PAPER PRESENTED TO

BELTCON WORKSHOP

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GETTING STARTED

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BELTCON WORKSHOP

GETTING STARTED

In this workshop we will look at the basics of getting started on a project. Each conveyor we design is part of a system, it is a link between 2 processes or portions of plant and must designed in such a manner that it satisfies the needs of the processes and the system in general.

Before we pull out our slide rule, pocket hand book, calculator or computer, or whatever else we are going to use to calculate the conveyor, we must look first at the purpose of the conveyor and how it is to achieve the system's needs. Conveyors handle various types of material and the configuration must cater for those differences, which mainly involve the chute design.

TYPICAL PRODUCT TYPES

	PRODUCT	USE	PROBLEMS
1.	Coal	Power Station	High moisture High fines
2.	Coal	Export gasification	Low degradation required
3.	Fly Ash	Waste product	Dusty if dry Sticky if wet Abrasive
4.	Coarse Ash	Waste product	Sticky Abrasive
5.	Limestone	Cement	Sticky if wet Very abrasive if dry
6.	Ores & Minerals	Various	Impact loadings

COAL - POWER STATIONS

In this application breakage of the product is not a main criteria since the product will ultimately be pulverised for boiler combustion.

The criteria here is handling the product with a high fines content and a high moisture content. The coal at some stage will have passed through a washing plant or will have been reclaimed from an open stockpile. Chute angles should be selected to prevent blockages and hang ups and a concentrated effort should be made on addressing the belt cleaning and any tailing chutes to prevent the accumulation of scrapings resulting ultimately in blockages. Drop boxes should be avoided for the following reasons :

1. Danger of combustion

2. A varying bias develops in the drop box favouring first a discharge from the left followed by discharge from the right. This can play havoc with central loading.

COAL - EXPORT AND GASIFICATION

The final product usually has the fines removed in which case chute angles may be lowered. Attempts to catch the material at shallow angles are required and the need to carefully guide the coal on to the next conveyor or piece of equipment as gently as possible is necessary to avoid degradation.

Wide, low speed belts with low discharges are ideal for this application. If large distances are involved or particularly high tonnages, quicker belts may be required. If the belt speed is such that a forward trajectory from the head pulley is generated, separate chutes for product and scrapings is required together with a logic and facility catering for the starting and stopping conditions.

FLY ASH - WASTE PRODUCT

Wide, low speed belts are ideal for this application, where attempts should be made to discharge the material and the scrapings directly onto the next conveyor without contact with the chutework. This is always providing that central loading can be achieved. Low discharges are therefore essential.

Should circumstances dictate, then fast running belts are required and chute contact cannot be avoided. Very large, steep angle chutes are required to minimise build up and allow self cleaning of build ups, within the chute, or allow suitable operating times to elapse before maintenance is required.
If it is not possible to have the scrapings face of the chute vertical, then it should be as steep as possible and should be either lined with a continuous strip of stainless steel or U.H.M.W. Polyethylene.

The belts will be subject to carrying water on a frequent basis as the fly ash conditions are washed out.

There are two areas to consider here, the first being that inclines on the belts must be low enough to handle the water and second, should the water not have cleared prior to the ash being sent, the ash will fluidise and spill from the belt as it fails to negotiate the incline. The density and angle of repose of the fly ash will vary tremendously dependent on whether the ash is dry or wet. Cognisance of this must be taken in the selection of belt speeds and widths.

COARSE ASH - WASTE PRODUCT

This material tends to be very sticky and abrasive. Direct loading from low discharge head pulleys onto the receiving belt is recommended for low tonnages and low speed belts.

LIMESTONE - CEMENT

High discharge narrow chutes should be avoided at all costs with this product. In it's dry condition the material will flow well however as the material speed increases in the chutes extremely high wear will result cutting through steel liners in a matter of days and washing out liner securing bolts. When wet, this product may stick to a vertical pane of glass. Low discharges with minimal chute contact are therefore the order of the day.

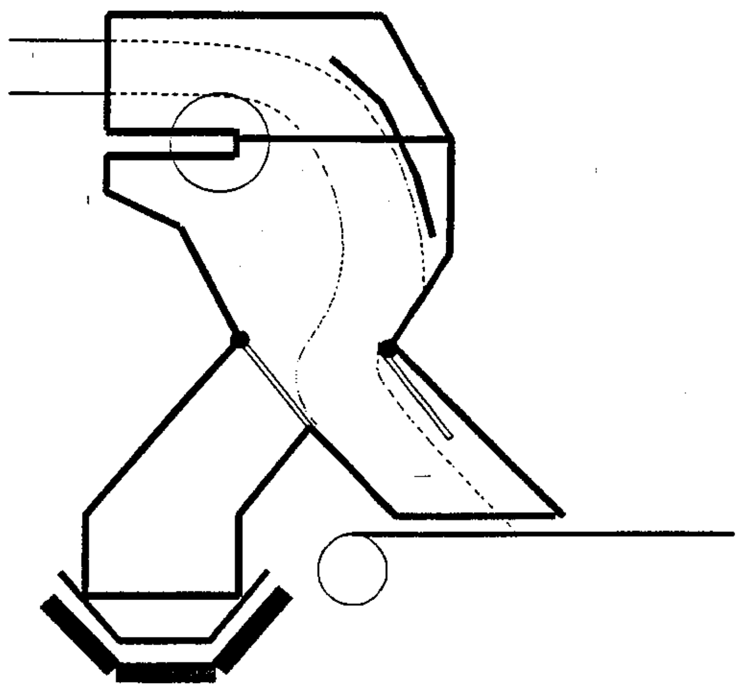
ORES AND MINERALS

Low discharges utilising drop/crash boxes are required to minimise impact abrasion within the chutework. Chutework must ensure that on the discharge the material speed and direction is as close as possible to that of the receiving belt otherwise severe belt wear will occur.

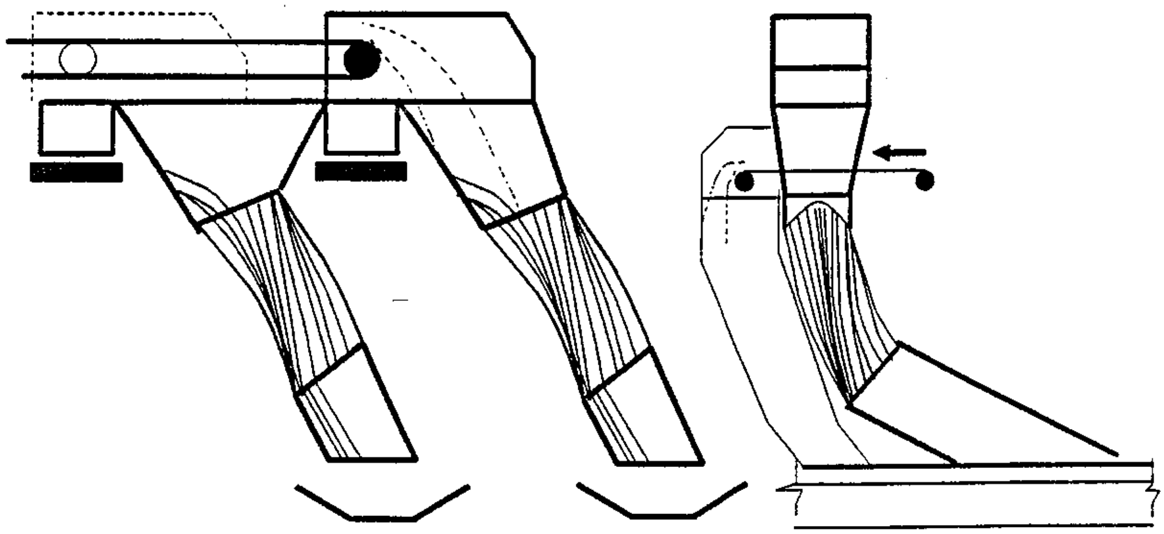
By considering the material we can decide upon the configuration of the conveyor. We are not however ready to calculate yet.

We now need to consider how the system is to be fed.

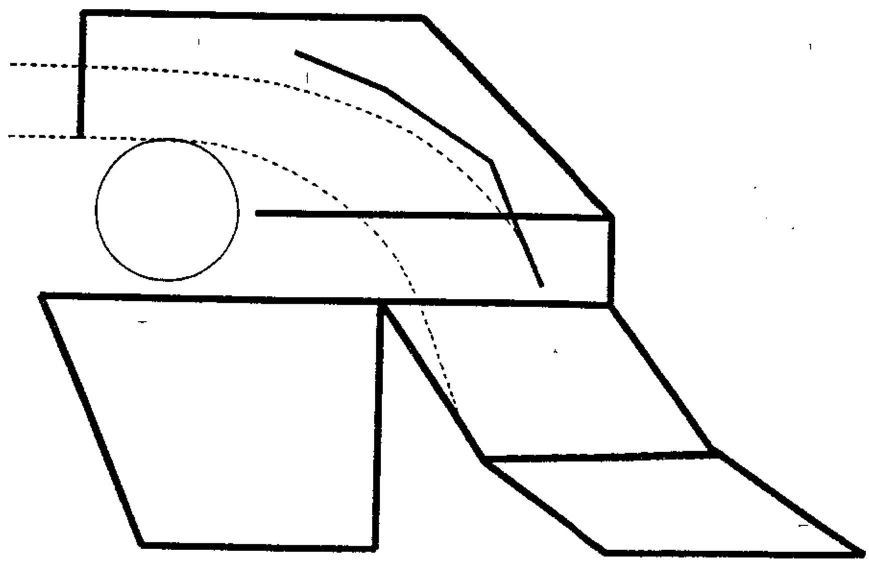
TWO WAY TRANSFER USING FLOPPER GATES



TYPICAL MOVING HEAD

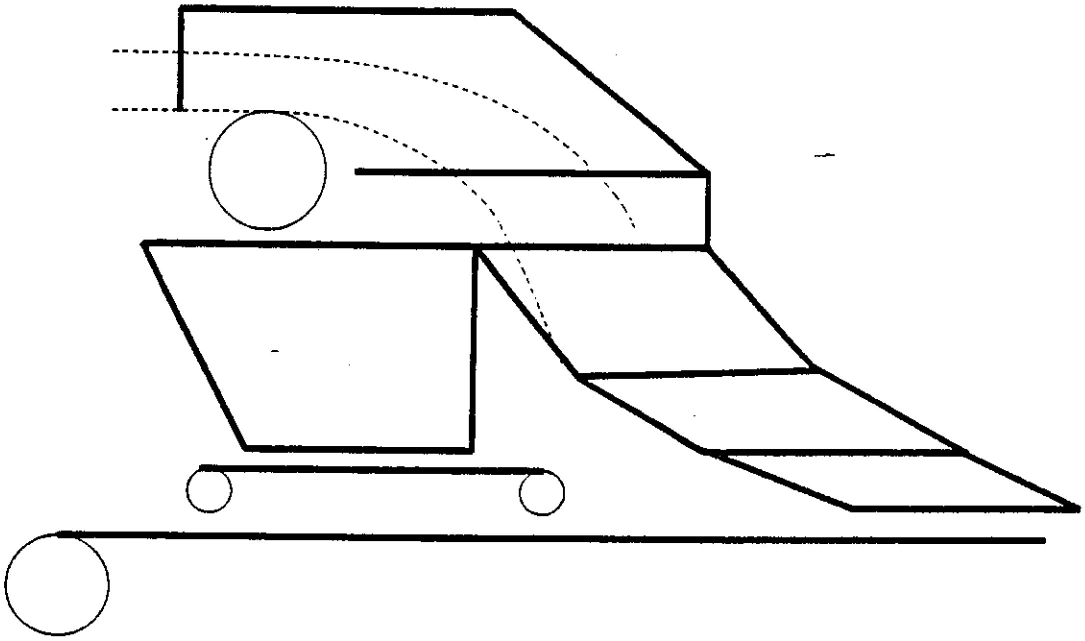


TYPICAL STRAIGHT TRANSFER



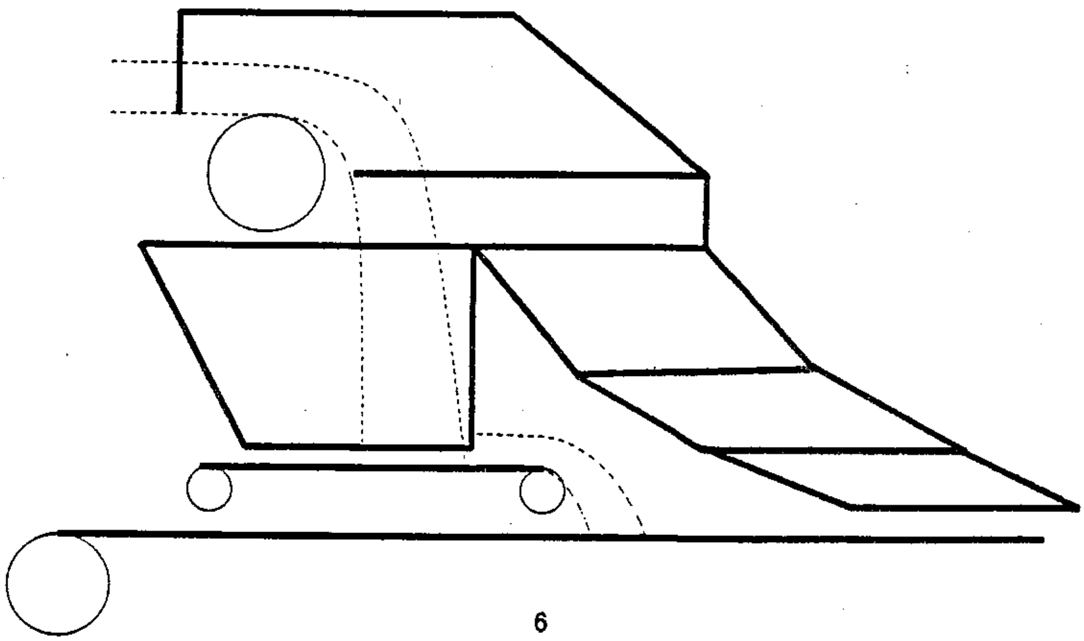
TYPICAL OFFSET TRANSFER

NORMAL RUNNING CONDITION

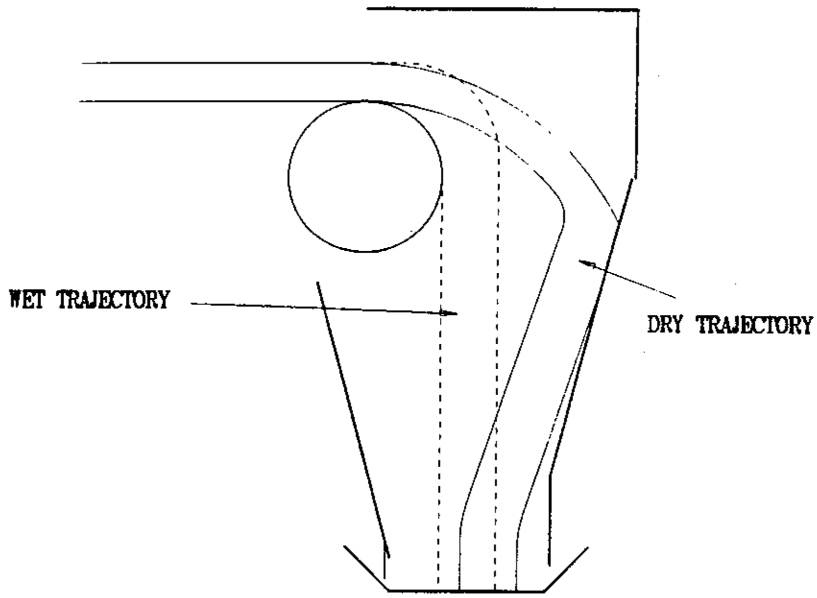


TYPICAL OFFSET TRANSFER

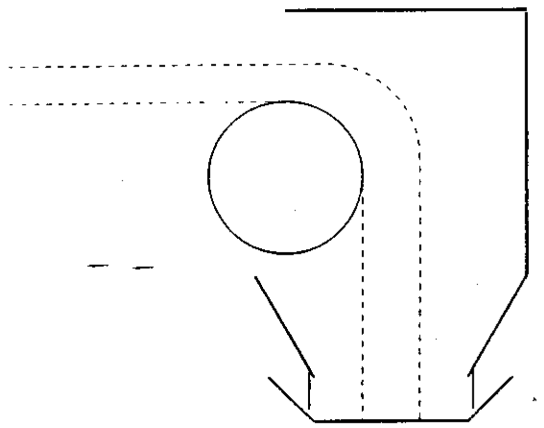
STARTING AND STOPPING CONDITION



TYPICAL TRANSFER FOR FLY/MIXED ASH AT HIGH BELT SPEED



TYPICAL TRANSFER FOR LIMESTONE AND COARSE ASH



SYSTEM FEEDS

1. Continuous Feed

a)	Sized dry product from	Apron Feeder Vibratory Feeder Belt Feeder Slide/Clamshell gate Travelling Plough
b)	R.O.M. dry product from	Apron Conveyor Belt Feeder

2. Uneven Feed

a)	Sized dry product from	Bucket Elevator Bucket Wheel Reclaimer
b)	R.O.M. product from	Twin Apron Feeders
c)	Damp product from	Vibrating Feeders Gates etc

AVERAGE AND MAXIMUM AND DESIGN CAPACITIES

Average Capacity

The average capacity may be determined on a yearly, monthly, weekly or daily basis. By determining the operating hours of the plant, holidays, system availability and other influencing factors, an average operating capacity can be determined. This average will form the basis for normal operation.

