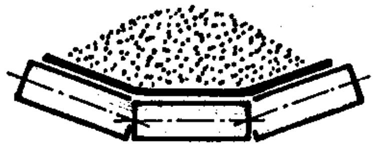
BELTCON 1



BELT CONVEYORS - DESIGN, OPERATION AND OPTIMIZATION

PAPER A2

CONVEYOR DESIGN AND DESIGN STANDARDS

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I N D E X

SUMMARY

This paper has been prepared with the intension of highlighting the problems faced by design engineers who are forced to undertake the design of belt conveyor systems using a multitude of design standards which have not been brought into line with modern technological advancements.

To overcome some of these problems, a basic outline of a universal standard has been proposed, which can easily be adapted to suit individual needs, without reducing the efficiency of the designer and his team.

1. INTRODUCTION

The design of belt conveyor systems has been one of the most common occurrences in the South African mining field for over one hundred years. Conveyors are seen on virtually all mining installations, and are the biggest problem for the plant maintenance engineer, being the cause of most plant shutdowns.

Why do belt conveyors cause such problems? It must be remembered that mining houses usually have a set of design standards to conform to; standards which are claimed have been developed over many years to suit their own needs in the materials handling field.

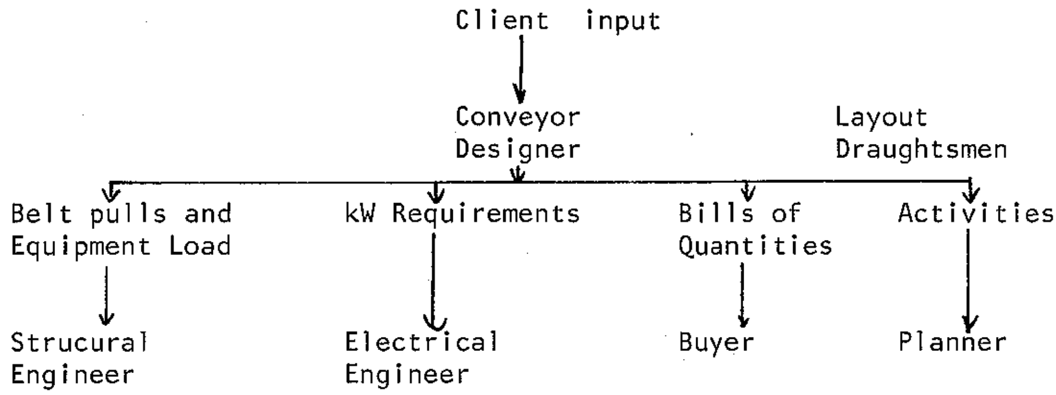
However, as I can understand the need for some aspects of a standard, others completely baffle me. It appears that having spent a great deal of time over certain requirements of a design standard, many of the fundamentals to which I am referring are of course the effects of overpowering on the whole conveyor system. Also, we know that to convey material from one point to another requires a specific amount of power using a belt designed to withstand a definite tension, so why is it that if a conveyor design problem is set to a number of designers, they will come up with many variations on a solution, even using the same design specification.

This of course comes down to the interpretation of, and the familiarity with the standard to be used. Basically I am suggesting that the standards as available to-day, leave a lot to be desired from the point of view of completeness, and ease of application.

2. JUSTIFICATION FOR A STANDARD

Do we need a standard at all? and if so, what form should it take? To answer this question let us look at a typical design office set up. On any project there are three key categories of staff, the designers, his draughtsmen and a group of peripheral staff, (planners, buyers, strucural, civil and electrical engineers). Thus we have a set up which looks as follows:-

Figure 1. Typical Project Engineering Flow Sheet



The designer is given a basic specification which will include material type and quantity to be conveyed from A to B. This he must transform into drawings for manufacture and fabrication, design data for civil, electrical and structural engineers, bills of quantities for buyers and activity networks for planners. With the exception of the planning information which is only realy relevant for the construction phase of the project, the designer has a problem which he will find very dificult to overcome, and that is to supply all the necessary information to each discipline on the project when they require it.

Therefore having obtained a scope of work from the client in question, the designer has to quickly produce the design data, but before he is able to proceed he must obtain information from his drawing office relating to the layout of the conveyors in the system. Now the problems begin: Prior to undertaking any calculations whatsoever the designer must check the specifications to which he must conform.

