STANDARDISING CONVEYORS - FACT OR FICTION

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INTRODUCTION

Standardising is always an issue on the majority of projects. The only variance is the extent of standardisation that is required per the contract.

If the aforesaid is true, then why is this an issue and why do we have the range of equipment available on the various sites? The more correct statement is probably not the various sites but the same sites.

The purpose of this paper is then to reflect on this issue and advise on ways forward to resolve the issue.

ASSUMPTIONS

This paper has been developed from knowledge and observations limited to the South African mining industry only. As an extension to this it was accepted that the reader is familiar with the common terms and basic design approaches used in the local industry.

DISCUSSION

COMMON GROUND

The point of departure has to be, but where does one start? In any design approach, be it ISO, CEMA or others, the first objective is to size the belt, then the drives and then the pulleys and somewhere in there are the idlers and all the other ancillary equipment.

The design criteria call for maximum standardisation of equipment on the project. Historically, these are words that consultants use to impress their client in an effort to give them that warm fuzzy feeling that their interests are being looked after. Wrong, this is where the game begins.

Depending on whether the point of departure is a lump sum turnkey or engineer procure and supply project, the approach differs.

If it is lump sum turnkey, the competing company will endeavour to sharpen the pencil in order to win the project: he must beat the opposition on price. The net result is that the most competitive capital cost is being chased with little or no regard for overall cost of ownership. Thus if a 90 mm bearing will do the job, then it will be offered and not the 110 diameter which should be the option for standardisation.

For engineer, procure, construct and project management (EPCM) projects the approach is different, or is it? The consulting company will still issue enquiries to the various suppliers for the supply of the conveyor package and then unfortunately the same scenario develops as before. In some instances the situation will be that in house designs are undertaken and then the possibility of implementing a larger degree of standardisation is feasible, provided the vision and foresight is in place.

In the real world the client ends up with various permutations of pulleys and drives in his stores for the same belt width, and that becomes a nightmare to control and manage. The readers who are exposed to this environment will nod in agreement that on many occasions, when a pulley needs to be replaced underground, and you communicated to the stores more than once exactly where the unit is, the incorrect one is still transported underground. The loss of production, and in this case unnecessary loss, leads to extended downtime periods and extreme levels of frustration.

Clearly there are these two considerations. The one is capital and the other operating costs. The common ground for moving forward lies in the area that total cost of ownership must be considered as the basis. Higher capital costs are acceptable against the background of lower operating costs and higher system availability. Incidentally, it was at the BELTCON forum where the question was asked what is an engineer? The answer is simply someone that can design something for 10 cents which any fool can design for one rand.

CONVEYOR DESIGN APPROACH

The thought processes must now be refocused on designing the so called "piece of elastic band". The belt is the first piece of equipment that is sized for the application. The crux of this paper is that everything must now be sized for the application accordingly.

The following questions are frequently asked on existing installations.

- What would happen if the system capacity is increased at the same belt speed?
  - The strength of the belt must be checked for adequacy.

- How far can this belt be extended?
  - The strength of the belt must be checked for adequacy.

What is not asked is what happens to the pulley and the bearing if the system capacity is increased and this is mostly where the problem lies.

If the pulleys had been sized in accordance with the belting requirements, this will not be a problem. This is best illustrated by means of an example.

Consider 900 mm belt width class 1000 belting with a safety factor of 10. The next class up is 1250 and the class down is 800.

For this 900 wide belting the maximum working tension will thus be:

For class 1000:

- Maximum allowable working tension = 900 mm ÷ 1000 mm x 1000 kNm ÷ 10 fos
  = 90 kN

For class 1250:

- Maximum allowable tension = 900 mm ÷ 1000 mm x 1250 kNm ÷ 10 fos
  = 56,7 kN

For class 800:

- Maximum allowable working tension = 900 mm ÷ 1000 mm x 800 kNm ÷ 10 fos
  = 72 kN

The proposal is then, assuming the design tension requirement is 80 kN, that the pulley will be sized in accordance with the 90 kN maximum belt tension requirement. It follows then that for any, and that literally means any, application per the design criteria, where the class 1000 belting will be used, the equivalent class 1000 belting pulley will be used accordingly in the standardisation approach.