Normal Capacity of the Belt (T.P.H.)

This is the condition in which the conveyor may be expected to be running when randomly inspected. The normal capacity should take into account surges and uneven feed rates etc.

Maximum Capacity of the Belt (T.P.H.)

This is the capacity at which the conveyor may be expected to run when catching up with lost production. This figure should include surges and uneven feed rates.

Design Capacity of the Belt

Belt width and speed selected to suit the maximum capacity including local surges due to carry over in the chutework during stopping and subsequent restarting at the lowest anticipated bulk density.

Power and tension calculations should be based on the maximum load conditions generated by the maximum capacity of the belt (T.P.H.) after taking into consideration the additional loading caused by the release of carry over in chutework during starting, at the highest bulk density anticipated. It may be required that an allowance over and above is requested by the client.

AREAS OF IMPORTANCE

Material Carry Over

The size of the conveyors and stopping times of the respective conveyors can give cause for concern and designers should consider the implications of material carry over in chutes.

Case 1

The 12 km overland conveyor fed onto a 1½ km inclined conveyor, at which point the transfer was able to contain the carry over of coal due to the varying stopping times, however the discharge from the chute at start up was quite capable of flooding the receiving belt beyond the point of spillage. Control of the outflow of the chute was therefore necessary.

By restricting the flow to 2500 ton/hr an acceptable local overload slug of some 35 metres could be handled without spillage or detriment to the system. The receiving belt discharged into a bin and the slug generated at the conveyor transfer caused no more problems.
The above is not always possible.

Case 2

A $2\frac{1}{2}$ km slightly declined conveyor was required to feed onto a 100 metre inclined conveyor. The inclined conveyor was one of a chain of similar conveyors feeding a stacking system.

The carry over generated if left to its own devices would flood the 100 metre conveyor causing spillage. More importantly the carry over was quite capable of generating a slug of material in excess of the length of the existing stacker boom lengths causing major overload of these conveyors. Here a vicious circle can easily be generated since a trip of the stacker boom on overload would generate a second slug at the transfer of the $2\frac{1}{2}$ km conveyor to the 100 metre conveyor thus priming the system upon start up for another boom belt trip .... Control of the outlet of the chute together with considerable braking of the $2\frac{1}{2}$ km conveyor was therefore necessary to give an acceptable situation.

The above demonstrates that it is insufficient to assume that carry over may be simply contained within a chute without consideration of the effects on the entire system.

Models can be created to monitor the flow of coal through the chutes under the various stopping conditions. These models allow for restricted flow onto the conveyors without creation of a spillage and totalise the surplus coal retained in the chutes. The following examples show the monitoring of the material at the discharge between two conveyors and the build up of coal that occurs within the chute during stopping. The model may be adjusted to cater for the various stopping conditions of the two conveyors and the allowable overload of the receiving conveyor.

As the conveyors stop the quantity of material on the receiving conveyor increases until it reaches a prescribed limit. From that point the surplus material is held inside the chute where it will be discharged on start-up, also at the prescribed limit of the conveyor capacity.

The model may also be run to consider the build up occurring within the chutes with the material at different bulk densities, since the chute opening will control the material on a volume basis not a ton/hour rate.

It follows that a number of such examples need to be considered at any transfer point where a danger exists in the feeding conveyor stopping in a longer time than the receiving conveyor. The most common case occurs in the discharge between a long yard belt and a stacker boom conveyor.

As mentioned earlier the slug of material generated under such stopping conditions also needs consideration. The following examples show the slugs effect on the receiving conveyor and any other small conveyor downstream of the discharge.

FILE : OVERLOAD

Keeve Steyn Incorporated

Keeve Steyn Inc.
Written by DCM
Updated 11-Jul-91
Prt.date 14-Jul-91

PORTION OF BELT OVERLOADED BY CARRY OVER DURING STOPPING

Material carried over.............. 5 Tons (From "Carryo") (Inc. mat.not held in chute)

Conveyor capacities	RECEIVING CONVEYOR	SHORTEST CONVEYOR	
Maximum cap. of belt	0.280 m^2	0.348 m^2	(At the point of spill)
Restricted cap. of belt	0.247 m^2	0.247 m^2	(Baffle or chute opening)
Selection (max=0, rest=1)	0	0	
Selected capacity	0.247 m^2	0.247 m^2	

RECEIVING CONVEYOR	FEED LENGTH (m)	FEED RATE (T/h)	BELT SPEED (m/s)	MATERIAL DENSITY (T/m^3)	CROSS SEC.AREA (m^2)	TOTAL MATERIAL (Tons)
NORMAL	1476.18	1800	3.34	0.85	0.1761	220.99
O/LOAD	23.82	2524	3.34	0.85	0.2470	5.00
TOTAL	1500.00	1812	3.34	0.85	N/A	225.99

The conveyor has therefore a minimum power requirement based on .... 1812 tons/hour.
and a minimum tension (sag) based on .... 2524 tons/hour.

SHORTEST CONVEYOR	FEED LENGTH (m)	FEED RATE (T/h)	BELT SPEED (m/s)	MATERIAL DENSITY (T/m^3)	CROSS SEC.AREA (m^2)	TOTAL MATERIAL (Tons)
NORMAL	476.18	1800	3.34	0.85	0.1761	71.29
O/LOAD	23.82	2524	3.34	0.85	0.2470	5.00
TOTAL	500.00	1835	3.34	0.85	N/A	76.29

The conveyor has therefore a minimum power requirement based on .... 1835 tons/hour.
and a minimum tension (sag) based on .... 2524 tons/hour.

N.B.!! The tons per hour caculated excludes the incline factor.

Keeve Steyn Incorporated

PORTION OF BELT OVERLOADED BY CARRY OVER DURING STOPPING

Material carried over............ 5 Tons (From "Carryo") (inc. mat.not held in chute)

Conveyor capacities	RECEIVING CONVEYOR	SHORTEST CONVEYOR	
Maximum cap. of belt	0.280 m^2	0.348 m^2	(At the point of spill)
Restricted cap. of belt	0.247 m^2	0.247 m^2	(Baffle or chute opening)
Selection (max=0, rest=1)	0	0	
Selected capacity	0.247 m^2	0.247 m^2	

RECEIVING CONVEYOR	FEED LENGTH (m)	FEED RATE (T/h)	BELT SPEED (m/s)	MATERIAL DENSITY (T/m^3)	CROSS SEC.AREA (m^2)	TOTAL MATERIAL (Tons)
NORMAL	76.18	1800	3.34	0.85	0.1761	11.40
O/LOAD	23.82	2524	3.34	0.85	0.2470	5.00
TOTAL	100.00	1973	3.34	0.85	N/A	16.40

The conveyor has therefore a minimum power requirement based on .... 1973 tons/hour.
and a minimum tension (sag) based on .... 2524 tons/hour.

SHORTEST CONVEYOR	FEED LENGTH (m)	FEED RATE (T/h)	BELT SPEED (m/s)	MATERIAL DENSITY (T/m^3)	CROSS SEC.AREA (m^2)	TOTAL MATERIAL (Tons)
NORMAL	-3.82	1800	3.44	0.85	0.1710	-0.55
O/LOAD	23.82	2600	3.44	0.85	0.2470	5.00
TOTAL	20.00	2600	3.44	0.85	N/A	4.20

The conveyor has therefore a minimum power requirement based on .... 2600 tons/hour.
and a minimum tension (sag) based on .... 2600 tons/hour.

N.B.!! The tons per hour caculated excludes the incline factor.

THE EFFECTS OF VARYING STOPPING TIMES

Whilst it is obvious that the design of the chute is such that it can both contain any material conveyed over during stoping and discharge that material totally upon re-starting of the system. The discharge of the material from that chute has to be considered.

Under normal circumstances the flow rate is only restricted by the height and width of the feed skirts. In most cases this will grossly overload the receiving conveyor and spillage will occur.

FILE : IDLERCAP

Keeve Steyn Incorporated

PAGE 1
WRITTEN : DCM (KSI)
UPDATED : 13-Jul-91
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3-ROLL BELT CAPACITY AND EDGE CLEARENCE PROGRAMME

STANDARD SELECTED	1 ISO

STANDARDS AVAILABLE	1	ISO
	2	CEMA
	3	DIN
	4	BS 2890

Data

Material depth	0.265	m
Offset(0)in line(1)	0	type
Belt width	1.350	m
Troughing angle	35	deg
Edge dist	0.092	m
Edge dist (forced)	0.105	m
Surcharge ang. (ISO)	18	deg
Surcharge ang. (DIN)	12.00	deg
Material width	1.025	m
Cross sec.area	0.196	m^2
Calc. belt width	1.350	m
Belt speed	3.34	m/s
Material density	0.85	t/m^3
Incline angle	5	deg
Incline factor	1	
Belt capacity	2000	t/h

Areas

radius	1.658	m
area 1	0.092	m^2
area 2	0.048	m^2
area 3	0.056	m^2
total	0.196	m^2

Standard edge dist.	0.0925	m

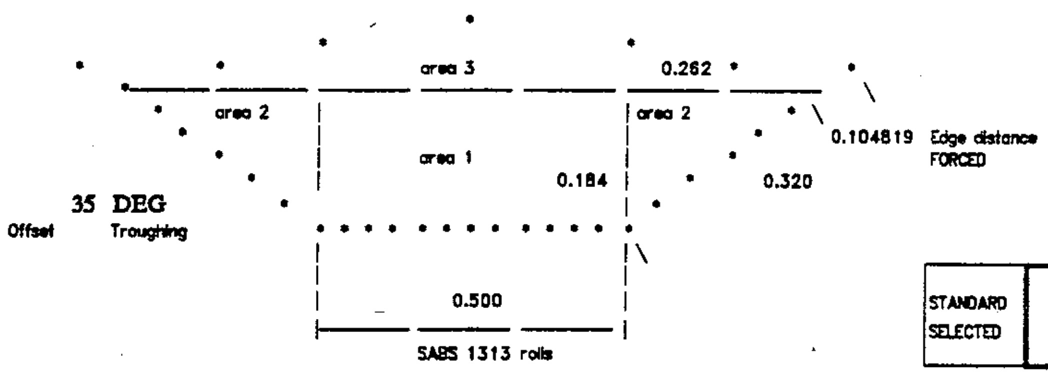
FILE : IDLERCAP

Keeve Steyn Incorporated

PAGE 2
WRITTEN : DCM (KSI)
UPDATED : 13-Jul-91
PRT.DATE: 14-Jul-91

3-ROLL BELT CAPACITY AND EDGE CLEARENCE PROGRAMME

CROSS SECTIONAL
AREA = 0.195687 m^2



STANDARD SELECTED	1 ISO

CONVEYOR PROFILE

FILE : CARRYO

Keeve Steyn Incorporated

PAGE 1
WRITTEN : CJJ (KSI)
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PRT.DATE : 13-Jul-91

MATERIAL ACCUMULATION AT TRANSFER POINT DURING EMERGENCY STOPPING

CONDITION
Emergency stop of conveyor
CON 2 in the INCLINES
loaded condition.

Conveyor	CON1 details	
Max.cap belt	2000	T/hr
Opp.tonnage	2000	T/hr
Belt width	1200	mm
Conv.length	2500	m
Conv. lift	-3.0	m
Belt velocity	3.42	m/s
Mat'l density	0.85	T/m^3
Deceleration	0.190	m/s^2
Acceleration	0	m/s^2

STOPPING TIME 18.00 sec

Conveyor	CON 2 details	
Max.cap belt	2400	T/hr
Opp.tonnage	2000	T/hr
Belt width	1350	mm
Conv.length	100	m
Conv. lift	5.0	m
Belt velocity	3.42	m/s
Mat'l density	0.85	T/m^3
Deceleration	0.650	m/s^2
Acceleration	0	m/s^2

STOPPING TIME 5.26 sec

Calculated stopping condition		
Total material discharged from	5.88	M^3
conveyor no. CON1	5.00	Tons
Total material extracted from chute	2.06	M^3
	1.75	Tons
Total material held in chute	3.82	M^3
	3.25	Tons
Total additional material to be	5.88	M^3
handled by conveyor no. CON 2	5.00	Tons
Total capacity of chute	8.00	M^3
	6.80	Tons

FILE : CARRYO

Keeve Steyn Incorporated

PAGE 2
WRITTEN : CJJ (KSI)
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PRT.DATE : 13-Jul-91

MATERIAL ACCUMULATION AT TRANSFER POINT DURING EMERGENCY STOPPING

CONDITION
Emergency stop of conveyor
CON 2 in the INCLINES
loaded condition.

TIME s	BELT VEL m/s	TRAVEL DISTANCE m	DEL'Y RATE T/s	DISCHARGE CUMMULATIVE TONS	BELT VEL m/s	TRAVEL DISTANCE m	EXTRACTION RATE T/s	EXTRACTION CUMMULATIVE TONS	MATERIAL CARRY OVER TONS	MATERIAL CARRY OVER m^3
0	3.42	0.00	0.00	0.00	3.42	0.00	0.00	0.00	0.00	0.00
1	3.23	3.32	0.54	0.54	2.77	3.09	0.54	0.54	0.00	0.00
2	3.04	6.46	0.51	1.05	2.12	5.54	0.51	1.05	0.00	0.00
3	2.85	9.40	0.48	1.53	1.47	7.34	0.38	1.43	0.10	0.12
4	2.66	12.16	0.45	1.98	0.82	8.48	0.22	1.65	0.32	0.38
5	2.47	14.73	0.42	2.39	0.17	8.98	0.10	1.75	0.64	0.76
6	2.28	17.10	0.39	2.78	0.00	9.00	0.00	1.75	1.02	1.20
7	2.09	19.29	0.35	3.13	0.00	9.00	0.00	1.75	1.38	1.62
8	1.90	21.28	0.32	3.46	0.00	9.00	0.00	1.75	1.70	2.00
9	1.71	23.09	0.29	3.75	0.00	9.00	0.00	1.75	2.00	2.35
10	1.52	24.70	0.26	4.01	0.00	9.00	0.00	1.75	2.26	2.66
11	1.33	26.13	0.23	4.24	0.00	9.00	0.00	1.75	2.49	2.93
12	1.14	27.36	0.20	4.44	0.00	9.00	0.00	1.75	2.69	3.17
13	0.95	28.41	0.17	4.61	0.00	9.00	0.00	1.75	2.86	3.37
14	0.76	29.26	0.14	4.75	0.00	9.00	0.00	1.75	3.00	3.53
15	0.57	29.93	0.11	4.86	0.00	9.00	0.00	1.75	3.11	3.66
16	0.38	30.40	0.08	4.94	0.00	9.00	0.00	1.75	3.18	3.75
17	0.19	30.69	0.05	4.98	0.00	9.00	0.00	1.75	3.23	3.80
18	0.00	30.78	0.02	5.00	0.00	9.00	0.00	1.75	3.25	3.82
19	0.00	30.78	0.00	5.00	0.00	9.00	0.00	1.75	3.25	3.82
20	0.00	30.78	0.00	5.00	0.00	9.00	0.00	1.75	3.25	3.82
21	0.00	30.78	0.00	5.00	0.00	9.00	0.00	1.75	3.25	3.82
22	0.00	30.78	0.00	5.00	0.00	9.00	0.00	1.75	3.25	3.82
23	0.00	30.78	0.00	5.00	0.00	9.00	0.00	1.75	3.25	3.82
24	0.00	30.78	0.00	5.00	0.00	9.00	0.00	1.75	3.25	3.82

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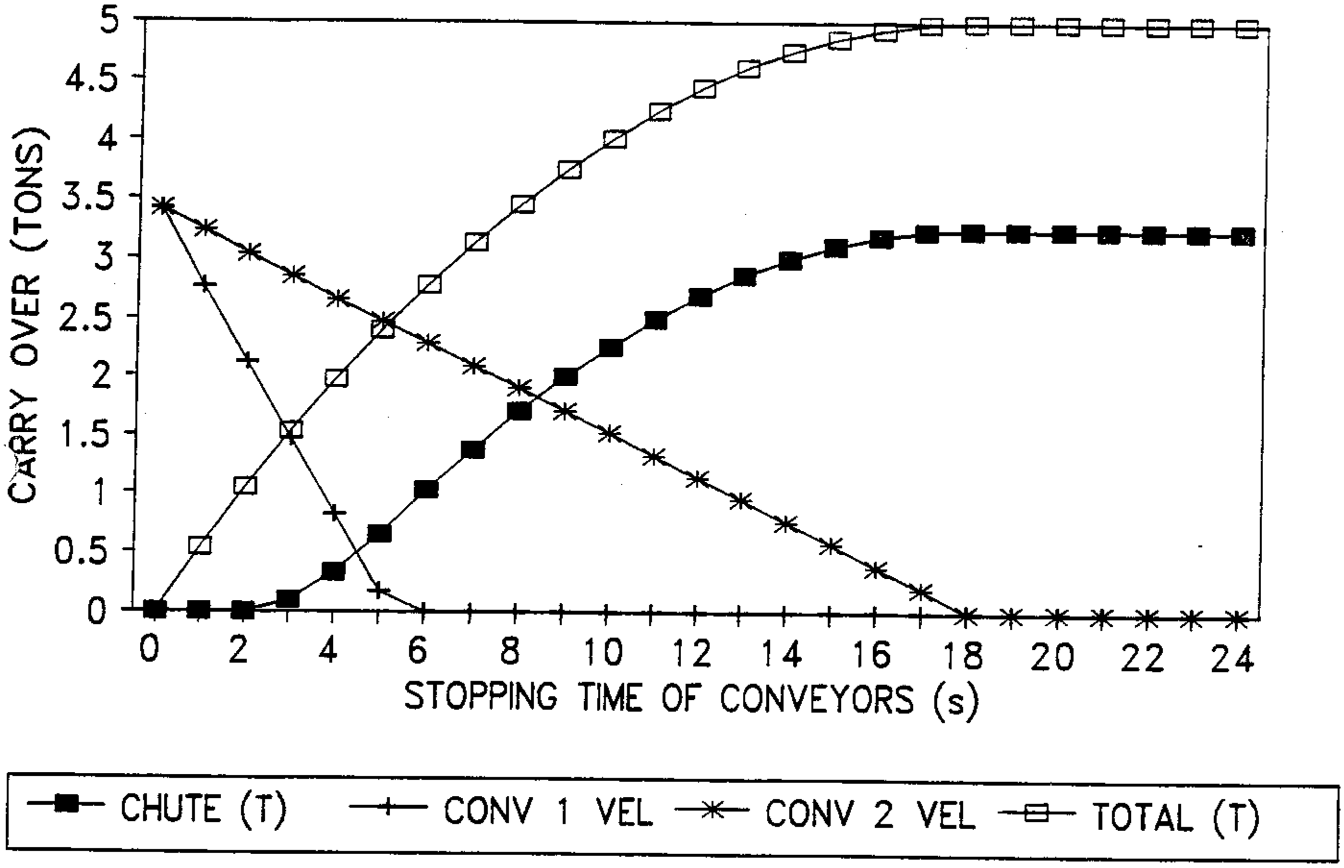
COAL CARRY OVER ANALYSIS