As virtually all clients have their own opinion on the subject of conveyor design, we can rest assured there will be some form of client input, whether it be a two volume manuscript or simply an, 'All drives shall ....' document.

The designer is confronted with conforming to the said specification, but much worse, he must ensure that his drawing office staff are aware that there is a specification to work to. Consider that the previous week they may have been working on another project and had to conform to a completly different specification.

What does the designer do? Does he circulate multiple copies to his drawing office with the instruction that it must be read prior to any work being started. If so, he will possibly not meet his deadline on the supply of data to the peripheral disiplines.

Does he try to check that his draughtsmen conform by 'looking over their shoulders' from time to time (which is the way mistakes are guaranteed to occur). Alternatively does he instruct his drawing office that there is a specification to work to, that it is lying around somewhere and to 'please check it if you are not to sure of how to proceed'.

In all the offices in which I have worked, the last two solutions have been applied, with the result that, almost without exception, the experienced draughtsmen who know how to make a system work will continue with very little reference to the said specification.

The problem may be that on this project 'the pulleys are much bigger, the take-up length must be selected using an ill defined formula and basically we don't know how to design a conveyor anymore'. If this probem is caught early enough we only have to change a quantity of drawings and are then back on the right road. However you can be sure that in practice it will be too late, and the designer has to go to the client and ask for a concession because he is not able to conform to the specification, and to make any changes to the drawings now will put him way behind schedule. Furthermore before the client will accept deviations to the proposed format, every avenue must be explored, and a report on the deviation prepared.

The designer is now behind whether he likes it or not and to make up time he must neglect the one function which completes the total conveyor design, that of secondary design. By secondary design, I mean the design which comes after the conceptual or general arrangement layouts are complete. This is the design of the chutes, the location of bearings, the belt cleaning system to be employed and the access for maintenance. This is left to a draughtsman without any engineering support. However, the secondary design usually encompasses the major problems of belt conveyor system design. These are areas with very little coverage in specifications, with comments such as, 'all conveyors will have pulleys at terminal points', being the limit to such specifications.

I pose the question again, do we need a design standard? Those who agree with the scenario I have set will probably say, 'Allow the designer the freedom to do the job'. However I feel that a standard is essential. There are very few specialist conveyor designers and thus some form of guidance must be given. However there should be only one standard, with one basic set of parameters and which can cater for the needs of every mining and process plant application. Without lessening the efficiency of the designer and his team such a standard will facilitate the efficiency the overcoming of the problems occuring in secondary design.

We know this has been tried repeatedly in the past, but always in isolation from the main stream of design and usually with the statement, 'but it caters for our own individual needs', as justification.

Having been confronted with conveyor design standards for a number of years, I have still to find a true specialist need, I know that some clients require less capacity on a belt, others require larger pulleys and thicker belts, requiring the use of complicated formula to arrive at a solution, but this can not be justification for devising completely individual specifications, which could more suitably be covered in a single paragraph of a comprehensive specification.

3. PRESENT DESIGN STANDARDS

Let us look at at the Conveyor design standards available, and in particular the four most commonly used, C.E.M.A., GOOD YEAR, ISCOR and A.A.C. If we consider the power and tension variation predicted by using these systems, as in Table 1, we see quite a wide range of possibilities. The reason for this is in the selection of the rolling resistance factor, (coefficient of friction, resistance to flexure or other commonly used terms) which varies between 0,016 and 0,035 as used in the above standards.

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Table 1 Power and Tension calculations.

1(a) based on belt capacity of 500tons per hour, belt width of 900mm and a belt velocity of 2,2m/sec.

Length	Lift	C.E.M.A.		GOOD YEAR		ISCOR		A.A.C.	
		power	tesn	power	tesn	power	tesn	power	tesn
m	m	kW	kN	kW	kN	kW	kN	kW	kN
30	0	6	9	15	16	16	19	12	18
200	60	101	65	99	64	104	66	102	66
1000	0	81	40	89	43	113	54	104	50
1000	40	132	72	143	77	167	88	158	84

1(b) based on belt capacity of 2000tons per hour, belt width of 1500mm and a belt velocity of 3m/sec.