The immediate observation will be that the pulley is now oversized for the specific application. Technically this is true but practically not. Due to the fact that this pulley will be subjected to a loading level probably less than what was designed for, technically the life expectancy of the unit is increased accordingly. Increased life expectancy is directly associated with less downtime thus resulting in a more reliable production unit. The implications of this in the bigger picture will be mentioned later.

The fun really begins when one looks at the T2 tension requirements.

All conveyor designers distinguish between single and dual drive pulley arrangements. On dual drive pulley arrangements cognisance is taken of 1 to 1, 2 to 1 and 2 to 2 drive arrangement configurations and then not to forget about the head drive pulley arrangement and all its permutations. In addition, the actual drive pulley now has the potential to be bare steel, plain rubber, grooved rubber of the chevron type, grooved rubber of the diamond pattern type, ceramic tiles and ceramic paste. The matter of the plain and or lagged with whatever type of material must not be used to cloud the issue. The principle is that on any site the design criteria will specify what the drive pulley surface needs to be. On multiple sites the aforesaid is critical for establishing the common denominator when large consortiums or groups are developing and using ore clearances systems. For purposes of this paper diamond grooved rubber lagging will be considered as the option for all the conveyor applications. Where other materials need to be considered the standardisation method needs to be adapted accordingly, and this is stating the obvious.

The industry norm is to use a friction factor typically in the region of 0,35 for this type of lagging. This value is used to derive the wrap angle required for any specific application. It is not the intention to elaborate on the derivation of formula but rather to reflect on the typical range of values applicable to the range of the application.

The standard formula used to derive the friction factor for belt drives is:

$$\frac{T1}{T2} = e^{\mu\varphi}$$

and the relationship T1 = Te + T2

and $$K = \frac{1}{e^{\mu\varphi} - 1}$$

Where: **e** = 2.8713

μ = friction factor referenced in text

φ = wrap angle in radians

T1 = maximum belt tension used to determine belt class

T2 = derived take up tension to ensure no slip during driving

Te = effective tension value required to overcome system resistances

K = drive friction factor for calculation purposes

The norm would be to specify a minimum wrap angle for purposes of ensuring positive drive. The wrap angle would typically vary from 180 degrees (not recommended) up to 210 degrees (very difficult to achieve in practice). The range of values would then typically be:

Calculated friction factors for the aforesaid using a friction value of 0.35 will be:

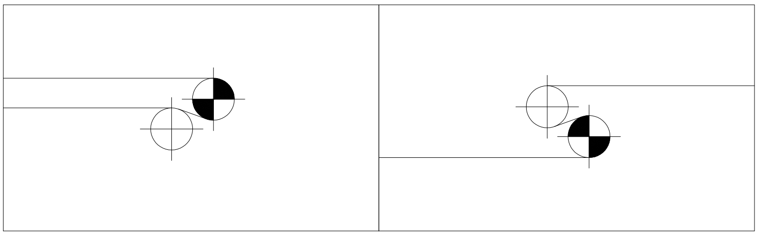
180 degrees K = 0.50	thus	T1 = Te x 1.50
185 degrees K = 0.48	thus	T1 = Te x 1.48
190 degrees K = 0.46	thus	T1 = Te x 1.46
195 degrees K = 0.44	thus	T1 = Te x 1.44
200 degrees K = 0.42	thus	T1 = Te x 1.42
205 degrees K = 0.40	thus	T1 = Te x 1.40
210 degrees K = 0.38	thus	T1 = Te x 1.38

The target range worth considering will be between 190 to 200 degrees. Expressed in terms of values it ranges from 0.46 to 0.42. This is not a big difference but that is not the point. The issue is that there is merit for an organisation to select a minimum requirement friction factor and then for the design to move forward based on this requirement. The purpose of the table is then to illustrate the sensitivity of the values over a practical range.