PAGE 3



FILE : TRAJ

Keeve Steyn Incorporated

PAGE 1
WRITTEN : CJJ (KSI)
UPDATED : 13-Jul-91
PRT.DATE: 15-Jul-91

3 ROLL CONVEYOR TRAJECTORY PLOTTING PROGRAMME

INPUT DATA

Belt velocity	3.34	m/s
Pulley diameter	800	mm
Belt thickness	20	mm
Belt width	1350	mm
Material surcharge(ISO)	18	Deg
Idler troughing angle	35	Deg
Incline angle	0	Deg
Material density	0.85	T/M^3
Feed rate (solve for)	2000	T/h

OUTPUT DATA

Material depth	265	mm
Effective rad. on belt line	420	mm
Effective rad. at mat'l top	685	mm
Angular velocity (top)	7.95	rad/s
Angular velocity (btm)	7.95	rad/s
Pulley velocity	75.94	rpm
Tangential velocity (top)	5.45	m/s
Tangential velocity (btm)	3.34	m/s
Angle of contact (top)	0.00	rad
Angle of contact (btm)	0.00	rad
VS^2/G (top)	1.66	
VS^2/G (btm)	2.71	
cos of incline angle	1	

FILE : TRAJ

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PAGE 2
WRITTEN : CJJ (KSI)
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PRT.DATE: 15-Jul-91

3 ROLL CONVEYOR TRAJECTORY PLOTTING PROGRAMME

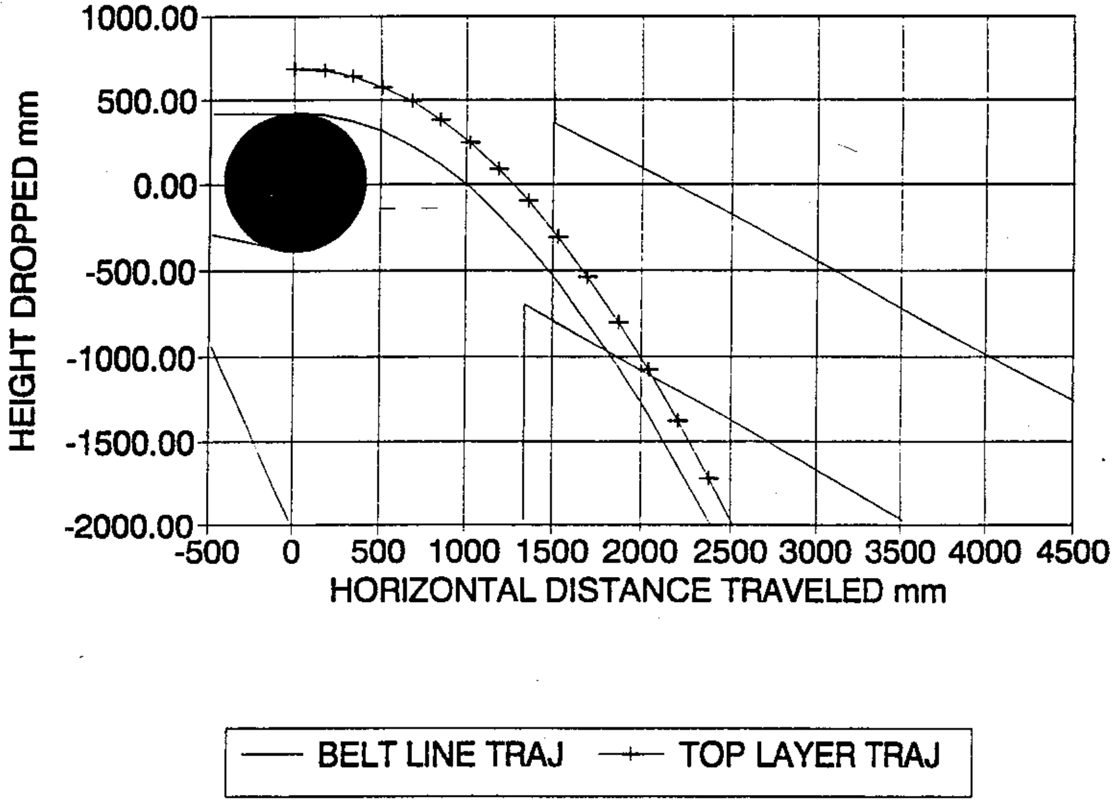
BTM OF MATERIAL m= 0.00

X1	D1	X axis	Y axis	NEW Y axis
		0.00	420.00	684.78
		0.00	420.00	684.78
167.00	12.26	167.00	407.74	672.52
334.00	49.05	334.00	370.95	635.73
501.00	110.36	501.00	309.64	574.42
668.00	196.20	668.00	223.80	488.58
835.00	306.56	835.00	113.44	378.22
1002.00	441.45	1002.00	-21.45	243.33
1169.00	600.86	1169.00	-180.86	83.92
1336.00	784.80	1336.00	-364.80	-100.02
1503.00	993.26	1503.00	-573.26	-308.48
1670.00	1226.25	1670.00	-806.25	-541.47
1837.00	1483.76	1837.00	-1063.76	-798.98
2004.00	1765.80	2004.00	-1345.80	-1081.02
2171.00	2072.36	2171.00	-1652.36	-1387.58
2338.00	2403.45	2338.00	-1983.45	-1718.67
2505.00	2759.06	2505.00	-2339.06	-2074.28

TOP OF MATERIAL m= 0.00

X1	D1	X axis	Y axis	NEW X axis	NEW Y axis
		0.00	684.78	0.00	684.78
		0.00	684.78	0.00	684.78
167.00	12.26	167.00	672.52	167.00	672.52
334.00	49.05	334.00	635.73	334.00	635.73
501.00	110.36	501.00	574.42	501.00	574.42
668.00	196.20	668.00	488.58	668.00	488.58
835.00	306.56	835.00	378.22	835.00	378.22
1002.00	441.45	1002.00	243.33	1002.00	243.33
1169.00	600.86	1169.00	83.92	1169.00	83.92
1336.00	784.80	1336.00	-100.02	1336.00	-100.02
1503.00	993.26	1503.00	-308.48	1503.00	-308.46
1670.00	1226.25	1670.00	-541.47	1670.00	-541.47
1837.00	1483.76	1837.00	-798.98	1837.00	-798.98
2004.00	1765.80	2004.00	-1081.02	2004.00	-1081.02
2171.00	2072.36	2171.00	-1387.58	2171.00	-1387.58
2338.00	2403.45	2338.00	-1718.67	2338.00	-1718.67
2505.00	2759.06	2505.00	-2074.28	2505.00	-2074.28

COAL TRAJECTORY



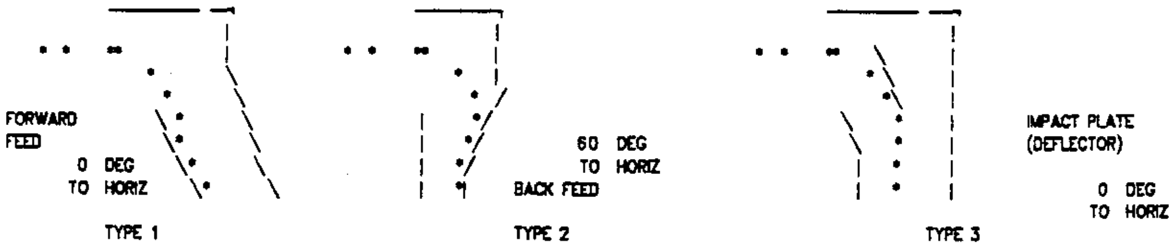
FILE : IMPACT

Keeve Steyn Incorporated

PAGE 1

CHUTE ANGLES AND ITS EFFECT ON MATERIAL IMPACT

WRITTEN D.C.MORGAN
UPDATE : 10-Jul-91
PRT.DATE 15-Jul-91
PRT.TIME 03:09:42 PM



CHUTE TYPE 2 SELECTED

INPUT DATA

CHUTE ANGLE (TO HORIZONTAL)	60	DEG	FRICTION ANGLE	30	DEG
INITIAL VELOCITY (HORIZONTAL)	3.42	M/S	CO-EF OF RESTITUTION (e)	0.7	
INITIAL VELOCITY (VERTICAL)	0	M/S	MEAN BREAKAGE ENERGY	35	
SPECIFIC MASS OF MATERIAL	1.4	T/M^3	PARTICAL SIZE	10	MM
VERTICAL FALL INCREMENT	0.4	M	PARTICAL MASS	0.0014	KG

FILE : IMPACT

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PAGE 2

CHUTE ANGLES AND ITS EFFECT ON MATERIAL IMPACT

CHUTE TYPE 2 SELECTED

WRITTEN D.C.MORGAN
DATE : 10-Jul-91
PRT.DATE 15-Jul-91
PRT.TIME 03:09:42 PM

VERTICAL1 FALL (MTRS)	VERTICAL VELOCITY (M/S)	HORIZ. VELOCITY (M/S)	RESULT. VELOCITY (M/S)	ANGLE OF TRAJ. (DEG)	IMPACT ANGLE (DEG)	μ	IMPACT LOSSES ΔEI/E	SLIDING LOSSES ΔEs/E	ENERGY LOSSES ΔE/E	INITIAL ENERGY E	ENERGY LOSSES ΔE
0	0.000	3.420	3.420	0.00	-60.00	0.5774	0.3825	-1.5725	-1.1900	0.008	-0.010
0.4	2.801	3.420	4.421	39.32	80.68	0.1642	0.4966	0.0134	0.5100	0.014	0.007
0.8	3.962	3.420	5.234	49.20	70.80	0.3482	0.4549	0.0551	0.5100	0.019	0.010
1.2	4.852	3.420	5.936	54.82	65.18	0.4625	0.4201	0.0899	0.5100	0.025	0.013
1.6	5.603	3.420	6.564	58.60	61.40	0.5452	0.3931	0.1169	0.5100	0.030	0.015
2	6.264	3.420	7.137	61.37	58.63	0.5774	0.3718	0.1701	0.5419	0.036	0.019
2.4	6.862	3.420	7.667	63.51	56.49	0.5774	0.3546	0.2339	0.5884	0.041	0.024
2.8	7.412	3.420	8.163	65.23	54.77	0.5774	0.3403	0.2822	0.6225	0.047	0.029
3.2	7.924	3.420	8.630	66.65	53.35	0.5774	0.3282	0.3201	0.6484	0.052	0.034
3.6	8.404	3.420	9.073	67.86	52.14	0.5774	0.3179	0.3506	0.6685	0.058	0.039
4	8.859	3.420	9.496	68.89	51.11	0.5774	0.3090	0.3757	0.6846	0.063	0.043
4.4	9.291	3.420	9.901	69.79	50.21	0.5774	0.3011	0.3966	0.6977	0.069	0.048
4.8	9.704	3.420	10.289	70.59	49.41	0.5774	0.2941	0.4143	0.7084	0.074	0.053
5.2	10.101	3.420	10.664	71.29	48.71	0.5774	0.2879	0.4295	0.7174	0.080	0.057
5.6	10.482	3.420	11.026	71.93	48.07	0.5774	0.2823	0.4427	0.7250	0.085	0.062
6	10.850	3.420	11.376	72.50	47.50	0.5774	0.2772	0.4542	0.7314	0.091	0.066
6.4	11.206	3.420	11.716	73.03	46.97	0.5774	0.2725	0.4644	0.7369	0.096	0.071
6.8	11.551	3.420	12.046	73.51	46.49	0.5774	0.2683	0.4734	0.7417	0.102	0.075
7.2	11.885	3.420	12.368	73.95	46.05	0.5774	0.2644	0.4815	0.7458	0.107	0.080
7.6	12.211	3.420	12.681	74.35	45.65	0.5774	0.2608	0.4887	0.7495	0.113	0.084
8	12.528	3.420	12.987	74.73	45.27	0.5774	0.2574	0.4953	0.7527	0.118	0.089

FILE : IMPACT

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PAGE 3

CHUTE ANGLES AND ITS EFFECT ON MATERIAL IMPACT

CHUTE TYPE 2 SELECTED

WRITTEN D.C.MORGAN
DATE : 10-Jul-91
PRT.DATE 15-Jul-91
PRT.TIME 03:09:42 PM

VERTICAL FALL (MTRS)	100 TO 53	53.0 TO 37.5	37.5 TO 26.5	26.5 TO 19.0	19.0 TO 13.2	13.2 TO 9.5	9.5 TO 6.7	6.7 TO 0.0	TOTALS
0	0.17%	0.02%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.20%
0.4	0.09%	0.01%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.10%
0.8	0.17%	0.02%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.20%
1.2	0.29%	0.04%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.33%
1.6	0.43%	0.05%	0.01%	0.00%	0.00%	0.00%	0.00%	0.00%	0.49%
2	0.68%	0.08%	0.01%	0.00%	0.00%	0.00%	0.00%	0.00%	0.77%
2.4	1.06%	0.13%	0.02%	0.00%	0.00%	0.00%	0.00%	0.00%	1.21%
2.8	1.53%	0.19%	0.02%	0.00%	0.00%	0.00%	0.00%	0.00%	1.74%
3.2	2.07%	0.26%	0.03%	0.00%	0.00%	0.00%	0.00%	0.00%	2.36%
3.6	2.69%	0.34%	0.04%	0.01%	0.00%	0.00%	0.00%	0.00%	3.07%
4	3.38%	0.42%	0.05%	0.01%	0.00%	0.00%	0.00%	0.00%	3.86%
4.4	4.15%	0.52%	0.06%	0.01%	0.00%	0.00%	0.00%	0.00%	4.74%
4.8	4.99%	0.63%	0.08%	0.01%	0.00%	0.00%	0.00%	0.00%	5.70%
5.2	5.90%	0.74%	0.09%	0.01%	0.00%	0.00%	0.00%	0.00%	6.75%
5.6	6.89%	0.86%	0.11%	0.01%	0.00%	0.00%	0.00%	0.00%	7.87%
6	7.94%	1.00%	0.12%	0.02%	0.00%	0.00%	0.00%	0.00%	9.08%
6.4	9.07%	1.14%	0.14%	0.02%	0.00%	0.00%	0.00%	0.00%	10.37%
6.8	10.27%	1.29%	0.16%	0.02%	0.00%	0.00%	0.00%	0.00%	11.74%
7.2	11.54%	1.45%	0.18%	0.02%	0.00%	0.00%	0.00%	0.00%	13.19%
7.6	12.88%	1.62%	0.20%	0.03%	0.00%	0.00%	0.00%	0.00%	14.73%
8	14.29%	1.79%	0.22%	0.03%	0.00%	0.00%	0.00%	0.00%	16.34%

FILE : IMPACT

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PAGE 4

CHUTE ANGLES AND ITS EFFECT ON MATERIAL IMPACT

CHUTE TYPE 2 SELECTED

WRITTEN D.C.MORGAN
DATE : 10-Jul-91
PRT.DATE 15-Jul-91
PRT.TIME 03:09:42 PM

VERT.FALL 4.8 SELECTED

PARTICAL MAX MIN	100 TO 53.0	53.0 TO 37.5	37.5 TO 26.5	26.5 TO 19.0	19.0 TO 13.2	13.2 TO 9.5	9.5 TO 6.7	6.7 TO 0.0	TOTALS
INITIAL	10.20%	5.60%	7.30%	8.80%	10.30%	9.00%	8.90%	39.90%	100.00%
53.0	6.05%	0.00%	0.00%	0.00%	0.03%	0.18%	0.82%	3.12%	10.20%
37.5		5.08%	0.00%	0.00%	0.00%	0.01%	0.09%	0.42%	5.60%
26.5			7.24%	0.00%	0.00%	0.00%	0.01%	0.05%	7.30%
19.0				8.79%	0.00%	0.00%	0.00%	0.01%	8.80%
13.2					10.30%	0.00%	0.00%	0.00%	10.30%
9.5						9.00%	0.00%	0.00%	9.00%
6.7							8.90%	0.00%	8.90%
0.0								39.90%	39.90%
FINAL	6.05%	5.08%	7.24%	8.79%	10.33%	9.19%	9.82%	43.50%	100.00%