Length	Lift	C.E.M.A.		GOOD YEAR		ISCOR		A.A.C.	
		power	tesn	power	tesn	power	tesn	power	tesn
m	m	kW	kN	kW	kN	kW	kN	kW	kN
30	0	18	22	36	41	38	42	37	42
200	60	378	167	380	168	403	176	391	172
1000	0	221	84	262	98	349	127	315	116
1000	40	439	174	479	188	567	217	533	206

On the shorter systems this difference is quite insignificant, except that the belt length factor plays an important part. However on the now common large overland type systems, these variations are unsatisfactory to say the least.

Are we able or prepared to accept such variations? Able, I will say yes, provided we take cognisance of the effects of owerpowering. However I am not convinced we should be prepared to accept these variations, apart from the overpowering factor there are purely economic conciderations to account for. This point is very noticeable when one becomes involved in economic evaluations (feasibility studies) of various alternative solutions to a specific materials handling problem. For instance, how competitive would

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a pneumatic conveying system or cable belt system be if designed to similar sets of standards as the conveyor. However as these standards are as yet, not available, the manufacturer of competitive systems has far reaching advantages over the conveyor manufacturers.

I am not for one moment suggesting that the competitive systems are under designed, simply that the designer is not limited to designing within a conservative specification.

Too often we see examples of conveyor systems feeding process plants, where to conform to specification the whole conveyor network is designed for a large amount of excess capacity. However, this philosophy is not transfered to the related equipment in the rest of the plant.

4. PROPOSED STANDARD FORMAT

4.1 Power and Tension

With power and tension calculations there exists the possibility for a combination of all four of the above standards by utilizing a single friction factor for the shorter belts, but eliminating the belt length factor which can easily be compensated for with the overrating factor of the motor. In progressing to the longer conveyors this factor could be variable, as advocated by C.E.M.A., only now be simply a function of belt length and capacity. Then we could use a simplified formula as follows:-

$$\text{Power (kW)} \doteq \frac{9{,}81}{1000} \text{x} L.V((kX+kY(Wm+Wb)+{,}015Wb)+(H.Wm))$$

Where L = Horizontal pulley centres (m)

H = Vertical pulley centres (m)

V = Belt velocity (m/sec.)

Wm = Mass of material per meter run (kg)

Wb = Mass of belt per metre run (kg)

0,015 = Return belt resistance

kX = Belt slide and Idler rotational resistance and can be obtained from:-

kX = 0,00068(Wm+Wb)+0,022(rotating mass of the Idler per meter) (kg/m)

kY = Resistance of the belt of flexure as it moves over the Idlers, and can be concidered to be the same as the friction factors given in all the specifications.

Typical values of kY are given in table 2 below.

Table 2. Selection of kY factor based on Belt length, lift and capacity.

Length m	Lift m	kY 500t/hr.	kY 1000t/hr.	kY 2000t/hr.	kY 3000t/hr.
100	20	0,035	0,030	0,026	0,022
200	20	0,032	0,026	0,022	0,020
200	40	0,030	0,022	0,020	0,020
400	20	0,030	0,022	0,020	0,020
400	40	0,026	0,020	0,020	0,020
800	40	0,022	0,020	0,020	0,020
1000	40	0,020	0,020	0,020	0,020

To enable the client to maintain control of the outcome of the calculation, it is necessary only to specify the kY factor to be used in a simple adendum to the main specification.

Belt tension calculation can be kept straightforward, provided the designer starts by considering the minimum belt tensions, at both the drive and tail pulleys, by using the following formulae :-

$$T_{min} = \frac{4,2x9,81}{1000} Si (Wb+Wm) \quad kN$$

Where 4,2 = Factor based on a 3% belt sag.

Si = Idler spacing, m

and

$$T_{slack\ side} = T_{effective} / e \quad - 1$$

Where Teffective is the installed drive effective tension and not the effective tension computed from the above power formula.

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The one problem that is encountered is in the selection of a coefficient of friction for the drive pulley. A standard such as given in Table 3 could be used.

Table 3 Coefficient of Friction for Drive Pulleys.

Plant Description	Conveyor Construction	Type of Take up: Automatic, Lagged	Automatic, Unlagged	Manual, Lagged	Manual, Unlagged
Wet	Covered	0,25	0,10	0,20	0,10
	Uncovered	0,20	0,10	0,20	0,10
Semi-wet	Covered	0,30	0,20	0,25	0,18
	Uncovered	0,25	0,15	0,22	0,13
Dry	Covered	0,35	0,22	0,25	0,20
	Uncovered	0,30	0,18	0,25	0,15

Table 3 has been compiled from empirical data such as that given in Table 4. It should be noted that these values are the limiting conditions (when the belt is on the point of slipping). The actual coefficients of friction developed between surfaces are, in practically all cases where slipping does not occur, in excess of those listed.