The principle is to err on the conservative side and always have what can be classed as a safe design for all applications within the set parameters. When working with steel cord belting, relaxation distances are relatively long and one would rather use wrap angles in the region of 190 degrees while it would be perfectly acceptable to use wrap angles of 200 degrees for ply type belting due to correspondingly shorter relaxation distances.

All this is fine, but the various drive configurations need to be brought into consideration as well. Typically, in 98 percent of all conveyor drive systems comprise single and dual drive pulley arrangements. The dual drive pulley arrangement can be subdivided into equal power on the primary and secondary pulleys. The final permutation will be three equal power units with two on the primary and one on the secondary drive pulley. All of these applications result in different tensioning requirements to prevent slip requirements.

To illustrate the point, for purposes of this paper the assumption is made that the wrap angle will be 200 degrees with a resulting friction factor of 1.42. In line with this 900 wide belting of class 1000 is used as before.



Single Head Drive Pulley **Single Intermediate Drive**

Figure 1: Single head drive pulley and single intermediate drive

On a single drive pulley arrangement the following applies: (excluding belt slope tension)
Single drive pulley application

Example for slip: When the T2 drive slip requirement is the governing factor.

- Maximum T1 tension = class 1000 kN/m ÷ 10 fos x 900mm ÷ 1000mm
  = 90 kN

- Maximum Te tension = 90 kN ÷ 1.42
  = 63,3 kN

- Maximum T2 tension = T1 – Te
  = 90 kN – 63,3 kN
  = 26,7 kN

Note: The sag tension must be calculated and checked against the minimum T2 requirement. If necessary the sag tension must then be used as the basis and the effective tension reduced accordingly. Although this is a definite requirement that the designer must consider, it will overcomplicate matters for illustrating the principle approach for standardisation and is conveniently accepted as being less than the T2 consideration for slip.

Example for sag:

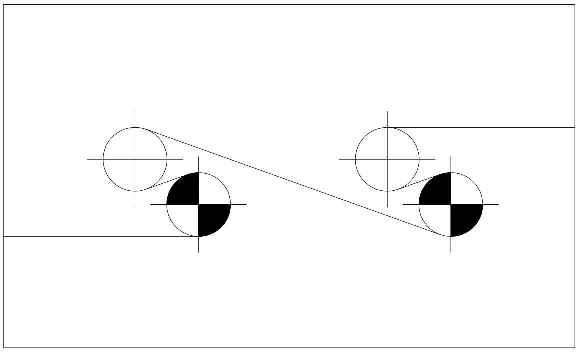
If the minimum sag tension requirement was 30 kN. The application will then be modified as follows:

- Maximum T1 tension = class 1000 kN/m ÷ 10 fos x 900mm ÷ 1000mm
  = 90 kN
- Maximum T2 tension = 30 kN
- Maximum Te tension = T1 – T2

  = 90 kN – 30 kN
  = 60 kN

The same approach must be used for multiple drive pulleys and number of drive units installed. In all instances the T1, T2 and Te tensions are calculated for the application.

On a dual drive pulley arrangement the following applies: (excluding belt slope tension)



Dual Drive Pulley Arrangement

Figure 2: Dual drive pulley arrangement

Dual drive pulley application with 1:1 power distribution.

Using the same analogy as before, let us review wrap angles for dual drive application with 1:1 power distribution:

Calculated friction factors for the aforesaid using a friction value of 0.35 will be:

360 degrees K = 0.25	thus	T1 = Te x 1.25
370 degrees K = 0.24	thus	T1 = Te x 1.24
380 degrees K = 0.23	thus	T1 = Te x 1.23
390 degrees K = 0.22	thus	T1 = Te x 1.22
400 degrees K = 0.21	thus	T1 = Te x 1.21
410 degrees K = 0.20	thus	T1 = Te x 1.20
420 degrees K = 0.19	thus	T1 = Te x 1.19

Using 400 degree wrap angle which is twice that used before.