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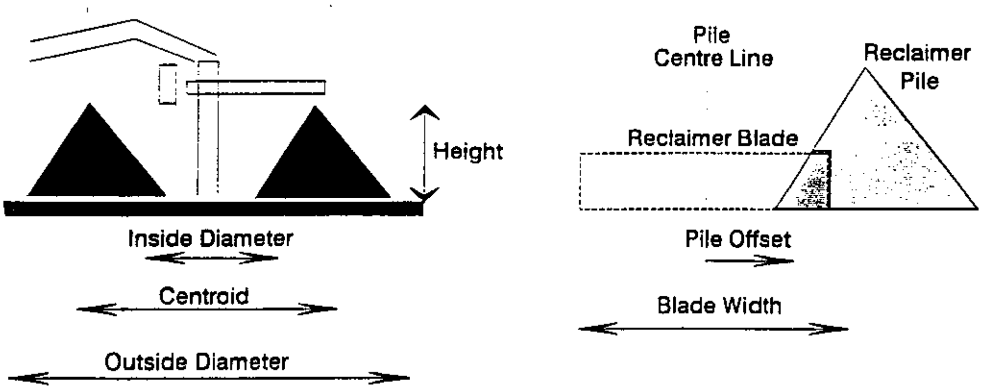
CIRCULAR STOCKYARD CALCULATION



INPUT DATA	
OUTER DIAMETER OF PILE	87,420 METRES
INNER DIAMETER OF PILE	6,600 METRES
ANGLE OF REPOSE	37,000 °
BULK DENSITY OF MAT'L	0,850 T/m^3

OUTPUT DATA	
HEIGHT OF STOCKPILE	15,226 METRES
DIAMETER OF CENTRIOD	47,010 METRES
DEG FULL PILE	232,728 °
DEG WEDGE	59,272 °

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SPLITTER BIN CAPACITY

ITEM	DIA	LENGTH	WITDTH	HEIGHT	RATIO	ANGLE	TOTAL VOLUME	QUANTITY	TOTAL CAPACITY
BOTTOM		1,800	0,900	0,400	2,000	0,000	0,648	4	2,203
BASE A		1,800	0,900	1,930	2,000	25,001	6,253	4	21,261
TRANS B		1,800	2,700	0,350	1,500	0,000	1,701	4	5,783
WEDGE C		1,800	2,700	2,895	1,500	25,001	24,622	4	83,715
TRANS D		4,500	2,700	0,900	1,667	0,000	10,935	4	37,179
WEDGE E		4,500	2,700	1,930	1,667	25,001	31,266	4	106,304
BIN F		4,500	4,500	7,129	1,000	0,000	144,367	4	490,846
BIN F		4,500	4,500						
CONE G	9,000			3,516		52,000	74,555	1	63,372
RELIEF H	9,000	9,000	9,000	1,456			-8,438	1	-7,172
								Sum ->	803,491

MATERIAL HEIGHT IN BIN	19,050 METRES
COAL DENSITY	0,850 TONS/m^3
ANGLE OF REPOSE	38 DEG
TOTAL BIN CAPACITY	**803,491 TONS**

KILOWATTS AND KILONEWTONS

CALCULATION OF POWER REQUIREMENTS AND TENSILE FORCES ON CONVEYOR BELTS

S.P. ZAMORANO

SWF BULK MATERIALS HANDLING (PTY) LTD

REP.001/SZ/tb

SUMMARY

The basic considerations required for the calculation of power requirements and tensile forces in conventional idler supported troughed conveyor belts are discussed.

The information of the use of different design standards as well as the importance of the use of engineering criteria are analysed regarding practical applications.

1.0 INTRODUCTION

This paper does not intend to present an "academic" discussion of conveyor belt design, but rather an analysis of the most common practical problems encountered when designing a conveyor belt.

The scope of the discussion is limited to power requirements and rigid body tensile forces on conventional conveyor belts. The calculation and analysis of elasto-dynamic forces as well as the implications of the use of horizontal curves are not treated here, as they are regarded as "specialities" and are not part of the routine design.

2.0 DESIGN STANDARDS

To date, many different standards for the design of conveyor belts have been written, within them, CEMA, (1) is perhaps the most famous, however, ISO 5084 (2) is becoming increasingly recognised as "the" international standard. Other recognised standards are the Good Year (3) and the British MHEA (4) between many others.

The main difference between the different standards is the form in which the conveyor belt resistances are calculated.

3.0 MAIN RESISTANCES

The main resistances in a conveyor belt can be divided into lifting and frictional resistances.

3.1 Lifting Resistance

This is one of the few parameters in the design of a conveyor belt that can be determined accurately. The lifting resistance is the force required to lift the material carried by the conveyor belt, from the feed point to the discharge point, is given by:

$$F_L = C_V \quad H \quad g \qquad [1]$$

Where
- C_V = Mass of material handled per meter
- H = Lift of the conveyor between load and unload area
- g = Gravity constant (9,81m/s²)

This resistance can be positive, null or negative, depending on the slope of the conveyor, and can vary along a complex profile conveyor (typically overland conveyors)

3.2 Frictional Resistance

The frictional resistance on a conveyor belt can be separated in two main components.

- Rotational resistance of the carrying and return strands idlers due to friction in the idler bearing and seals. This resistance is mainly dependant on the type of bearing and seals used.

- Belt resistance, formed by the resistance caused by the belt indentation on the idler, the resistance produced by the belt sag between idlers and the resistance produced by the repeated flexure of the belt.

The CEMA standard tries to calculate these two components separately using a factor called kx that accounts for the idler resistance and the resistance due to the slide between the idler and the belt, while another factor called ky is used to calculate the flexure resistance of the belt. An additional factor kt is used to account for the effect of the variation of temperature (due mainly to increased bearing resistance at sub-zero conditions).

ISO on the other hand makes use of a ficticious friction value, taking 0,02 for average applications, while for well aligned installations with low internal friction materials this value can be dropped to 0,016. For badly aligned conveyors and high internal friction materials this value is increased to 0,03 while for downhill regenerative conveyors, this factor is chosen as 0,012 (these are conveyors that "generate" energy instead of absorbing it). In general the selection of the friction is left to the designer and the same factor is used for the carry and return strands.

Good Year uses a system similar to ISO, but the basic friction value is taken as 0,022 (10% higher than ISO), varying also accordingly to external conditions. Although CEMA takes into account the effect of idler spacing, belt tension and material characteristics, it must be noted that important factors like belt material, belt hysteresis and the change of belt properties with temperature are not taken into account.

The use of CEMA standard by unexperienced personnel can easily lead to a false sense of confidence if the person using the method is not conscious that the values on CEMA's tables are not exact values, but rather average values from numerous different applications. However, there are many designers who believe they can calculate the frictional resistance of a conveyor belt within 5% accuracy, when in practice this value can change 20% or more due to different circumstances in the same conveyor.

On the other hand, the use of a constant frictional factor, as recommended by ISO (providing the right criteria is applied), has been proven to be sufficiently accurate for normal applications.

When the accurate calculation of belt tensions is required like in long undulating conveyors, the use of research data relevant to the application must be used (for example references (5) and (6)), and all probable loading and environmental conditions simulated.

The frictional resistance of a conveyor belt can be calculated with the formula:

$F_r = M \, L \, g \, [M_{RO} + M_{RV} + (2 \, M_B + C_V) \, \mathrm{COS} \; x \,]$ [2]

Where	M	=	coefficient of friction
	g	=	constant of gravity
	L	=	conveyor length
	M_{RO}	=	rotating mass of carrying idlers per meter of conveyor
	M_{RV}	=	rotating mass of return idlers per meter of conveyor
	M_B	=	belt mass per meter
	C_V	=	mass of material handled per meter
	x	=	angle of inclination of the conveyor

In case of conveyors with multiple loading or unloading points, or when partial load conditions are to be analysed, equation [2] must be applied separately for the different sections of the belt. "Section" can be defined as a portion of the belt with the same loading condition, slope and idler spacing.

4.0 SECONDARY RESISTANCES

In some standards like Good Year the secondary resistances are calculated by adding an equivalent length to the real length of the conveyor (in a similar way to the hydraulic piping systems); this length is 60m for a "typical" conveyor.

In the case of the MHEA, different friction factors are used for different lengths of conveyors, for belt and idler friction factor, while the equivalent length (for conveyors below 1000m long) is given by:

$$L_c = 0{,}665\ L + 30 \qquad [3]$$

In general the use of MHEA leads to an equivalent friction factor as low as 0.016 on very long conveyors.

CEMA Standard makes provision for the calculation of the force required to accelerate the material and the friction between the material and the skirtplates in the loading area.

In the ISO standard, an approximate value of the secondary resistances can be obtained using a C factor, that multiplies the main frictional resistances. This factor is a function of length and is presented on figure 2. However, in case of short, high capacity or high speed conveyors, the secondary resistances can be calculated in detail with the help of table 1. It must be noted that the use of the C factor is recommended only for conveyors longer than 80m. Special care must be taken with high speed, high capacity conveyors, for example, on a shiploader boom conveyor transporting coal at a rate of 12000 tons per hour at 7 m/2 requires 147 kw to accelerate the material, while about 45kw are required to overcome the frictional resistances, this is of course an extreme example.

5.0 SPECIAL RESISTANCES

These resistances are present only on some conveyors, many different formulae are available and they will not be discussed in detail; formulae for the calculation of some of them is presented on table 2.

6.0 POWER REQUIREMENTS

The power required to drive a conveyor is given by:

$$P = F_T\,V/\eta \qquad [4]$$

F_T = summary of all resistances
V = conveyor belt speed
η = drive efficiency

The question of when to use single or multiple motors is determined by factors like cost and belt tensions as discussed on point 7.

7.0 TRANSMISSION OF POWER TO THE BELT

Power is transmitted to the belt by means of the friction between the drive pulley(s) ad the belt. In general the use of a single drive results in the minimum cost for the drive, but it increases the tensions in the belt, therefore increasing the belt cost.

The minimum tension F_z (see figure 3) required for the transmission of a peripheral force F_u from the driving pulley to the belt is given by:

$$F_z > \frac{F_u}{e^{n\theta} - 1} \qquad [5]$$

Where n = coefficient of friction between the pulley and the belt

θ = angle of wrap around the pulley

The coefficient of friction is given different values by different sources, usually varying according to pulley lagging. For example CEMA proposes a value of 0,25 for unlagged pulleys and 0.35 for lagged pulleys while ISO proposes different values according to the type of lagging and operational conditions (table 3).

It is generally assumed that friction is strictly of coulomb type, this assumption is not strictly valid and values normally used are not actual friction coefficients, but empirical values which are lower in magnitude than the actual values, the empirical coefficients used provide in general a conservative design. Although a lot of research has been done on the subject, the mechanisms of friction development between the pulley and the belts are not fully understood.

It is a common practice to use a reduced friction factor when using "fixed" take-ups (i.e that don't move during operated) in order to compensate for the tension reduction due to belt stretch. However, this method is more of a "thumbsuck" than anything else, the correct procedure is to calculate the belt elastic stretch on loaded condition, and then calculate the necessary extra tension in the take-up in order to "pre-stretch" the belt before the start-up.

The maximum tension in the belt (with a single drive) is given by:

$$Fm = F_u + Fz \qquad [6]$$

The use of dual drives allows the reduction of the minimum tension, as only a fraction of the effective tension is transmitted in the secondary drive, so the minimum tension is reduced.

In general the minimum tension is produced when 1/3 of the power is transmitted by the secondary drive.

In the case of long, horizontal conveyors, the use of a tail drive results in further reduction of the maximum tension, as the resistance of the return strand is taken by this drive.

In the case of long, high lift or very long horizontal conveyors, where belt tensions are too high, maximum tension can be further reduced with the use of linear or "booster" drives. In this system, that consist of a booster belt located underneath the main belt, power transmission takes place by the coulomb friction between the two belts (see figure 7). No design standard exists for this sort of drive, and their design is regarded as an speciality.

It is a common belief that the power sharing ration between two drives can be changed by changing their respective angles of wrap, this is not true, as the angle of wrap only determines (for a given Fz tension) the maximum power that can be transmitted without slip on the pulley.

8.0 MINIMUM BELT TENSION

Further to the minimum tension required to prevent slip, a minimum tension is required to prevent excessive sag of the belt between idlers.

The sag between idlers is given by:

$$F_s > \frac{l\,(M_B + C_V)\,g}{8\ .\ S} \qquad [7]$$

Where

l = spacing between idlers
s = belt sag

The belt sag is usually limited to a 2% (0.02), both for the carry and return strands (CEMA allows 3% sag). Sag should be calculated for different loading conditions in the case of complex profile conveyors.

9.0 CONVEYOR BELT CAPACITY

In the previous points, the load on the belt has been used for calculations, but which load is used for which calculation has not been defined.

When talking about conveyor capacity many different terms are used:

- Average capacity
- Maximum "flow sheet" capacity
- Peak capacity
- Design capacity
- Flooded belt capacity

Many times the metallurgical or mining engineer calculates the maximum load ever to be present in the conveyor and calls it design capacity, when in fact this value should be treated as a peak capacity. The following criteria should be used after checking the nature of the loading:-

- **Average capacity**
  Volumetric loading should be 90% or less, drive capacity should be ample to handle the required throughput.

- **Maximum "flow sheet"**
  After the probability of this value to increase has been assessed as being minimal, the belt can be designed to be 100% loaded (providing good control can be exerted on loading, eg via feeders) and the drive 100% utilised. However, care must be exercised mainly in the case of process plants, where conditions are likely to change dramatically.

- On **peak capacity** the conveyor should be 100% full and drive can be up to 10% overload (peak capacity should be experienced for no more than 30 minutes or a belt cycle, whichever longer).

- **Design capacity**
  Conveyor should be loaded close to 100%, depending on the ability to control the flow (in case of cyclic loads like in crusher and BW reclaimers volumetric utilisation should be reduced), the same applies for drive selection.

- **Flooded belt capacity**
  On this condition belt is assumed to be loaded up to the edge, the drive is required to start up and empty the belt, drive overload of 20% or more can be allowed, depending of motor characteristics. Obviously this capacity is only required if the flooding of the belt is probable.

As with the lifting resistance the volumetric capacity of a conveyor belt is calculated in quite a standard way, using the cross section of the loaded belt (figures 4 to 6), the edge distance varies slightly between standards, for example in ISO its given by:

$$d = 0{,}05\ W + 0{,}025\ (mm) \qquad [8]$$

with W = belt width, while Good Year uses the formulae

$$d = 0{,}055\ W + 0{,}9\ (mm) \qquad [9]$$

The DIN standard (7) is slightly different, calculating the top part of the area as a triangle and using the dynamic angle of slope instead of the surcharge angle used by other standards, however, equivalence factor with ISO are provided in the same code.

10.0 BELT CLASS

The main mechanism to determine the required belt class is through the safety factor, that generally is 10 for fabric belts and 6.7 for steel cord belts (GOOD YEAR), these factors already include an allowance for start up factors between 1.3 and 1.6, therefore, if dynamic forces are known, a relevant safety factor of 6.25 for fabric and 4.2 for steel cord could be used. However, care must be taken when using these factors, not only quasi-static forces must be calculated but also elasto-dynamic transient forces.

When a complete elasto-dynamic analysis is performed a safety factor of 4 for static condition and 3 for transient conditions can be used, in the case of steel cord belt. Actually the limiting factor on steel cord belts is the fatigue resistance of the belt splice, that varies between 28 and 50% of the belt rating.

In the case of long, high capacity conveyors the use of traditional safety factors is no longer appropriate and elasto-dynamic forces must be calculated through a proper analysis method.

11.0 STANDARDISATION V'S RATIONALISATION

No magical formula exist to determine when to use single or multiple drives or when to use higher or lower class belts, the decision must be taken on a one tc one basis, while experience allows the formation of a "data base" that helps to take design decisions.

A design parameter commonly used in our days is standardisation, ie, the maximum number of standard components is used, even if these components are over designed for a particular application. In this form spare and inventory costs are reduced and maintenance simplified. However, sometimes these otherwise healthy principle is taken up to extremes, and for example a 30kW drive is used for a conveyor with an absorbed power of 3.5kW (this is a real life example), the cost of a 7.5kW drive (already oversized) plus a complete spare drive is about 2/3 of the cost of the 30kW unit.

Maybe rationalisation should replace standardisation as the way to follow, in other words the emphasis is put in reducing the overall capital and operational cost making use of standardisation when required.

Over dimensioning is regarded as a healthy practice in the materials handling industry, but an oversized drive requires an oversized belt that requires oversized pulleys and so on. When for example only the drive is oversized, the probability of damaging the conveyor increases dramatically.

There are many examples when drive oversizing, with no allowance for dynamic forces had caused conveyor failure.

12.0 PRACTICAL APPLICATION

An existing silo feed conveyor (See figure 8) is analysed and two alternative designs compared.

12.1 Data

Material	coal, sized
Bulk density	900kg/m³
Surcharge anglr	10° (original design 20° (actual design value)
Capacity	1200 tph
Trough angle	35°

12.2 Original Design

Designed by consultants in 1989. According to ISO 5048, using C factor plus special resistances.