Therefore, the convention of using these values does not reflect what actually occurs at the drive pulley.

If one considers a drive pulley under operating conditions then the higher tensioned belt section is stretched more than on the lower tensioned section, thus the belt entering the positive drive will be travelling faster than when it leaves it. The elastic recovery of the belt occurs over only a part of the total angle of contact, and it is at this point, where creep takes place, that the driving is done, that the driving is done, while making full use of the coefficient of friction.

By applying the classic tension formula to the whole angle of wrap a a fictitious coefficient of friction is being used.

Table 4. Recommended Drive Coefficient of Friction of Various Standards.

Condition	C.E.M.A.	STEVENS ADAMSON	BRIDGESTONE	LINATEX	REMA TIP TOP
Bare pulley	0,25	0,35	0,20	---	---
Lagged	0,35	0,35	---	0,60	0,45
Dry lagged	0,35	0,35	0,35	0,60	0,45
Wet lagged	0,35	0,35	0,25	0,80	0,35
Wet & dirty	0,35	0,35	0,20	0,40	0,25

The advantage of working from minimum drive tension back to the maximum drive tension, can be better explained if one looks at the design of pulleys and shafts. Over the years there has been a lot written about the design of a pulley shaft, with the aim of trying to eliminate the high failure rate and the cost associated with such failures.

I feel that there are only two basic reasons for pulley failure, firstly the bad manufacturing procedures, and secondly, failure owing to an inability to calculate the minimum drive tension. The latter case of incorrect design results in the counterwieght mass having to be increased to overcome drive slip on startup, with the result that pulley shafts are subjected to excessive loads, producing eventual failure.

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By contrast, if the minimum drive tension is used as a design basis, we can overcome failures in pulleys caused by inaccurate design. Thus the maximum tension will be obtained from :-

$$T_{maximum} = T_{miniumum} + T_{effective}$$

Where $T_{effective}$ is computed from shaft power and not the installed power.

Note that the formulae discussed above are applicable to 90% of the conveyor installations being designed to-day. However a little more analysis is required for some overland and complex systems.

4.2 Pulley and Shaft Standards

There are presently two major standards used for pulley and shaft selection, these being the ISCOR and AAC systems. I know much has been written about the high degree of oversizing adopted by both standards, but I feel that as the pulley is one of the least expensive components in a conveyor installation, we should not be over concerned on the point.

Efforts should rather be directed at reducing the amount of variations there are in the selection of face width and bearing centres. At the moment both ISCOR and AAC have two sizes per belt width, all different. This should be reduced to a single size per belt width, and this size should be as big as possible to allow easy access and hence reduce the damage to coveyor belts. A standard along the lines of table 5 based on the ISCOR specification would be the most acceptable.

Pulley and shaft diameters should be kept to a minimum of two per conveyor, with as much standardization as possible being employed on the whole conveyor system. The selection of pulleys and shafts could be from a table similar to that shown as Table 6.

Table 5. Pulley Face Width and Bearing Centres

Belt width mm	Face width mm	Bearing centre mm
450	550	890
600	700	1140
750	900	1370
900	1050	1520
1050	1200	1670
1200	1350	1850
1350	1500	2000
1500	1700	2300
1800	2000	2630
2100	2300	2930

4.3 Selection of Belt Width and Velocity

The selection of belt width and velocity is probably the most frustrating of problems facing the designer. There are a variety of factors being used, factors such as :- the belt width must be three times the maximum lump size, the belt width must be such that the system can cater for 66% excess capacity, and if a tripper is used the factors must be increased by a further 30% etc.

This type of factor forms the basis for most standards in use to-day, and these could therefore be rationalized into a single more acceptable standard to make the designer's task easier.

The first necessary step is the removal of the age old belt speed restrictions, after all speeds in excess of 4m/sec are now quite common.

I am not advocating that the highest possible belt speed be used for all installations; I simply suggest that belt speeds should not be selected only on the basis of past experience, but on the basis of belt length, transfer point and economic considerations.