Example for slip: When the T2 drive slip requirement is the governing factor.

- Maximum T1 tension = class 1000 kN/m ÷ 10 fos x 900mm ÷ 1000mm
  = 90 kN
- Maximum Te tension = 90 kN ÷ 1.21
  = 74,38 kN
- Maximum T2 tension = T1 – Te
  = 90 kN – 74,38 kN
  = 15,62 kN

Dual drive pulley application with 2:1 power distribution.

Using the same analogy as before, let us review wrap angles for dual drive application with 2:1 power distribution:

Calculated friction factors for the aforesaid using a friction value of 0.35 will be:

360 degrees K = 0.17	thus	T1 = Te x 1.17
370 degrees K = 0.16	thus	T1 = Te x 1.16
380 degrees K = 0.15	thus	T1 = Te x 1.15
390 degrees K = 0.15	thus	T1 = Te x 1.15
400 degrees K = 0.14	thus	T1 = Te x 1.14
410 degrees K = 0.13	thus	T1 = Te x 1.13
420 degrees K = 0.13	thus	T1 = Te x 1.13

Using 400 degree wrap angle which is the same as the previous example.

Example for slip: When the T2 drive slip requirement is the governing factor.

- Maximum T1 tension = class 1000 kN/m ÷ 10 fos x 900mm ÷ 1000mm
  = 90 kN
- Maximum Te tension = 90 kN ÷ 1.14
  = 78,95 kN
- Maximum T2 tension = T1 – Te
  = 90 kN – 78,95 kN
  = 11,05 kN

In summary the results from examples are then as follows:

Application	Maximum T1 tension High tension pulleys	Maximum Te tension Available for driving	Required T2 tension Low tension pulleys
Single drive pulley	90 kN	63,30 = 1 x 63,30 kN	26,70
Dual drive pulley 1:1	90 kN	74,38 = 2 x 37,19 kN	15,62
Dual drive pulley 2:1	90 kN	78,95 = 3 x 26,32 kN	11,05

Table 1: T1, Te and T2 tensions for various drive pulley configurations

Interesting observation is that the tension values of the high-tension pulleys are the same across the range.

The low tension pulleys vary the most. Interesting to note is that T2 is the highest for the single drive pulley application, then the dual drive with 1:1 power sharing and dual drive pulleys with 2:1 power sharing the least. It follows then that the lower T2 tension pulleys can be used for the higher tension applications but not vice versa. Care must be taken when making the decision at this point in time as the system may become oversized purely from a standardisation perspective. The overall application needs to be reviewed with respect to the T2 requirements..

There is a further observation relative to being able to use the bigger pulley in place of the smaller pulley. By using the somewhat larger pulley for the lighter duty, the life expectancy of the unit will also increase. This leads to marginally more reliable installations with possibly less downtime for maintenance as the replacement period is increased. The point that needs to be made is that the life expectancy of the "oversized" unit will not be lost in the application but will lead to enhancing the reliability of the system.

For projects in most cases the expectations of the end user's production personnel are hardly ever considered. At the start of any project, the timeline and capital project costs are the prevailing issues. As the project nears completion there is a complete paradigm shift from providing the basic system to a system that must be "100 percent reliable" at all costs that will last "hundreds of years". The case of buying the Beetle but expecting the Rolls Royce and in my experience this is fact. Industry, or rather end users, should take note of this and change their approach to equipment being supplied on their sites. On future projects equipment selection based on standardisation should the prime consideration, if not the only consideration. It is true that on existing conveyor systems one cannot readily retrofit the alternative standard as there could be major interface issues.

The reverse observation is also true. The Te values are lower for the single drive pulley application thus indication that less work can be done for these applications. The most work can thus be done on the same belt class for dual drive with the 2:1 power sharing arrangement.