Belt width	1600 mm
Belt speed	2,25 m/s
Volumetric loading (10° surcharge angle)	58.1 %
Idler spacing	1.25 carry 2.25 return
Belt class	ST 1250 15/5 covers
Belt mass	52.7kg/m
Absorbed power	231.09 kW
Installed power	400 kW
Brake	1050 N.m (high speed)
Coasting time	2.05 sec
Braking time	1.25 sec
Holdback	130 kN.m
Take up mass	20 tons
Drive pulley	1300/280 mm bearings
Low tension pulleys	1000/160 mm bearings
Gearbox	400 kw 43.863:1

12.3 Alternative Design

Belt width	1200 mm
Belt speed	3.5 m/s
Volumetric loading (20° surcharge angle)	64.8 %
Idler spacing	2.25 carry .5 return

Belt class	800 kN/m 6.3/3 covers
Belt mass	17 kg/m
Absorbed power	205 kW
Installed power	220 kW
Brake	None
Coasting time	8 sec
Braking time	-
Hold back	22 kN.m
Take up mass	8 tons
Drive pulley	650/160 mm bearings
Low tensiqon pulleys	500/125 mm bearings
Gearbox	220 kW 14.1:1

12.4 **Cost Comparison (estimated prices)**

ITEM	CURRENT DESIGN	NEW DESIGN
BELTING	232 000	113 000
DRIVE	240 000	111 000
PULLEYS	128 000	43 000
IDLERS	102 000	46 000
STEELWORK	840 000	650 000
TOTAL	**1 542 000**	**963 000**

__NOTE:__ PRICES IN RANDS, 1 USS = 2,9 RANDS, BASE AUGUST 1991

12.5 **Discussion**

The two designs presented here are based on ISO 5048, the huge differences in specifications and costs are mainly based on different engineering criteria.

On the original design a friction factor (for the belt) of 0.024 was used, compared with a factor of 0.02 for the alternative design, however, this factor has little influence as most of the power is used to lift the material.

Both designs aim to a volumetric utilisation of the belt of 2/3 of the maximum, but the original design uses an ultraconservative 10° surcharge angle, and a relative low speed conveyor. This low speed is not justifed as material degradation is not a concern.

On the original design, the belt has an expected wear life of 35 years, while in the alternative this life is about 15 years (using Continental's formulae). However, chances are that belts will be replaced within 8 years due to side damage, ply separation, belt tearing, etc. In other words, the use of a thicker belt does not guarantee a longer life.

This comparison is not presented as a critic to the original design (that was motivated mainly by standardisation contraints), but as a way to illustrate that the criteria used in the design is the paramount factor for the final result, and even when using the same design standard, totally different results can be produced.

For further illustration, calculations for the alternative design, using ISO "C" factor, and calculating secondary resistances, as well as results of a program specialised on drive component specification, are enclosed.

13.0 CONCLUSION

Adequate conveyor belt design is the result of the application of common sense, experience and engineering criteria. Design standards should be regarded as guidelines and not as the "bible".

REFERENCES

1. BELT CONVEYORS FOR BULK MATERIALS,M CONVEYOR EQUIPMENT MANUFACTURERS ASSOCIATION (CEMA), 1979

2. INTERNATIONAL STANDARD ISO 5084, 1979

3. HANDBOOK OF CONVEYOR AND ELEVATOR BELTING, GOOD YEAR, 1976

4. RECOMMENDED PRACTICE FOR TROUGHED BELT CONVEYORS, MECHANICAL HANDLING ENGINEERS ASSOCIATION, 1986

5. ROBERTS, A.W; HARRISON, A; RECENT RESEARCH DEVELOPMENTS IN BELT CONVEYOR TECHNOLOGY. BELTCON 5, JOHANNESBURG, 1989

6. GREVNE, A; HAGER, A - THE ENERGY-SAVING DESIGN OF BELT CONVEYORS; BULK SOLIDS HANDLING, AUGUST, 1990

7. DIN STANDARD 22101

Table 1 — Rules for calculating the secondary resistances

Symbol	Type of resistance	Unit
F_{bA}	Inertial and frictional resistance at the loading point and in the acceleration area between the handled material and the belt : $F_{bA} = I_V \gamma (v - v_o)$	N
F_f	Frictional resistance between handled material and the skirtplates in the acceleration area : $F_f = \dfrac{\mu_2 I_V^2 \gamma g l_b}{\left(\dfrac{v + v_o}{2}\right)^2 b_1^2}$	N
F_l	Wrap resistance between belt and pulleys : a) for fabric carcass belts : $F_l = 9 B \left(140 + 0{,}01 \dfrac{F}{B}\right) \dfrac{d}{D}$ b) for metal carcass belts : $F_l = 12 B \left(200 + 0{,}01 \dfrac{F}{B}\right) \dfrac{d}{D}$	N
F_t	Pulley bearing resistance (not to be calculated for the driving pulleys) : $F_t = 0{,}005 \dfrac{d_o}{D} F_T$	N

Symbol	Description	Unit
b_1	inter-skirtplate width	m
D	pulley diameter	m
d	belt thickness	m
d_o	shaft diameter inside bearing	m
F	average belt tension at the pulley	N
F_T	vectorial sum of the two belt tensions acting on the pulley and of the forces due to the mass of the revolving parts of the pulley	N
l_b	acceleration length $l_{b\,min} = \dfrac{v^2 - v_o^2}{2 g \mu_1}$	m
v_o	handled material conveying speed component in the belt moving direction	m/s
μ_1	0,5 to 0,7, friction coefficient between material and belt	—
μ_2	0,5 to 0,7, friction coefficient between material and skirtplates	—
I_V	conveyor capacity	m^3/s

Table 2 — Rules for calculating the special resistances

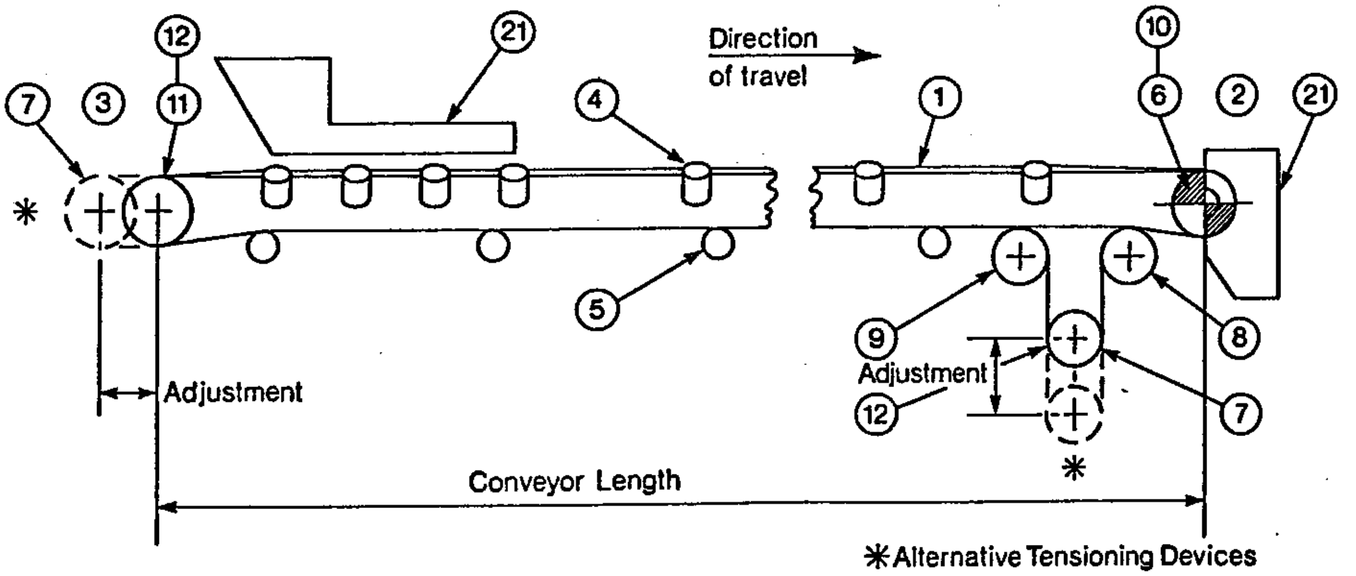
Symbol	Type of resistance	Unit
F_ε	Resistance due to idler tilting a) Case of carrying idlers equipped with three equal length rollers : $F_\varepsilon = C_\varepsilon \mu_0 L_\varepsilon (q_B + q_G) g \cos\delta \sin\varepsilon$ b) Case of return idlers equipped with two rollers : $F_\varepsilon = \mu_0 L_\varepsilon q_B g \cos\lambda \cos\delta \sin\varepsilon$	N
F_{gL}	Resistance due to friction between handled material and skirtplates : $F_{gL} = \dfrac{\mu_2 I_V^2 \gamma g l}{v^2 b_1^2}$	N
F_r	Friction resistance due to the belt cleaners : $F_r = A p \mu_3$	N
F_a	Resistance due to friction at a discharge plough : $F_a = B k_a$	N

Symbol	Description	Unit
A	contact area between belt and belt cleaner	m^2
b_1	inter-skirtplate width	m
C_ε	0,4 trough factor for 30° trough	—
	0,5 trough factor for 45° trough	—
k_a	scraping factor (normally 1 500 N/m)	N/m
L_ε	length of the installation equipped with tilted idlers	m
l	length of the installation equipped with skirtplates	m
p	pressure between belt cleaner and belt (normally 3×10^4 to 10×10^4 N/m^2)	N/m^2
ε	tilt angle of the idler axis with respect to the plane perpendicular to the belt longitudinal axis	degrees
μ_0	0,3 to 0,4 friction coefficient between carrying idlers and belt	—
μ_2	0,5 and 0,7, friction coefficient between material and skirtplates	—
μ_3	friction coefficient between belt and belt cleaner	—

Table 3 — Friction coefficient, μ, between driving pulleys and rubber belting

Pulley lagging / Operating conditions	Smooth bare rim steel pulley	Rubber lagging with herringbone-patterned grooves	Polyurethane lagging with herringbone-patterned grooves	Ceramic lagging with herringbone-patterned grooves
Dry condition operation	0,35 to 0,4	0,4 to 0,45	0,35 to 0,4	0,4 to 0,45
Clean wet condition (water) operation	0,1	0,35	0,35	0,35 to 0,4
Operation under wet and dirty (clay or loam) conditions	0,05 to 0,1	0,25 to 0,3	0,2	0,35

1. *The Belt.* Carries the material and transmits the power.
2. *Head of Conveyor.* The discharge end of the conveyor.
3. *Tail of Conveyor.* The loading end of the conveyor.
4. *Carrying Idlers.* Idlers which support the loaded belt. An assembly of one or more carrying idlers, suitably mounted, comprises a carrying idler set.
5. *Return Idlers.* Idlers which support the empty side of belt. An assembly of one or more return idlers, suitably mounted, comprises a return idler set.
6. *Drive.* The equipment which drives the belt, comprising power unit, transmission and driving pulley or pulleys.
   Power Unit. Motor or engine.
   Transmission. An assembly of devices coupling the power unit to the drive pulley/pulleys to drive the belt at the desired speed.
   Driving Pulley. A pulley which drives the belt.
7. *Take-up Device.* A device for taking up slack and applying tension to the belt. Also for storing excess belt, ie storage loop.
8. *Snub Pulley.* A pulley used to increase the arc of contact of the belt on the drive pulley.
9. *Bend Pulley.* A pulley used to change the direction of the belt.
10. *Head Pulley.* The terminal pulley at the head end of a conveyor. This may be a drive pulley.
11. *Tail Pulley.* The terminal pulley at the tail end of a conveyor. This may be a take-up pulley.
12. *Take-up Pulley.* The travelling pulley used in the take-up device.
13. *Anti Run Back.* An automatic device for preventing a loaded elevating conveyor running backwards when the power source is removed.
14. *Retarder.* A device for preventing the over-speeding of a regenerative conveyor.
15. *Brake.* A device for bringing conveyor to rest and maintaining it at rest.
16. *Cleaner.* A device for removing material which may adhere to the belt or pulley.
17. *Handing of Conveyor.* The sides of a unidirectional belt conveyor are left-hand or right-hand when looking from the tail towards the head of the conveyor.
18. *Tripper.* A device usually comprising two or more pulleys, mounted either in a fixed position or on a travelling carriage, for discharging material from a belt conveyor continuously or at selected points or at any point along the length.
19. *Shuttle Conveyor.* A unidirectional or reversible belt conveyor having over-end discharge, the whole being mounted on a travelling carriage capable of being shuttled backwards and forwards along a track, discharging continuously or at selected points.
20. *Plough.* A blade or blades mounted obliquely across the belt to discharge the material by deflecting it from the belt. *Note*: Can be useful where headroom and space is limited, but is not recommended for any duty other than slow speed conveyors handling non-abrasive free flowing materials.
21. *Chute.* A straight, curved or spiral, open topped or enclosed smooth trough, by which materials are directed and lowered by gravity.
22. *Safeguard.* A guard or device designed to protect persons from danger.