I feel that to use the criterion I have set out will automatically result in the selection of the most suitable belt width and speed. My reasoning here is that, for inplant installations belt widths and speeds are almost always selected on the basis of standardization and possible transfer point problems. By contrast, the larger overland systems are selected on the basis of capital costs and the associated oprating and maintenance costs, because as belt speeds increase operating and maintenance costs usually follow suit.

Consider the suggested methods of selecting a belt width and speed. Firstly, the amount of material on a belt must be related to the expected transfer point problems. A flat feed point fed by a controlled system will be far easier to design than an inclined feed point fed from a crusher, where surges are very common. Therefore to suggest a similar standard for both applications is not practical.

We often are told that conveyors should not be fed at angles of 8° incline feed points and very tight vertical curves, with the result that the feed point stays clean, but at the curve the belt has lifted causing spillage.

I would like to suggest that a belt can be easily fed at angles of up to 16°, provided the belt width and speed are correctly selected. It may be necessary to install belts with thicker covers, but this can form the basis for a better design.

Thus the type of standard that could be used is shown in Table 7.

Table 7 Implant Conveyor Load Factors

Loading Point Type	Feed Type	Overload Factor
Horizontal	Uniform	1,20
Horizontal	Surge	1,50
Incline	Uniform	1,50
Horizontal	Surge	1,75
Tripper	.....	1,75
Shuttle	.....	1,50

The overload factor would be used to increase the design tonnage for selection purposes.

For overland conveyors it is common to use horizontal loading points, and we are not confronted with the same problems. As mentioned earlier it is only necessary to consider the economics of the system, with the following limitations as given in Table 8.

Table 8 Overland Conveyor Minimum Belt Widths and Maximum Speeds

Terminal Pulley Centres (m)	Belt Width (mm)	Belt Speed (m/sec)
300 to 500	600	3,50
500 to 1000	750	3,50
over 1000	900	7,00

The overload factor used should always be a minimum of 1,2 times the design tonnage.

4.4 Idler Standards

4.4.1 Introduction

The introduction of the SABS Idler specification will ensure a more uniform selection of idlers. As a result the choice of type and spacing for Idlers should be on a more scientific basis. The types of Idler to be used on conveyors are; transision, troughing, impact and return idlers. At this time there is no satisfactory training idler available so they should be avioded.

4.4.2 Troughing Idler Spacing

Two types of troughing idler are used frequently, fixed and suspended roll. There is very little difference between the two, except the training characteristics and possible cost savings associated with the suspended roll.

The question of idler spacing needs be considered more carefully. The restrictive standards as applied to-day do more harm than good to a conveyor system. Idlers are the highest maintenance cost item on a conveyor installation and the biggest cause of belt damage, therefore 'the fewer the better'.

Idler spacing must be selected on the grounds of available belt tension, fatigue life of the idler bearings, and structural considerations. The upper spacing limit should be set at 2200mm. Account should be taken of four and five roll sets, but no significance can be attached to the claim that four and five roll idlers give better belt life.

4.5 Drive Standards

The standardization of drives is the key to most successful conveyor systems. The problem is however that some drives have to be drastically oversized to obtain some degree of conformity.

By considering this point at an early stage in the design process, it is usually possible to overcome the problem, therefore simple cost analyses of all the possible solutions can quickly decide on the drive sizes to be adopted. Also it is at this point in time when a final selection of belts can be carried out, because there is often scope to change belt speeds to the required degree of standardization, and we should not be afraid to to this.

5. Conclusion

To conclude I would like to reiterate the need for a single design standard, which could be applied to any conveyor installation. However, this standard must be such that it allows the client a small amount of individuallity and flexibility.

The design system as outlined in this paper can offer this flexability, by allowing the client the freedom to select the kY factor, the drive coefficient of friciton and the load factor for selecting the belt width and speed. Coupled with this we can have a very efficient system especially if it is adapted to computerised calculation techniques. I know to-day that many such design programmes are available, but because of the variations in standards that must be incorporated, their credability is unjustly made suspect, forcing the designer to revert to the longwinded number crunching exercises which ovbiously reduce his effectiveness in the drawing office.

