between rollers in a set
- Design of sealing arrangement to match the shell and bearing design lives. Seals have in practice turned out to be the limiting life factor in most applications.

Suppliers and designers still interpret these and other factors to suit themselves to provide the cheapest idler.

2.6.3 Gear Reducers

Most client specifications for reducers, apart from detail specifications regarding efficiencies, bearings, seals, etc. specifications are concentrated around the type of reducer, a service factor of 1,25 based on absorbed or installed power and a L10 bearing life.

Critical aspects, as listed below are often completly ignored:

- Design stresses on input, intermediate and output shafts
- Thermal rating with special reference to calculated temperature rise/operating environment/external cooling/fan cooling etc.
- Gear ratings, factors of safety on gearing, DM or AGMA rating associated to the type of lubrication based on absorbed power and with start-up conditions or start-up factor.
- After all, the most important factor in a reducer is the gearing, which is mostly ignored by clients.

2.6.4 Fluid Couplings

The influence of the starting factors on conveyor design was covered in section 2.5.2. To achieve the starting factor for 1,4 as used in our example, a soft starting device has to be used. Fluid couplings are currently the most widely used soft starting devices used in our industry.

Buyers often specify that the installed power of conveyors be up to 25% more than the minimum calculated motor power. In addition, certain standard motor sizes are specified for standardisation. These specifications do not necessarily create the problems. The problem comes in when the specification requires the fluid couplings to be sized on the installed motor power.

Sometimes the installed motor power could be so far in excess of the calculated absorbed power, that the selection of a constant fill fluid coupling to limit the starting torque, and at the same time dissipate the heat generated through the oversize drive, becomes impossible.

This again illustrates that specifications should be prepared carefully taking all consequences into consideration, and more important, that bigger is not necessarily better.

3. SPECIAL CONSIDERATIONS FOR MORE SIGNIFICANT AND COMPLEX INSTALLATIONS

3.1 Introduction

Complex conveyor installations such as very long overland conveyors of undulating profile, long inclined conveyors with very high belt tensions, and shiftable systems, need to be analysed in a greater depth.

It is not the object of this paper to discuss these considerations in detail, but for the sake of completeness, some of these considerations are discussed broadly.

3.2 Tension Distribution Considerations

3.2.1 Normal Running Conditions

The belt profiles typically of overland conveyors, have great bearing on their operation.

Due to the numerous combinations of load cases for undulating profiles, it has become normal to specify suitability for the "worst load condition". The problem here is that the true situation for the actual "worst load condition" can be missed unless the designer is aware of the possible pitfalls and adopts the correct approach.

In the case of undulating overland conveyors, additionally the power should be calculated for the inclined sections only loaded, for all cases where it is possible to occur in practice, as well as for the declined sections only loaded, and all possible permutations to determine the absorbed and regenerative power requirements.

This will give the range of power requirements and an indication of belt tensions. It is mandatory that not only the maximum, but also the minimum tensions then be checked for all positions of elevation change as well as the terminal points. Having determined the minimum tension in the system based on normally calculated slack-side tension at the drive, it must be compared with the minimum allowable tension in the system, and if below this value, the difference must be added to the tension distribution, thus deriving a new maximum tension.

This process must be repeated for all the possible load conditions as will become apparent especially in the case of long conveyors with multiple and significant changes in elevation along the route.

Straight inclined shaft type conveyors are often significantly affected by the sag requirement as well. For example, it is convenient for designers to consider only the naturally higher tensions at the head end for the drive requirements which often leads designers to overlook the fact that the tail tensions may be totally inadequate and again additional tension must be added to the entire tension distribution.

For the sometimes very large number of permutations and combinations of running conditions and subsequent tension adjustment, the situation lends itself readily to computer analysis. Indeed this approach is almost mandatory for systems which require optimisation, for example for long overland systems.

This approach should also be applied to coasting or braking conditions, and the tensions again increased accordingly.

.2.2 Starting and Stopping

Dynamic considerations relating to coasting

Probably one of the most significant and yet most overlooked aspects of overland conveyor design is that of coasting. Sometimes designs are offered based on simplistic approaches with the qualifier that "in the event of an order, implications relating to dynamic influences will be investigated". How can any meaningful adjudication be done when it is possible that the criteria of greatest importance has been temporarily ignored?.

This importance is best illustrated by Example 2 in Section 2.1, an actual installation designed by the writers, which is approximately 4km long with a slight overall decline giving a lift of -11m. The profile however, is such that the lowest point along the conveyor is approximately -41m. As a result, the conveyor is in two main sections, the first from the tail to the mid point with a negative lift of -41m, and the second from mid point to head with a positive lift of 30m. Because of the differing lifts, there is a marked tendancy for the second section to coast to a halt sooner than the first half. Consequently, there is a tendancy for the belt to collapse and buckle at the lowest point of the carrying run. In practice, this was overcome at the design stage by the significant addition of tension to the slack-side such that a stronger and more expensive belt was required, thus affecting the capital cost of the conveyor.