Consideration must be given to the duty requirements of the conveyor and a decision made accordingly. Generally the higher the belt class the higher the duty requirements. From an application perspective, the higher the belt class, the more power will be required for conveying the material. It will thus be logical to consider this type of application to be dual drive pulley applications with 2:1 power sharing as opposed to single drive pulley units.

Equally important it may be that the application cannot support the 2:1 power distribution as access becomes problematic. Under these circumstances the 1:1 distribution will have to be acceptable.

The aforesaid covers all the pertinent issues with respect to pulleys. What about drive units?

Drive unit sizes are directly related to calculated effective tension levels. For this approach one would then need to review the effective tension available after calculating the T1 and T2 values. The prime consideration should then be that once the belt class is determined, the drive unit size must be determined that will result in the power unit being adequate for the belt class being utilised at its maximum capacity.

Absorbed power is the product of the effective tension with the belt speed. Consideration is then given to the reducer efficiencies where a service factor is applied to this value to determine the minimum installed power. The entire process is as simplistic as that.

By example consider the 900 belt width class 1000 belting application. From Table 1 the effective tension required for driving is 74,38 kN. Of course you would need two power units each requiring 37,19 kN. Assuming a belt speed of 3 m/s, drive efficiency of 94% and a service factor of 1,2 on installed power the following will then apply.

Absorbed power is then: 37,19 kN x 3 m/s = 111,57 kW

Demand power on the input shaft is 111,57 kW ÷ 94% = 118,69 kW

Minimum installed power with service factor of 1,2 = 118,69 x 1,2 = 142,42 kW

Next motor size up is 160 kW

Thus final installed power is 160 kW on each drive pulley.

When now reviewing this application, the user will essentially have what can be described as a system balanced between the power and the belt class requirement. There should not be an application where the user will be in a situation that the 2 off 160 kW drive units will not be adequate for the belt class at that speed for any profile. As soon as the belt class requirements are exceeded, the power requirements will also become marginal and the system is thus balanced.

The reverse is not true. If the drive units were sized on less that the effective tension based on maximum belt class values the system will essentially be classed as being underpowered for the strength available from the class of belting. To really drive this point home consider the application if the original 80 kN was required for T1.

Using the same philosophy, the effective tension will be 1,21 of T1.

- Maximum Te tension = 80 kN ÷ 1.21
  = 66,11 kN
  = 2 x 33,06 kN

- Maximum T2 tension = T1 – Te
  = 80 kN – 66,11 kN
  = 13,89 kN

Absorbed power is then: 33,06 kN x 3 m/s = 99,18 kW

Demand power on the input shaft is 99,18 kW ÷ 94% = 105,51 kW

Minimum installed power with service factor of 1,2 = 105,51 x 1,2 = 126,61 kW

Next motor size up is 132 kW

Thus final installed power is 132 kW on each drive pulley.

From the previous example it is obvious that a situation can thus develop where the user will require the 900 wide class 1000 belt system to operate at a higher capacity but the drive equipment is inadequate for the application. The standard 160 kW system will meet all the class 1000 belting requirements but the designed for purpose 132 kW drive unit will not. From the end user perspective the 160 kW application now becomes the obvious choice.

The next objection that is normally raised against the standard approach is the strength of the steelwork. This is a valid question but the same answer will apply as with the previous. If the steelwork has been designed for the T1 maximum tension value of 90 kN as per the example, any requirement pertaining to the class of belt will be acceptable.

If the steelwork was designed for the 80 kN tension requirement then this becomes the limiting and marginalizing factor. In order for the client to maximise on his investment, the maximum value that he will be able to squeeze out of the class 1000 belt will be 80 kN in spite of the belt being able to operate at the higher tension of 90 kN. Once again it is obvious that the user will opt for the higher tension value as being the optimum value.

With the conservatives now running out of objections the point of varying belt speeds will be addressed. So what if the speed of the belt was to be changed?

The mechanical sizing of the equipment is directly related to the tension values in the system. Pulleys are sized accordingly and so the steelwork. In real terms only the size of the power units are affected by the speed of the belt. The torque remains the same, as a conveyor is essentially a constant torque machine.