BELT	PULLEY	HEAVY DUTY		MEDIUM DUTY		LIGHT DUTY		BELT TYPE		MAXIMUM
		SHAFT	BEARING	SHAFT	BEARING	SHAFT	BEARING	PLY	STEEL	SHAFT LOAD
WIDTH	DIA. D	DIA. d	DIA. d1	DIA. d	DIA. d1	DIA. d	DIA. d1	CLASS	CORE	kN
450	300	90	75	75	50	-	-	200		18
	400	100	75	75	50	-	-	250		22
	500	125	100	100	75	75	50	630		56
	630	140	110	110	90	90	75	800		72
600	400	110	90	90	75	75	50	250		30
	500	125	100	100	75	90	75	630		75
	630	140	110	110	90	100	90	800		95
	710	160	125	125	100	110	90	1250		150
750	400	125	100	100	75	75	50	250		35
	500	140	110	125	100	90	75	630		95
	630	160	125	140	110	110	90	800	ST500	120
	710	180	140	160	125	125	100	1250	ST630	190
	800	200	160	180	140	140	110	1250	ST1250	280
900	400	140	110	110	110	90	90	250		45
	500	160	125	140	140	110	110	630		110
	630	180	140	160	125	125	100	800	ST500	145
	710	200	160	180	140	140	110	1250	ST630	225
	800	220	180	200	160	160	125	1250	ST1250	340
	1000	240	200	220	180	180	140	1600	ST1600	430

ALL DIMENSIONS IN MILLIMETRES

MAXIMUM LOAD FIGURE = PERMISSIBLE LOAD ON PULLEY = TWICE BELT TENSION

HEAVY DUTY = 100% MAXIMUM LOAD

MEDIUM DUTY = 60% MAXIMUM LOAD

LIGHT DUTY = 30% MAXIMUM LOAD

DENOTES RATING BASED UPON STEEL CORE BELT

FOR ALLOWABLE LOAD ON BEARING SEE BEARING RATING TABLES

BELT RATING CHART

Table 6a

BELT	PULLEY	HEAVY DUTY		MEDIUM DUTY		LIGHT DUTY		BELT TYPE		MAXIMUM
WIDTH	DIA. D	SHAFT DIA.d	BEARING DIA. d1	SHAFT DIA.d	BEARING DIA.d1	SHAFT DIA.d	BEARING DIA. d1	PLY CLASS	STEEL CORE	SHAFT LOAD kN
1050	500	180	140	140	110	110	75	630		130
	630	200	160	160	125	110	90	800	ST500	170
	710	220	180	180	140	125	100	1250	ST630	260
	800	240	200	200	160	140	110	1250	ST1250	395
	1000	250	220	220	180	160	125	1600	ST1600	505
	1250	360	340	260	220	180	140	2000	ST3150	985
1200	500	180	140	140	110	110	90	630		150
	630	200	160	160	125	125	100	800	ST500	190
	710	220	180	180	140	140	110	1250	ST630	300
	800	240	200	200	160	160	125	1250	ST1250	450
	1000	260	220	220	180	180	140	1600	ST1600	575
	1250	360	340	300	260	220	180	2000	ST3150	1130
1350	500	200	160	180	140	140	110	630		170
	630	220	180	200	160	160	125	800	ST500	215
	710	240	200	220	180	180	140	1250	ST630	340
	800	280	240	240	200	200	160	1250	ST1250	505
	1000	300	260	260	220	220	180	1600	ST1600	650
	1250	360	320	300	260	240	200	2000	ST3150	1270
1500	630	240	200	200	160	140	110	800	ST500	240
	710	280	240	220	180	160	125	1250	ST630	375
	800	300	260	240	200	180	140	1250	ST1250	560
	1000	320	280	260	220	200	160	1600	ST1600	720
	1250	360	320	280	240	220	180	2000	ST3150	1400
	1400	400	380	320	280	240	200	2500	ST4000	1800
1800	710	300	260	260	220	180	160	1250	ST630	450
	800	320	280	280	240	200	180	1250	ST1250	670
	1000	340	300	300	260	240	260	1600	ST1600	865
	1250	380	340	320	280	260	220	2000	ST3150	1700
	1400	410	380	340	300	380	240	2500	ST4000	2100
	1500	430	400	360	320	300	260		ST5000	2700
2100	710	300	260	260	220	180	160	1250	ST630	525
	800	320	280	280	240	200	180	1250	ST1250	790
	1000	340	300	300	260	240	200	1600	ST1600	1010
	1250	380	340	320	280	260	220	2000	ST3150	1900
	1400	410	380	340	300	280	240	2500	ST4000	2500
	1500	430	400	380	340	300	280		ST5000	3100

SEE GENERAL NOTES ON SHEET 1

BELT RATING CHART

Table 6b