Dynamic considerations relating to transient stress.

Again it is necessary, especially, for example, in the case of significantly inclined shaft conveyors with steelcord belts, to analyse the dynamic transient stresses occurring in the system. The reason for this is basically two-fold, i.e.:

i) To predict the maximum instantaneous tensions due to the arrival of additional transient tensions at all pulleys.

ii) To predict and quantify the actual maximum belt tension so that the required safety factor based on normal maximum tensions can be rationalised.

In the case of the inclined conveyor given as example 1 in section 2.1, again an actual conveyor designed by the writers, and currently being commissioned, the dynamic transient stresses were calculated using the theory as published by Dr.Harrison. This installation is actually a good example where the client was fully aware of these conditions, and not only requested analysis, but also ensured that the tenderers had the necessary technical capability.

Using a computer, the dynamic stresses for the different degrees of loading were analysed. As a result, the safety factor was calculated to be a required value of 5,5. This is somewhat less than the usual value of 6,7, but the implications with respect to the cost of the belt were quite significant and 5,5 was accepted.

One therefore draws the conclusion that to standardise on a factor of safety of 6,7 for all cases would be incorrect. However, calculations show that one hardly ever has to exceed 6,7 and it is therefore considered wise to adjudicate on this value. Thereafter, any cost saving via the reduced safety factor can be passed directly onto the owner.

Brakes

The use of brakes in conveyor design to reduce stopping time is normally considered (rightly or wrongly) for one of the following reasons:

- to prevent "running away" of regenerative conveyors
- to comply with safety regulations
- to change the coasting characteristics
- to prevent run back on conveyors where the need for holdbacks is marginal

Often, specifications read: "all conveyors shall be fitted with brakes" which appears on the surface, be a reasonable and sound concept. In practice however, brakes are often poorly maintained and the designs must therefore be such that conveyors can operate adequately without them. As such, their value to the system decreases, and where their effect is mandatory, other contingencies must be employed (e.g. downhill conveyors).

It has been found that the use of brakes is generally not practical on overland and similar conveyors for reasons of safety or to reduce the stopping period for other reasons, e.g. the carry over of material. It is far better to design the chutes to accommodate such carry over and to design overall systems for proper coasting compatibility.

3.3 The use of Automatic Take-up Winches

Belt conveyors are often specified or designed with automatic take-up winch systems, without consideration of the possible effect on tension distribution.

This can be demonstrated by the example of a long conveyor with a head drive station. Whereas a normal counterweight system ensures a constant slack-side tension at the head end and a near constant and often similar tension at the tail, the winch detracts significantly from this.

Consider the case where a motorised winch applies pre-tension to the conveyor at rest. The conveyor is started, and if fully loaded, more slack-side tension may be required and applied. This maintains a reasonable low tension on the slack-side and tail pulleys during the starting.

The problem arises immediately when the complete belt reaches full speed. The additional tension generated by accellerating the belt ceases to exist, and the belt tries to shrink accordingly. As a result, the slack-side tensions increase, often significantly.

More importantly however, is when the systems trips before reaching full speed. Here, not only is the above additional tension lost, but also the whole of the effective starting tension as well, with even higher slack-side tension increases.

4.0 EQUIPMENT STANDARDS

4.1 Introduction

The lack of direction from the industry, and the ineffectiveness of the current associations and other bodies, are best illustrated by the following examples.

4.2 Idlers

The only standard that currently officially exist, is the SABS 1313 specification for rollers and brackets. This is purely a dimensional specification which make idler rolls and brackets interchangeable.

There is currently no national standard for design information regarding recommended loading, seal and bearing configurations, rotating masses and bracket design for strength, deflection and vibration. The current standard also does not extend to wider belts which are coming more frequently into use. Garland (suspended) type idler configurations are also not included in this specification.

4.3 Conveyor Pulleys

Conveyor pulleys are even further behind in the sense that there is no dimensional specification making pulleys interchangeable in the industry. It is currently left to designers to determine pulley diameters, face widths, bearing centres, end disk design and shaft turndowns, provided certain minimum criteria are met.

Again, the whole industry can benefit by introducing certain standards.

End users can benefit by greater interchangeability between pulleys supplied on different projects purchased from different contractors, while suppliers can benefit by more streamlined production lines and smaller spares keeping.

It is realised that large variations in powers and tensions make compiling of a standard including all aspects extremely difficult. This does not mean that certain preferred pulley diameters, face widths, bearing centres and shaft sizes cannot be listed.