Note: Numbers correspond to those given in the Definitions

FIG 1 TYPICAL TROUGHED BELT CONVEYOR

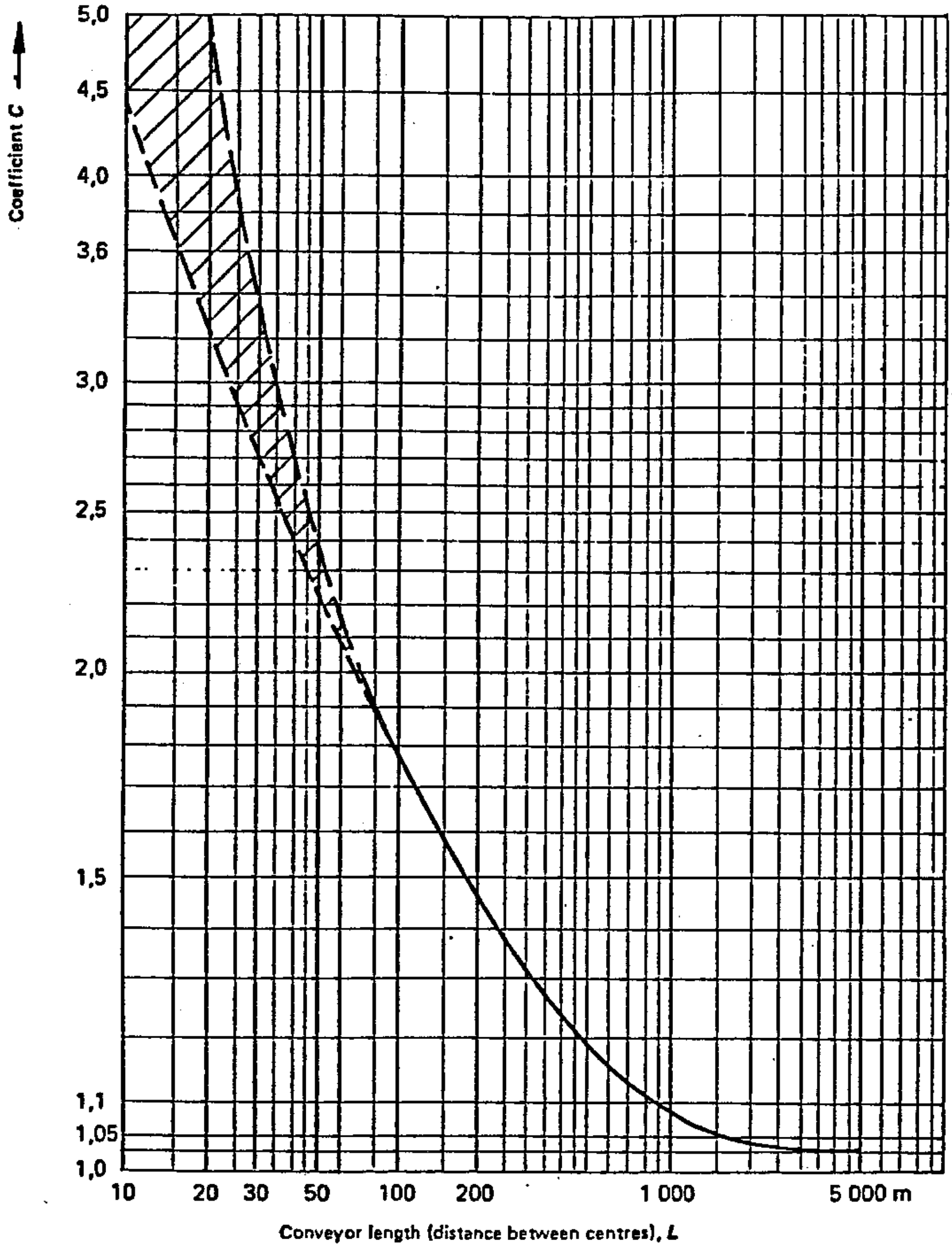


Figure 2 — Coefficient *C* as a function of *L*

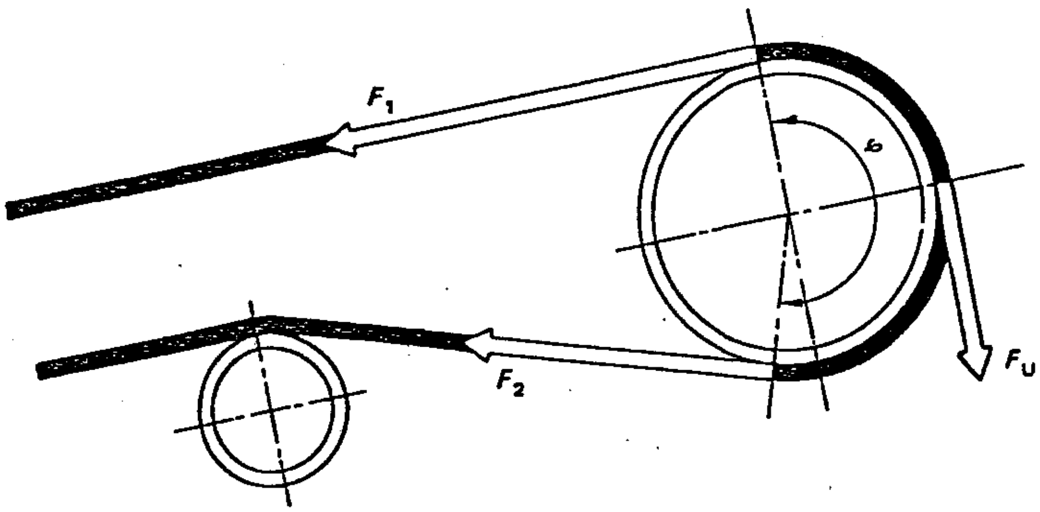


Figure 3 — Tensile forces exerted on the belt

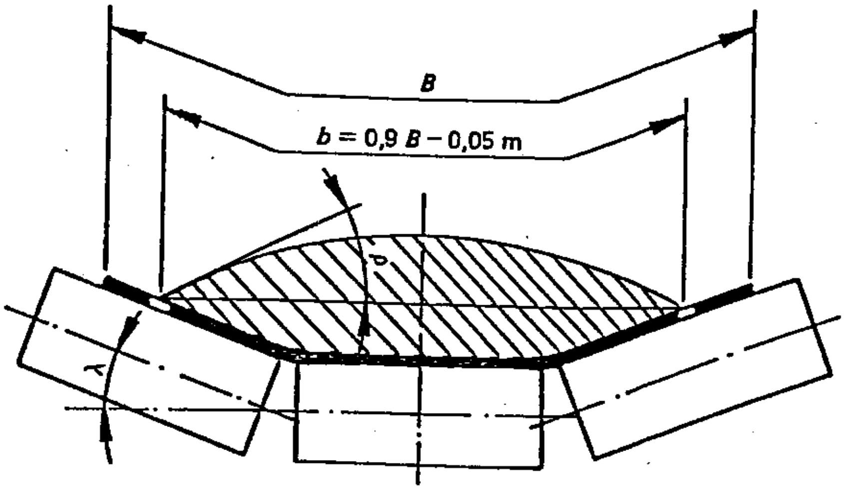


Figure 4 — Trough section with three carrying idlers

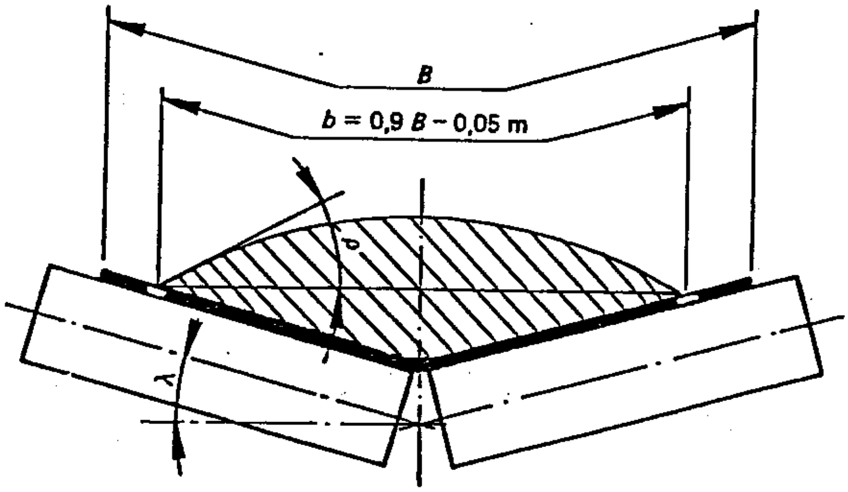


Figure 5 — Trough section with two carrying idlers

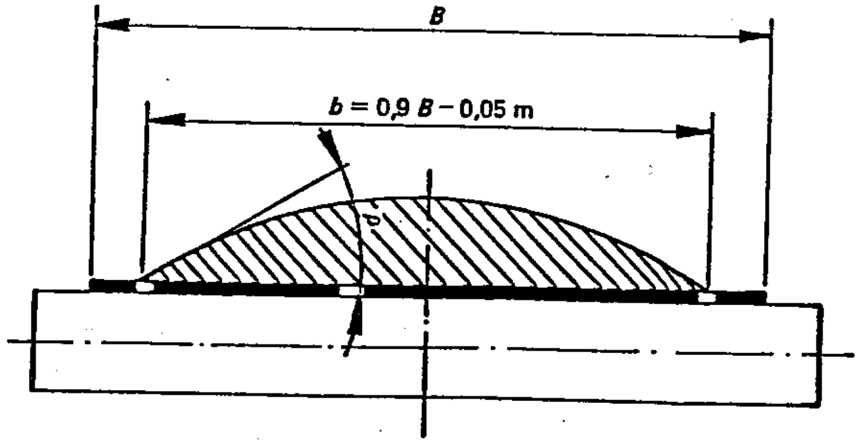
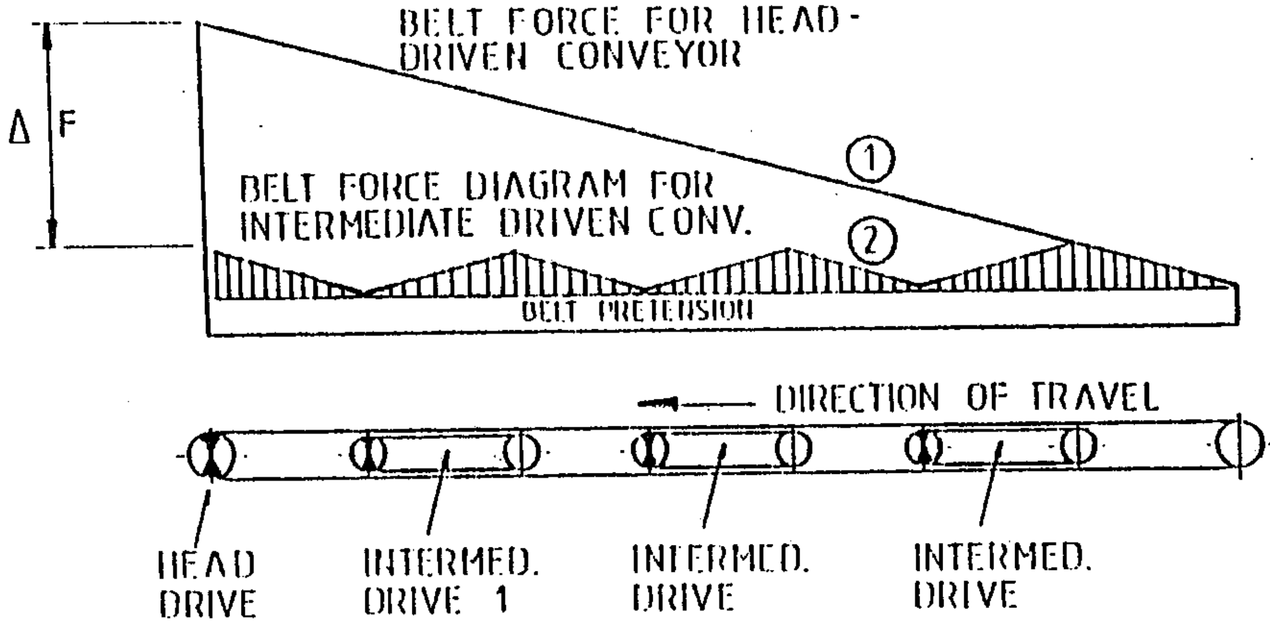


Figure 6 — Section with one carrying idler



COMPARISON OF BELT FORCES FOR HEADDRIVEN (1) VERSUS HEAD AND INTERMEDIATELY DRIVEN (2) CONVEYOR

Fig. 7

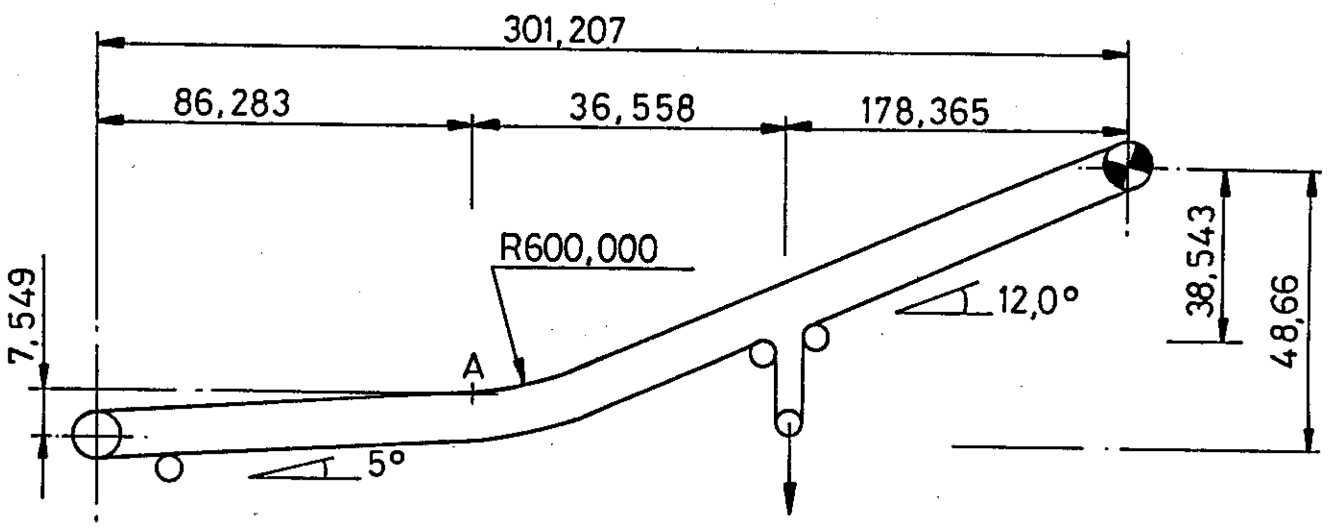


Fig. 8
SILO FEED CONVEYOR

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**********SWF CONVEYOR BELT PROGRAM ************
******ACCORDING TO ISO SPECIFICATION 5048********
    *********VERSION MARCH 1991***********

CONVEYOR BELT NAME                                  =silo feed
CENTRE DISTANCE(m)                                  = 301
LIFT(m)                                             = 49
NEEDED CAPACITY(t/h)                                = 1200
MATERIAL DENSITY(t/cubm)                            = .9
SYSTEM FRICTION FACTOR                              = .02
IDDLER UNIT MASS(kg/m)                              = 28
RETURN SIDE UNIT MASS(kg/m)                         = 24
ADDITIONAL RESISTANCE (kN)                          = 1

BASIC OUTPUT


BELT WIDTH      (mm)                                = 1200
BELT SPEED      (m/s)                               = 3.5
ABSORVED POWER   (kW)                               = 201.357
APPROXIMATE BELT CLASS (kN/m)                       = 800
EFFECTIVE TENSION (kN)                              = 57.53058
APPROXIMATE MOTOR SIZE   (kW)                       = 315
NO LOAD POWER (kW)                                  = 10.74823

********** SAG TENSION ******

IDLERS CENTRE DISTANCE(m)                           = 2.25
SAG TENSION(kN)                                     = 15.4836
TAKE-UP TENSION FOR SAG(kN)                         = 17.81053
HOR DISTANCE BETWEEN TAKE-UP AND FEEDING POINT(m)= 140
VER DISTANCE BETWEEN TAKE-UP AND FEEDING POINT(m)=-10


******** CONCAVE RADIUS *************
CONCAVE RADIUS HORIZONTAL DISTANCE (m)              = 86
CONCAVE RADIUS VERTICAL DISTANCE (m)                = 7.5
CONCAVE RADIUS (m)                                  = 488.4564

********* NUMBER OF PLIES ***********

NUMBER OF PLIES (SINTHETIC BELT)                    = 3

********* PULLEY SHAFTS ************

QUANTITY OF DRIVE PULLEYS                           = 1
PULLEY RADIUS (m)                                   = .375
POWER SHARING FACTOR                                = 100
HIGH TENSION PULLEY WRAP ANGLE                      = 210
INTERMEDIATE TENSION PULLEY WRAP ANGLE              = 180
LOW TENSION PULLEY WRAP ANGLE                       = 180
PRIMARY DRIVE PULLEY SHAFT DIAM(mm)                 = 190.3909
SECONDARY DRIVE PULLEY SHAFT DIAM(mm)               = 0
HIGH TENSION PULLEY SHAFT DIAM(mm)                  = 189.9418
INTERMEDIATE TENSION PULLEY SHAFT DIAM(mm)          = 134.7291
LOW TENSION PULLEY SHAFT DIAM(mm)                   = 134.7291

********** TAKE UP MOVEMENT *****

TAKE-UP                                             =MOVABLE
REQUIRED TAKE-UP MOVEMENT                           = 6.746786
```

*********** TOP COVER THICKNESS INFORMATION*********

CHUTE HEIGHT	= 2
DEAD BOXES	= Ø
SLABBINESS	= .3
MAXIMUM LUMP SIZE	= 3Ø
IMPACT CATEGORY	= 1
TOP COVER THICKNESS (FOR IMPACT)	= 1.845426
TOP COVER THICKNESS (FOR WEAR)	= 2.5

************HOLDBACK FORCE************

INCLINED LENGTH (m)	= 22Ø
HOLDBACK FORCE (kN)	= 41.11185
PULLEY FRICTION FACTOR	= .25
NEEDED HOLDBACK PULLEY WRAP ANGLE	= 15.91829
HORIZONTAL DISTANCE BETWEEN HOLDBACK AND TAKE-UP (m)	= 18Ø
VERTICAL DISTANCE BETWEEN HOLDBACK AND TAKE-UP (m)	=-39

******UTILIZATION PERSENTAGE*********

TROUGH ANGLE (degrees)	= 45
SURCHARGE ANGLE(degrees)	= 2Ø
STEEPEST INCLINE(degrees)	= 12
UTILISATION PERSENTAGE	= 64.78883
MAXIMUM VOLUME CAPACITY (t/h)	= 1852.171

************BELT STRETCH AT FULL LOAD***********

BELT MODULE (kN/m)	= 6ØØØ
BELT STRETCH (m)	= 4.4Ø93Ø8
EXTRA PRETENSION FOR FIXED TAKE-UP (kN)	= 52.73591

************* MOTOR INFORMATION ************

MOTOR SIZE (kW)	= 22Ø
AMPERES (38Ø VOLTS)	= 378.14Ø5
MOTOR QUANTITY	= 1
GEARBOX EFFECTIVITY	= 97

****** DIFFERENT BELT TENSIONS ******

LAY-OUT 1= HEAD DRIVE	
WRAP ANGLE (degrees)	= 21Ø
PULLEY FRICTION FACTOR	= .25
RETURN SIDE TENSION (kN/m)	= 31.961Ø6
MAXIMUM TENSION (kN/m)	= 79.9Ø321
HOR DISTANCE BETWEEN DRIVE AND TAKE-UP(m)	= 18Ø
VERT DISTANCE BETWEEN DRIVE AND TAKE-UP(m)	=-39
TAKE-UP BELT TENSION (kN)	= 38.68466
TAKE-UP FORCE (kN)	= 77.36931
TAKE-UP REEVING FACTOR	= 2

*********BELT INFORMATION**************

BELT CLASS	= 8ØØ
BELT TIPE	=SINTHETIC
BELT MASS	= 17

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**********SWF CONVEYOR BELT PROGRAM ************
******ACCORDING TO ISO SPECIFICATION 5048********
    *********VERSION MARCH 1991***********

CONVEYOR BELT NAME                                    =Silo feed - detail calculatio
CENTRE DISTANCE(m)                                    = 301
LIFT(m)                                               = 49
NEEDED CAPACITY(t/h)                                  = 1200
MATERIAL DENSITY(t/cubm)                              = .9
SYSTEM FRICTION FACTOR                                = .02
IDDLER UNIT MASS(kg/m)                                = 28
RETURN SIDE UNIT MASS(kg/m)                           = 24
ADDITIONAL RESISTANCE (kN)                            = 1

BASIC OUTPUT


BELT WIDTH     (mm)                                   = 1200
BELT SPEED     (m/s)                                  = 3.5
ABSORVED POWER  (kW)                                  = 201.357
APPROXIMATE BELT CLASS (kN/m)                         = 800
EFFECTIVE TENSION (kN)                                = 57.53058
APPROXIMATE MOTOR SIZE  (kW)                          = 315
NO LOAD POWER (kW)                                    = 10.74823

********** SAG TENSION ******

IDLERS CENTRE DISTANCE(m)                             = 2.25
SAG TENSION(kN)                                       = 15.4836
TAKE-UP TENSION FOR SAG(kN)                           = 17.81053
HOR DISTANCE BETWEEN TAKE-UP AND FEEDING POINT(m)= 140
VER DISTANCE BETWEEN TAKE-UP AND FEEDING POINT(m)=-10

************SHORT CONVEYOR DETAILS*************
PULLEY DIAMETER (m)                                   = .65
BELT THICKNESS (m)                                    = .015
NON DRIVING PULLEY SHAFT DIAMETER (m)                 = .125
PULLEY MASS (kg)                                      = 400
SKIRT LENTGH (m)                                      = 5
SKIRT WIDTH (m)                                       = .8
ADDITIONAL RESISTANCE (kN)                            = 1
TAKE-UP TENSION (kN)                                  = 39.09781
BELT TENSION (kN/m)                                   = 81.45436
POWER (kW)                                            = 205.266

************* MOTOR INFORMATION ************

MOTOR SIZE (kW)                                       = 220
AMPERES (380 VOLTS)                                   = 378.1405
MOTOR QUANTITY                                        = 1
GEARBOX EFFECTIVITY                                   = 97

****** DIFFERENT BELT TENSIONS ******

LAY-OUT 9= SHORT CONVEYOR (-60 m)
WRAP ANGLE (degrees)                                  = 210
PULLEY FRICTION FACTOR                                = .25
RETURN SIDE TENSION (kN/m)                            = 32.58151
MAXIMUM TENSION (kN/m)                                = 81.45436
```

SURTEES AND SON (PTY) LTD

CONVEYOR START UP PROGRAM & FALK HOLDBACK SELECTION PROCEDURE

CONVEYOR PARAMETERS — DATE:- 03/04/91

CLIENT:- BELTCON — CONVEYOR NO:- 1.00

NUMBER OF DRIVES:	1
NUMBER OF DRIVE PULLEYS:-	1
LENGTH (L)	301.00 m
HEIGHT FROM TAIL TO HEAD PULLEY	39.00 m
FULL BELT SPEED (Vf)	3.50 m/sec
MATERIAL MASS IN TONS PER HOUR	1200.00 ton/h
BELT MASS (Mg')	17.00 kg/m
TROUGHING IDLER MASS IN Kg. (ENTER 0 IF UNKNOWN)	25.00 kg
TROUGHING IDLER SPACING	2.25 m
CALCULATED TROUGHING IDLER MASS per.. METRE.	11.11 kg/m
RETURN IDLER MASS IN Kg.(ENTER 0 IF UNKNOWN).	31.00 kg
RETURN IDLER SPACING	4.50 m
CALCULAED RETURN IDLER MASS per. METRE.	15.07 kg/m
EST. TOTAL DRIVE/S MASSES ON BELT LINE(0 IF UNKNOWN)	2000 kg
EST. TOTAL PULLEY MASSES ON BELT LINE(0 IF UNKNOWN)	3000 kg

SELECT TYPE OF FLUID COUPLING:-TRACTION "T" USE 180% START FACTOR.
DELAY "TV" USE 150% START FACTOR.
DOUBLE DELAY "TVV" USE 140% START FACTOR.
TURBO SOFT START "TSS" USE 130% START FACTOR.
ACCELERATION CONTROL "TPE" USE 120% START FACTOR.