Typically the variance in speed could typically be from one mine site to another mine in the same mining group. At the one mine the operation will be 3 m/s while the other will be 4 m/s. The one mine could require the capacity to be 900 tons per hour while the other will be 1200 tons per hour.

Believe it or not, both conveyor systems will still be 900 belt width class 1000 belting. As per the previous the power will be derived as follows.

Absorbed power is then: 37,19 kN x 4 m/s = 148,76 kW

Demand power on the input shaft is 148,76 kW ÷ 94% = 158,26 kW

Minimum installed power with service factor of 1,2 = 158,26 x 1,2 = 189,91 kW

Next motor size up is 200 kW

Thus final installed power is 200 kW on each drive pulley. As per the previous example, the identical conveyor will be fitted with a 132 kW power units.

The same owner will thus have two mines where the same conveyor will effectively be installed with only the installed power differing.

But, but, but, the conservatives will stutter, the gearbox will be different and how will that be handled, surely it must change. Ever heard of an adapter flange? By using an adapter flange exactly the same drive pulley and low speed coupling will be used on both applications.

In the real world it is impossible to standardise 100 percent. The better statement to make is that conveyors can be standardised to a very large extent. With some initiative and careful thought this objective can be readily achieved. There are more plus points in the process of creating common parts than parts only applicable to specific applications.

As these points are being mentioned it should be becoming glaringly obvious to those in the industry that certain user/mining groups have specific applications relative to their conveyor systems. These applications are currently been recognised in the industry by some of the visionary mining groups operational in South Africa. Most users have statements in their specifications requiring that equipment be standardised as far as possible. Unfortunately these statements are not really enforced and projects are predominately capital cost driven and not from a total cost of ownership perspective.

It is absolute common sense that cost of ownership is reduced immediately when less components are been held in stock. When only 3 spare drive units are to be kept in stock, as opposed to 5 different units and only 3 pulleys per belt width per class as opposed to 6 or 7, there must be associated financial benefits in this approach. These cost savings become legion as soon as standard units become available not only project-wide or site-wide, but also group wide. And so the advantages will snowball.

Central purchasing initiatives with the added advantages of central stockholding becomes the order of the day. There are some user groups that are slowly moving in this direction and they will be reaping these dividends soon.

The purpose of this paper has been to ask the question whether conveyors can be standardised. From the discussion it is believed that the answer is a definite affirmative.

CONCLUSIONS

There is indeed a possibility to standardise on conveyor equipment for various applications.

Belting specifications must be used as the starting point for optimum standardisation on all conveyor systems.

Pulleys require standardisation in conjunction with the class of belting used for the application.

There are options available where conveyor components can be standardised without necessarily over sizing of the units.

Power units can be standardised in accordance with the class of belting for the application.

RECOMMENDATIONS

All mining houses should review their conveyor applications in order to rationalise on all the equipment currently being used.

Total cost of ownership on conveyor-type projects must become a reality and more tangible and not a nice to have issue, as there are fundamental cost advantages in this approach.

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AUTHOR'S CV

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To date the author has been directly involved in the conveyor industry for over 28 years. During this time he has been exposed to all facets of the conveyor industry ranging from mechanical design, manufacture, installation, commissioning, visual inspection and testing of conveyor systems, audits and feasibility studies. The engineering of conveyor systems for both underground and surface applications are second nature to him. As current chairman of the Conveyor Manufacturers Association he is also active by serving on the SABS technical committees responsible for reviewing the national standards relative to all conveyor related issues.

Highlights of his career must be the four patents that he has been able to register as well as various firsts of conveyor installations conveyor systems like the first powered tripper drive in South Africa. Another milestone worth mentioning is the class 1250 PVC dual booster conveyor designed for an underground application at an overall length of 7300 m.

Simon is currently employed with Sandvik Materials Handling as manager of engineering for underground materials handling systems.