5. CONCLUSION

This paper used only a few of a great number of examples to illustrate the possible short-comings of enquiry specifications, and in doing so, has shown how some of the "tricks of the trade" can be employed.

The following appeals are therefore made:

5.1 To the Buyers/End Users

The general appeal to the buyers and all people involved in the preparation of specifications, is to carefully review and research their requirements, and then specify them clearly. Also buyers are requested to guard against over specifying, and introducing clauses to cover themselves or for the sake of increased service factors.

5.2 To the Contractors and Suppliers

The challenge to contractors and suppliers is to work together to produce meaningful industry standards and design guidelines without suppressing technological advancement, design ingenuity and enterpreneurical skills.

6. REFERENCES

6.1 Internation Standard ISO/DIS 5048;
Continuous Mechanical Handling Equipment - Belt conveyors with carrying idlers - Calculation of operating power and tensile forces.

6.2 Goodyear Handbook of Conveyor and Elevator belting (Metric edition).

6.3 Conveyor Equipment Manufacturers Association (CEMA);
Belt Conveyors for Bulk Materials.

APPENDIX 1 - EXAMPLE CALCULATIONS

CALCULATION FOR SAG - FRICTION AND START FACTORS

Conveyor Design Data

Design capacity	=	2 500 tph
Width	=	1 800 mm
Volumetric loading	=	85%
Surcharge angle	=	20°
Lift	=	20 m
Belt speed	=	2,6 m/s
Bulk density	=	0,85 t/m^3
Idlers	=	3 roll, 35°

Results of Basic Tension Calculations

Te	=	141,792 kN
Terf	=	14,605 kN
Ter	=	12,53 kN
Tem	=	62,257 kN
Te lift	=	52,40 kN
P absorbed	=	400 kW
P installed (minimum)	=	1,2 x 400 (client specification)
	=	480 kW
Actual installed	=	2 x 250 kW drive at head end

Sag Tension Calculations

For static belt conditions:

Slag tension at 2% maximum

Trough side - L	=	1,5 m
thus T_{c02}	=	6,25 x 1,5 x 9,81 (267,10 + 36,43)/1000
	=	27,915 kN

Return:

$T_{RO2} = 6{,}25 \times 3 \times 9{,}81 \times 36{,}43/1\ 000$

$= 6{,}701$ kN

For 1% sag:

$$T_{CO1} = \frac{1{,}5 \times 9{,}81 \times (267{,}10 + 36{,}4)}{8 \times 0{,}01 \times 1\ 000}$$

$= 55{,}831$ kN

$$T_{RO1} = \frac{3 \times 9{,}81 \times 36{,}43}{8 \times 0{,}01 \times 1\ 000}$$

$= 13{,}42$ kN

4. Drive Friction Factor Calculations

Assume automatic gravity take-up

μ	=	0,3	=	0,25
Angle of wrap	=	210°	=	210°
thus T_1/T_2	=	$e^{\mu\Theta}$	=	$e^{\mu\Theta}$
$T_1 - T_2$	=	Te	=	Te
thus T_2	=	$\frac{Te}{e^{\mu\Theta-1}}$	=	$\frac{Te}{e^{\mu\Theta}-1}$
	=	$\frac{Te}{3{,}033 - 1}$	=	$\frac{Te}{2{,}5 - 1}$
	=	$\frac{Te}{2{,}003}$	=	$\frac{Te}{1{,}5}$

5. Pre-tension Calculations (T_2)

Based on absorbed power $= \frac{141{,}792}{2 \times 2{,}003} = \frac{141{,}792}{2 \times 1{,}5}$

$= 35{,}40$ kN $= 47{,}264$ kN

Based on start-up factor
of 1,4 from fluid couplings

thus $T_t1 = \frac{141{,}792 \times 1{,}4}{2 \times 2{,}003} = \frac{141{,}792 \times 1{,}4}{2 \times 1{,}5}$

$= 49{,}6$ kN $= 66{,}170$ kN

Based on installed power:

$$T_{2}2 = \frac{500 \times 0{,}96}{2 \times 2{,}6 \times 2{,}003} = \frac{500 \times 0{,}96}{2 \times 2{,}6 \times 1{,}5}$$

$$= 46{,}1 \text{ kN} \qquad = 61{,}538 \text{ kN}$$

Based on installed power with
start-up factor of 1,4 for
fluid coupling

$$T_{23} = \frac{500 \times 0{,}96 \times 1{,}4}{2 \times 2{,}6 \times 2{,}003} = \frac{500 \times 0{,}96 \times 1{,}4}{2 \times 2{,}6 \times 1{,}5}$$

$$= 64{,}6 \text{ kN} \qquad = 86{,}154 \text{ kN}$$