SELECTED STARTING FACTOR (%)	130 %
DRIVE PULLEY DIA (D1)	650 mm
HEAD PULLEY DIA (D2)	650 mm
PRIMARY OR HEAD DRIVE PULLEY ANGLE OF WRAP (0)	210 (DEG)
SECONDARY DRIVE ANGLE OF WRAP (ENTER 0 IF ONE DRIVE)	0 (DEG)
HEAD PULLEY ANGLE OF WRAP (0)	210 (DEG)
PULLEY FACE	1300 mm
PULLEY BEARING CENTRES	1600 mm
FRICTION FACTOR (U) (OLD=0.35,NEW=0.4)	0.25
BELT WIDTH	1200 mm
TYPE OF BELT USED (STEELCORD=1,FABRIC=2 & COTTON=3)	2
HEIGHT FROM HEAD TO DRIVE PULLEY_(0 IF DRIVE AT HEAD)	0.00 m
HORIZONTAL LENGTH FROM HEAD TO DRIVE(0 IF DRIVE ATHEAD)	0 m
HORIZONTAL LENGTH BEFORE RISE_(ENTER 0 IF INCLINE)	86 m
ENTER INSTALLED MOTOR POWER PER DRIVE.	260 Kw.
START UP TIME REQUIRED	20.00 sec.

CONVEYOR CALCULATIONS

SELECTED LENGTH CO-EFFICIENT	1.6000
SELECTED f VALUE	0.0200
DRIVE PULLEY R.P.M.	92.14 r.p.m
CALCULATED POWER (ABSORBED) @ FULL LOAD:-	
FULL LOAD ABSORBED POWER REQUIRED AT MOTOR/S.	255.78 Kw
FULL LOAD ABSORBED POWER PER DRIVE.	127.89 Kw
FULL LOAD ESTIMATED STARTING TIME FROM BREAKAWAY.	27.58 Sec
CALCULATED EMPTY BELT:-	
EMPTY BELT ABSORBED POWER REQUIRED AT MOTOR/S Pa	40.32 Kw
EMPTY ESTIMATED STARTING TIME FROM BREAKAWAY (t)	10.85 Sec
CALCULATED STOPPING TIME & DISTANCE:-	
ESTIMATED COASTING TIME FOR LOADED BELT:-	8.03 Sec.
ESTIMATED COASTING DISTANCE FOR LOADED BELT:-	19.84 m
H.S.BRAKING TORQUE NEEDED (0 IF NO BRAKE):-	0 Nm.
L.S.TAIL BRAKE TORQUE NEEDED FOR STOPPING TIME:-	0 Nm.

SELECTED TENSIONS:-

EFFECTIVE TENSION (TE)(G24)	40711 N
TENSION T2 (T2 TO ISO) T2=Te/(e^5ø-1)	18510 N
TENSION T2=T2*START FACTOR*eff.	23101 N
TENSION T2,Pins*eff/Vel*K.	32566 N
TENSION T2,(2% SAG AT TAIL)Ts-.15LMb+Mbgh:-	23223 N
SELECTED TENSION T2,MAX ABOVE	32566 N
TENSION (T1) STEADY OPERATION	81277 N
TENSION (T1) STARTING:-	101429 N
TENSION BETWEEN PRIMARY & SECONDARY DRIVES:-	0 N
TENSION AT TAIL:-	33292 N
LIMITATION DUE TO BELT SAG (MAX 2% SAG.):-	
MINIMUM TENSILE FORCE NOT EXCEED T2 (ON CARRYING SIDE):-	21772 N
SELECTED BELTING TYPE:-	FABRIC
SELECTED MINIMUM BELT TENSION (STEADY OPERATION)	451.54 KN/m
SELECTED MINIMUM BELT TENSION (STARTING)	467.70 KN/m
SELECTED BELT CLASS	500 Class

DYNAMICS TEST FOR IDEAL STARTING & STOPPING:-

MIN. TORQUE RAMP UP OR STOPPING TIME (AT T.REL=10):-	1.30 Sec.
TORQUE RAMP PLUS STARTING TIME FROM BREAKAWAY, LOADED:-	28.88 Sec.
MINIMUM TIME DELAY BETWEEN DRIVES,(CONSULT SURTEES):-	0.65 Sec.

SELECTED EQUIPMENT SCHEDULE:-

HIGH SPEED FLUID COUPLING SELECTED:-

SELECTED VOITH H.S.FLUID COUP. SIZE.(ON INSTALLED POWER)	562
SELECTED VOITH FLUID COUPLING TYPE:-	TSS
VOITH COUPLING MASS ON GEARBOX SHAFT(WITH MAX OIL FILL):	147.00 Kg
FALK HOLDBACK IF NECESSARY:-	REQUIRED
POSITION OF FALK HOLDBACK:-	1 OF ON DRIVE

LOW SPEED COUPLING SELECTION:-

1. MAINA DOUBLE ENGAGEMENT GEAR COUPLING FOR FOOT MOUNTED GEARBOXES:-

SIZE SELECTED ON ABSORBED TORQUE:-	GD-4A
CATALOGUE TORQUE RATING:-	12900 Nm.
PEAK TORQUE RATING:-	32250 Nm.
MAXIMUM BORE CAPACITY:-	112 mm
SIZE SELECTED ON DRIVE PULLEY SHAFT DIA.:-	GD-4A
CATALOGUE TORQUE RATING:-	12900 Nm.
PEAK TORQUE RATING:-	32250 Nm.
MAXIMUM BORE CAPACITY:-	112 mm

2. SURLOCK TYPE 30 RIGID FLANGE COUPLINGS FOR SHAFT MOUNTED GEARBOXES:

SIZE SELECTED ON ABSORBED TORQUE:-	GAS 400/115
CATALOGUE TORQUE RATING:-	35200 Nm.
MAXIMUM BORE CAPACITY:-	115 mm
SIZE SELECTED ON DRIVE PULLEY SHAFT DIA.:-	GAS 400/115
CATALOGUE TORQUE RATING:-	35200 Nm.
MAXIMUM BORE CAPACITY:-	115 mm

BRAKE SELECTION:-

HIGH SPEED DRUM BRAKE:-

1. SELECTED ON ENTERED BRAKE TORQUE ABOVE PER DRIVE:-	NO BRAKE
DRUM DIAMETER:-	NO BRAKE mm
RATED TORQUE:-	NO BRAKE Nm.
2. SELECTED ON ABSORBED MOTOR TORQUE PER DRIVE:-	NO BRAKE
DRUM DIAMETER:-	NO BRAKE mm
RATED TORQUE:-	NO BRAKE Nm.

PAGE 3

HOLDBACK SELECTION STEPS :-

STEP 1:- SELECT HOLDBACK ON TOTAL SYSTEM ABS, POWER * START FACTOR,
WHEN THE HOLDBACK IS FITTED TO HEAD PULLEY:-
CALCULATED..,Ta*eff*$*No.drives. 32.42 KNm

STEP 2:- SELECT HOLDBACK ON TOTAL SYSTEM INSTALLED TORQUE,
WHEN THE HOLDBACK IS FITTED TO HEAD PULLEY:-
CALCULATED..,Ti*eff*No.drives. 46.64 KNm

STEP 3:- SELECTED ON RUNBACK TORQUE :-
CALCULATED..,mgh - IDLER RESISTANCE FORCE ON INCLINE:- 2.91 KNm

STEP 4:- SELECTION ON MAX. TRANSMITTABLE TORQUE DUE TO PULLEY WRAP :-
WHEN HOLDBACK IS FITTED TO HEAD PULLEY:-
TENSION T2..,SELECTED T2 + BELT MASS HEAD TO DRIVE (mgh 32.71 KN
CALCULATED..,T2(e^5ø-1):- 20.50 KNm
WHEN HOLDBACK IS FITTED TO SINGLE DRIVE PULLEY:-
CALCULATED..,T2(e^5ø-1):- 33.40 KNm
WHEN HOLDBACK IS FITTED TO PRIMARY & SECONDARY DRIVE PULLEY:-
CALCULATED ON PRIMARY...,T12(e^uθ-1):- 0.00 KNm
CALCULATED ON SECONDARY...,T2(e^uθ-1):- 0.00 KNm

CALCULATED USING ESCOM SPEC.(NWS 1556):- 12.61 KNm
CALCULATED USING ANGLO AMERICAN SPEC.(T2*K*Dd/2000):- 43.88 KNm

HOLDBACK SELECTION ON TORQUE USING ONLY ONE, IS:-
SELECTION @ ABSORBED L.S.TORQUE * eff.(CATALOGUE TORQUE 24.94 KNm
CHECK MAXIMUM SYSTEM TORQUE, STEP 1 & 2 (PEAK TORQUE)== 32.42 KNm

FINAL FALK HOLDBACK SELECTION, USE SIZE:- 1095 NRT
CHECK FALK HOLDBACK "CATALOGUE TORQUE" RATING, WHICH IS 37.960 KNm
CHECK FALK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:- 56.952 KNm
CHECK FALK HOLDBACK MAXIMUM BORE CAPACITY:- 145 mm

HOLDBACK SELECTION ON TORQUE FOR PRIMARY & SECONDARY DRIVES:-
SELECT EACH HOLDBACK ON 60% OF ABSORBED TORQUE========= 14.96 KNm

FINAL FALK HOLDBACK SELECTION, USE TOW OFF SIZE:- 1085 NRT
CHECK FALK HOLDBACK "CATALOGUE TORQUE" RATING, WHICH IS 21.696 KNm
CHECK FALK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:- 32.544 KNm
CHECK FALK HOLDBACK MAXIMUM BORE CAPACITY:- 130 mm

PAGE 4

STEP 5:- MINIMUM PULLEY SHAFT DIAMETER:-

MATERIAL(ENTER 1 FOR EN3,2 # EN8,3 # EN9 & 4 # EN19)___ 2
MAXIMUM WORKING SHEAR STRESS FOR FATIGUE 70 MPa.
MAXIMUM WORKING COMBINED STRESS AT BEARING 100 MPa.
MAXIMUM WORKING SHEAR STRESS STATIC CONDITIONS 120 MPa.

CALCULATED BEARING CENTER TO HUB DISTANCE (a) 250 mm
PULLEY HUB SPACING ..FACE - 100 1900 mm
MAXIMUM ALLOWABLE DEFLECTION eg.1/2500(RINGFEDER 7012)__ 1850
NORMAL RUNNING TOTAL SYSTEM TORQUE.(PULLEY POWER):- 24.94 KNm
RUNNING TORQUE ON EACH DRIVE PULLEY SHAFT.:- 12.47 KNm
MAX. BENDING MOMENT ON DRIVE PULLEY SHAFT.:- 12.22 KNm
MAX. BENDING MOMENT ON HEAD PULLEY SHAFT.:- 20.32 KNm
NETT TENSION ON DRIVE PULLEY.(P2=T1+T2) :- 98 KN
NETT TENSION ON HEAD PULLEY.(P2=T1+T1) :- 163 KN

CALCULATE DRIVE PULLEY SHAFT(IF DRIVE AT HEAD USE THESE DIA'S.):-
CALC. DRIVE SHAFT DIA. BASED ON TORSION:- 97 mm
CALC. BEARING SHAFT DIA. COMBINED TORSION.(Te):- 127 mm
CALC. BEARING SHAFT DIA. COMBINED BENDING (Me):- 127 mm
CALC. HUB SHAFT DIA. BASED ON DEFLECTION (5 min.):- 166 mm
CALC .HUB SHAFT DIA. DEFLECTION (ESCOM 1:2500):- 160 mm
SELECTED HUB SHAFT DIA.(STD. RINGFEDER SIZE)============ 170 mm
SELECTED BEARING SHAFT DIA.============================= 145 mm
SELECT MINIMUM DRIVE OR HOLDBACK SHAFT DIA.============= 100 mm

CALCULATE HEAD PULLEY SHAFT:-
CALC. HOLDBACK SHAFT DIA. BASED ON TORSION :- 122 mm
CALC. BEARING SHAFT DIA. COMBINED TORSION.(Te):- 139 mm
CALC. BEARING SHAFT DIA. COMBINED BENDING (Me):- 139 mm
CALC. HEAD SHAFT DIA. BASED ON DEFLECTION (5 min.):- 188 mm
CALC.HEAD SHAFT DIA. DEFLECTION (ESCOM 1:2500):- 181 mm
SELECTED HUB SHAFT DIA.(STD. RINGFEDER SIZE)============ 190 mm
SELECTED BEARING SHAFT DIA.============================= 145 mm
SELECT MINIMUM HOLDBACK SHAFT DIA.====================== 125 mm

MINIMUM HOLDBACK SHAFT DIA. BASED ON PURE STATIC SHEAR STRESS
@ Ta.total*$==== 111 mm
HOLDBACK SELECTION ON MAX. BORE IS:-

SELECTED HOLDBACK TO FIT ON HEAD PULLEY, SHAFT DIA.===== 125 mm

FINAL FALK HOLDBACK SELECTION, USE SIZE:- 1085 NRT
CHECK FALK HOLDBACK "CATALOGUE TORQUE" RATING IS:- 21.696 KNm
CHECK FALK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:- 32.544 KNm
CHECK FALK HOLDBACK MAXIMUM BORE CAPACITY:- 130 mm

SELECTED HOLDBACK TO FIT ON DRIVE PULLEY, SHAFT DIA.==== 100 mm

FINAL FALK HOLDBACK SELECTION, USE SIZE:- 1075 NRT
CHECK FALK HOLDBACK "CATALOGUE TORQUE" RATING IS:- 13.560 KNm
CHECK FALK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:- 20.340 KNm
CHECK FALK HOLDBACK MAXIMUM BORE CAPACITY:- 100 mm

BELTCON 6

WORKSHOP

SESSION 3 :

NUTS AND BOLTS

By G. G. Shortt
Anglo American Corporation of S.A. Ltd
Johannesburg September 1991

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FIGURES

TABLES

BELTCON 6 WORKSHOP

SESSION 3

THE NUTS AND BOLTS OF THE DESIGN

1. INTRODUCTION

We have seen that the sizing of the belt is dependent on various factors, and have seen how to apply those factors to aid us in the selection of belt sizes. We moved on to finding the power of the conveyor system, as well as how to establish the tension at the drive pulley. The question often asked by the ordinary conveyor designer is; "I can see how that is done, but how do I apply it to my design; what can I do with all this information?"

It is true that many conveyor designers (and potential conveyor designers) are discouraged by long, complicated, heavily mathematical dissertations on aspects of conveyor design. These masterpieces, while very necessary in the field of understanding the conveyor, often result in the ordinary designer feeling very inadequate. However, the detailed analyses are often an essential path to the validation, and even compilation, of the "simple" methods.

What I will try to do today, is to show simply, how we can apply some of the concepts we have seen and heard to the design of our conveyors.

Aspects of conveyor design that we will consider are

(1) Conveyor idlers - Selecting the idler pitch.
(2) Vertical curves - The basic design criteria for establishing and laying out the optimum curve.
(3) Holdbacks - A look at the various means avaiable for the selection of the holdback.
(4) Belt flap - What it is and a simple method to avoid it.

2. IDLERS

Probably the most heartache for the operating personnel is caused by the idlers on a conveyor. Or rather, the incorrect selection of idlers themselves, as well as the pitching of the idlers on both the carrying and return strands. We have all seen idlers that are jammed solid, with the belt scouring across the top of it. We have also seen many "daylight" idlers, those with sections of the shell worn away, merrily rotating, singing as they go, flashing daylight through them as they rotate, chewing great chunks out of the conveyor cover. Alternatively, we have seen idlers with the bearing housing collapsed, and the shell rotating around what is left of the shaft. Many of us will have seen idlers with one end in the bracket, and the other end lying in the spillage around it.

All this sounds very negative, but these incidents must be seen as lessons for the designers, constructors and operators of conveyors. We must learn from what we see on site, not simply blame "those useless operators", or "those high-and-mighty design engineers", depending on which side of the fence you happen to find yourself. At the Anglo American Corporation, we find ourselves on both sides of the fence, since we are not only designers of the systems operating, we are also part of the team that operates the systems.

Let us look at a typical example of the selection of the idler pitch, both for the carrying and return strands. We will look at the determination of the load carried by the rolls, as well as the pitfalls that we can identify, and ways to avoid them. We will look at the determination of a dynamic load factor, how to apply it and we will look at those idlers that have special loads, and how to determine those loads.

2.1 Determination of the Load Carried by the Idler.

This is both an easy and complicated task, depending on your situation, or on the accuracy that you wish to achieve.

The load carried by the most heavily loaded roll in an idler set is given by the following :

$$W_a = \left\{ g \left[\{B + (n \cdot Z \cdot f_1 \cdot f_2)\}/n \right] \cdot 10^{-3} \right\} \text{ kN/m} \quad \ldots\ldots\ldots\ldots (2.1)$$

The symbols have the following meaning :

W_a = Actual load carried by the most heavily loaded roll. kN/m
g = Gravitation constant 9,81 m/s².
B = Mass of the belting. kg/m
n = Number of rolls in the idler set.
Z = Material load. kg/m
f_1 = Dynamic load factor.
f_2 = Burden factor.

The material load may be found from the well-known formula:

$$Z = \frac{Cdc}{3{,}6 \cdot S} \quad \text{kg/m} \quad \ldots\ldots\ldots\ldots (2.2)$$

Where : Cdc = Design capacity t/h
S = Belt speed m/s

Determination of the Dynamic Load Factor

$$f_1 = \left[\{(Ca) \cdot S^2\} + 1 \right] \quad \ldots\ldots\ldots\ldots (2.3)$$

The lump size factor **Ca** is dependent on the idler form and the material lump size, and is therefore somewhat empirical. The factor is shown in table 1.

TABLE 1	LUMP SIZE FACTOR Ca	
LUMP SIZE	IDLER FORM	
RANGE (mm)	FIXED	GARLAND
-5 +0	0	0
-25 +5	0,005	0
-100 +0	0,009	0,005
-100 +50	0,014	0,009
+100	0,050	0,020

<u>Determination of the Burden Factor and Gauge Length</u>

TABLE 2	BURDEN FACTOR & GAUGE LENGTH	
IDLER TYPE	f_2	GAUGE LENGTH (mm)
3-ROLL	0,66	0,350W + 40
5-ROLL	0,47	0,213W + 25
2-ROLL VEE	0,60	0,544W - 5
FLAT	1,10	1,017W + 87,14
PICKING (*)	1,00	1,017(W -300) + 87,14

* The picking idler has the form as proposed by AAC as a dimensional standard. See paragraph 2.8 and Figure 8.

The gauge lengths shown are for idler rolls conforming to the boundary dimensions as given in SABS 1313.
The factor for 2-roll vee idlers is to cater for the additional load on the inboard bearings, while the value for flat idlers caters for an amount of belt wander.
The variable W in table 2 indicates the belt width.

We must not be misled by this burden factor. The value given above shows the burden factor for a belt loaded to 100% of its capacity. Note also that 100% capacity is based on the area of cross-section for the belt with the load edge on the freeboard line as shown in figure 1 below.

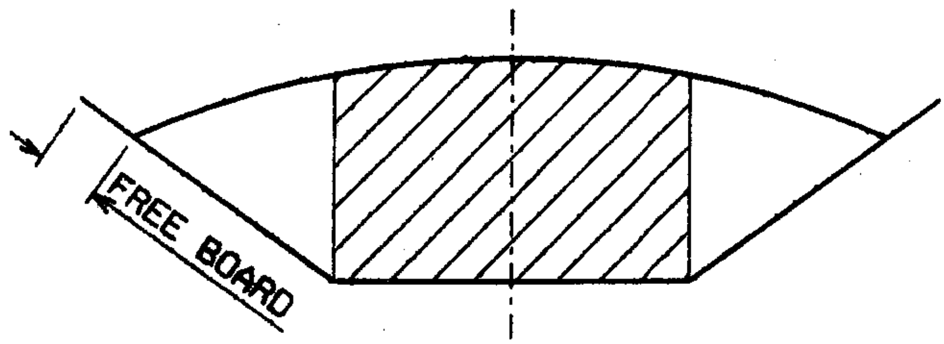


FIGURE 1
BELT LOAD PROFILE

If the belt has a load less than 100%, the load on the centre roll (usually the most heavily loaded) will rise, to a value where the percentage of the burden carried on the centre roll reaches 100%. This is true for 3-roll and 5-roll idlers only, of course. The 2-roll vee form idler will always have a burden factor of 0,60; and the flat forms will have the factor 1,1 for reasons given above.

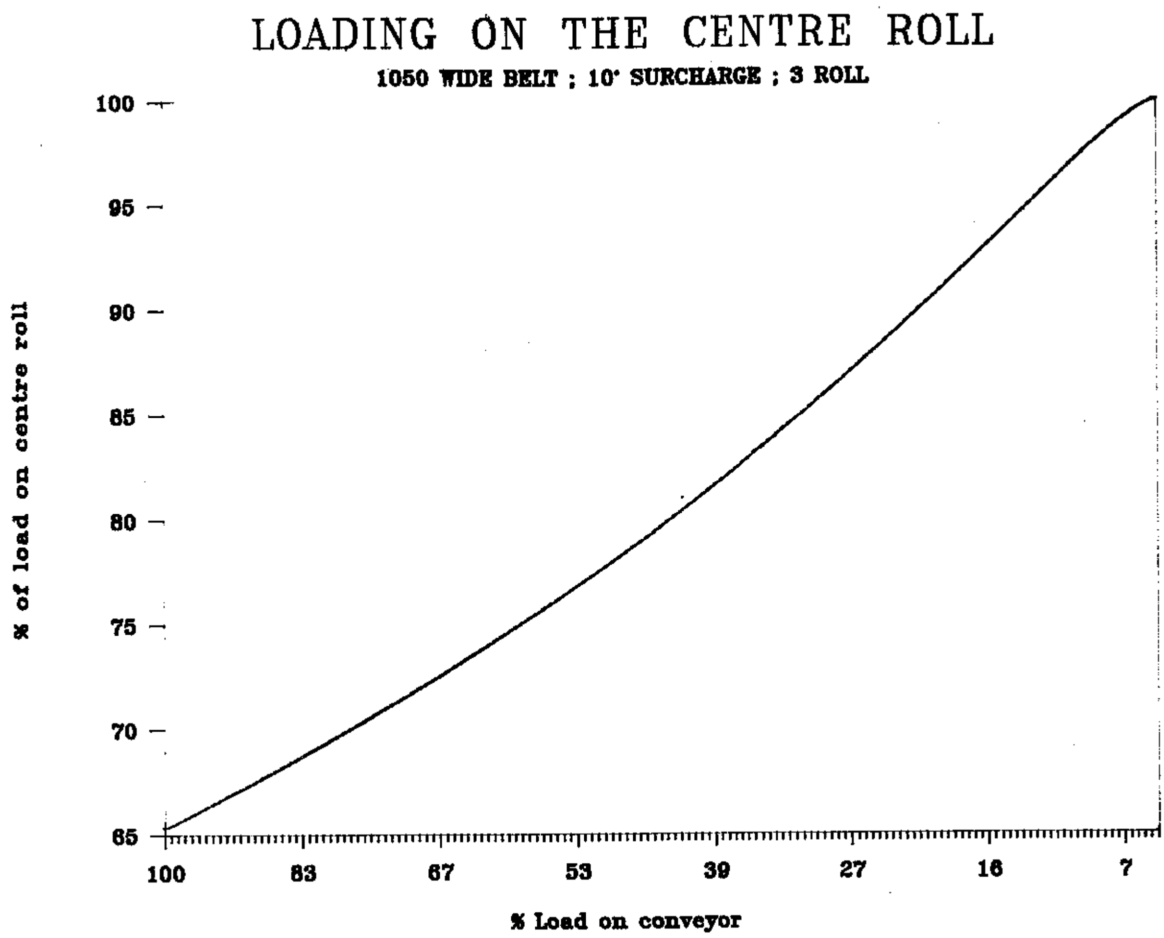
The burden factor variation is shown on the graph , figure 2, below and is indicative of idlers conforming to SABS 1313 as noted above.

2.2 Maximum Idler Pitch

The idler pitch is influenced by the factors given above and by the specification of the idler bearing life, as well as the allowable slope of deflection of the shaft through the bearing.

Pitch limited by Bearing Life

Much has been written on the subject of the idler life limited by the bearing performance. Major bearing manufacturers have long used the L_{10h} life calculation, which is a prediction of the bearing endurance limit. Recently, SKF introduced the new life formula which caters for wear and environmental factors, rather than a straight bearing endurance limit. This system has its own limitations, however, since not all the factors for the bearings used in idlers nowadays are covered.



The values on the X axis are a direct import from a computer listing and are indicative only

FIGURE 2

GRAPH SHOWING THE RELATIONSHIP BETWEEN BELT LOAD AND CENTRE ROLL LOADING - 3-ROLL IDLERS

Allowable Load

The allowable load to satisfy the specification of the life of the idler roll bearings can be summarised as follows :

$$W_L = 2 \cdot \left[\frac{\pi \cdot (Di) \cdot C^p}{3,6 \cdot H \cdot S} \right]^{1/p} \qquad \text{.............................(2.4)}$$

Here, W_L = Allowable load kN
Di = Idler roll diameter. mm
C = Bearing load rating (from supplier's tables) kN.
p = 3 for ball bearings.
= (10/3) for roller bearings.
H = Specified life in hours.
A common value used today is 75 000 hours.

How Long is a Year ?

If we assume a year of 340 days with a double 8-hour shift per day, a utilisation of 85% and an availability factor of 80%, we can expect the conveyor to work for 3700 hours per year. Using the value of 75 000 hours, we can then expect to replace our idlers in (0,9·75000)/3700 = 18 years!

We know that this kind of life is simply not achieved, because of environmental conditions, bearing seal design, high pressure hosing down as macro-cleaning, overloading, spillage and so on. A more realistic value would be to quote a life expectancy of around 35 000 to 40 000 hours, to cater for the bearing endurance limit, and to make allowances for the factors enumerated above. For conveyors running at speeds greater than 5 m/s, the life endurance limit specification could be increased to 50 000 hours.

Increasing Production in the Future

We must be aware of the intention to increase production in the future, usually by increasing the belt speed. If the factors are not properly considered at the design stage, we could end up with a large pile of junked idler rolls. Communication with the user is imperative here.

Idler Pitch Based on Endurance Limit

The idler pitch is then given by $P = \frac{W_L}{W_m}$ m ....(2.5)

Some extracts of bearing data are shown in Table 3.

TABLE 3	BEARING DATA			
BEARING TYPE	C (kN)			δ (rad)
	Ser 20	Ser 25	Ser 30	All series
Seize resistant cage	10,400	12,700	22,800	0,00407
Deep groove ball	9,360	14,000	19,500	0,00174
Spherical roller	30,500	35,700	48,900	0,02617
Taper roller	22,900	25,500	33,600	0,00058

The value δ indicates the allowable slope of deflection of the idler shaft through the bearing. The idler series is as defined in SABS 1313.

Pitch Limited by the Slope of Deflection

Here we make use of standard beam theory and well known, tried and tested strength of materials theory.
The slope of deflection of a beam simply supported at the ends and having two equal loads equispaced along its length is given by :

$$\delta = \left[\frac{R_{w} \cdot A \cdot (L-2A)}{4 \cdot E \cdot I}\right] \text{ radians} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.6)$$

Where R_{w} = Total load on the beam kN
L = Length between supports mm
A = Distance from the support to the load at each end. mm
This is the distance from the bearing centre to the edge of the idler broached flat.
E = 210 000 N/mm^2
I = $[(\pi/4) \cdot d^4]$ mm^4 (d = idler shaft diameter mm)

$$\text{Thus} \quad W_a = \left[\frac{4 \cdot E \cdot I \cdot \delta}{A(L-2A)}\right] \text{ kN} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.7)$$

Here W_a = allowable load to limit the deflection.

Values for the distance from the support to the load have been collected and averaged for a number of major idler suppliers. The average value for A is then 45 mm. Of course, if your project has a specified idler supplier, this value can be obtained from them.

Allowance for Manufacturing Tolerances

All mechanical items must lie within certain tolerances. In order to allow the idler manufacturer some leeway, we allow an assembly tolerance of 2,0 millirad (0,002 radians). Putting all this together, we come up with the following expression :

$$W_a = \frac{4 \cdot 210000 \cdot \pi(\delta - 0,002) \cdot d^4}{64 \cdot 45(L - 90) \cdot 1000} \quad \text{kN}$$

$$= \frac{41,233(\delta - 0,002) \cdot d^4}{45(L-90)} \quad \text{kN}$$

Rounding off the decimals, we get

$$W_a = 0,92 \left[\frac{(\delta - 0,002) \cdot d^4}{(L-90)} \right] \quad \text{kN} \quad \ldots\ldots\ldots(2.8)$$

Idler Pitch Based on Limiting deflection

The idler pitch is then given by $P = \dfrac{W_a}{W_m}$ m ....(2.9)

2.3 Pitching Idlers in Convex Curves

There are numerous examples of conveyors which operate beautifully everywhere, except at the convex vertical curve. Here the operating personnel report rapid and repeated idler failure,on both the carrying and return stands, with spillage the order of the day. The most common cause of failure of idlers in these situations is the increase in idler loading as a result of the forced vertical misalignment of the idler with respect to the belt line, forcing the belt into the convex curve.

The additional load on the centre roll of a three or five roll set is given by the expression :

$$W_c = \frac{T_c}{R} \quad \text{kN/m} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2.10)$$

Here T_c = Belt tension (kN) at the curve in question
R = Radius of the curve m

Then the actual loading on the roll is given by

$$W_x = (W_m + W_c) \quad \text{kN/m} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2.11)$$

The idler pitch is then given by

$$P = \frac{W_L}{W_x} \text{ m} \quad \text{(life)} \qquad \ldots\ldots\ldots(2.12)$$

or by
$$P = \frac{W_d}{W_x} \text{ m} \quad \text{(deflection)} \qquad \ldots\ldots\ldots(2.13)$$

Choosing the Correct Pitch

In every case, the lesser dimension is considered to be the maximum value for the idler pitch.

2.4 Return Idlers

In the case of return idlers, the idler pitch can be established by using the same procedure as noted above. Of course, in this case the material load and the burden factor is generally zero (unless we have a double-deck system).

WARNING

A word of warning. There are some idler manufacturers who claim extended idler pitch capability, based on the design of their idler, its load capabilities, their seal design, or some other magical arrangement. While this sounds great and is probably mathematically defensible, we must be aware that the pitch of the idler will also establish the minimum tension in the conveyor system. Overspecification of the idler pitch could result in increased belt standing tensions, which in turn lead to increased belt class and therefore cost. This is an excellent example of the interaction of the component parts of a conveyor. So be careful!

In Anglo American, we tend to specify carrying idler pitches of 1,0 m, 1,2 m, and 1,35 m, depending on the stringer length. We like to pitch the idlers on the stringer such that the pitch is halved at each end. As for return idlers, these are generally pitched at three to four times, and very occasionally six times the carrying idler pitch. See figure 3.

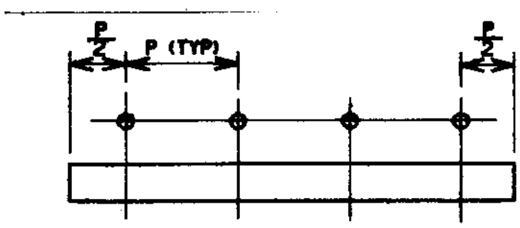


FIGURE 3
LOCATION OF IDLERS AT THE END OF THE STRINGER

Limiting Sag

A useful rule of thumb is to limit the return idler sag to one half of the idler roll diameter. This has the effect of reducing the approach and depart angles of the belt onto the idler rolls. This in turn minimises the effects of minor misalignments, reduces return strand drag and generally improves the "look" of your conveyor. It was fashionable at one time (and probably still is in some places) to specify a return idler sag not exceeding 3% of idler pitch. With the advent of more predictable idler performance and subsequently longer return idler pitches, the return belt starts to resemble a festoon!

2.5 Dynamic Behaviour

A further restriction on the return idler pitch is the dynamic behaviour of the belt, but more about that in section 5.

2.6 Dimensional Standards

South Africa was probably the first country to establish and operate a **national** standard for the dimensions of belt conveyor idlers and rolls. This took the form of SABS 1313, currently under review, and took about 2 years to motivate. There have been some omissions though, probably for very good reason, and also possibly because the idler forms omitted were simply not considered at the time. These omissions make a conveyor designers life difficult, since the specification of garland, fixed form suspended and picking idlers can lead to some interesting scenarios.

2.7 Garland Idlers

We have had the experience where the installation of an inclined underground conveyor with a lift of +100 m went without a hitch. Then someone discovered that the idlers (5-roll garlands) were ordered from two different suppliers. However, the installation complete, and the belt being pulled in, it was discovered that the centre roll of maker A's idler was 200 mm lower than that of maker B's idler. The idler support structure was designed around one of the manufacturer's idlers and the other simply did not fit. As an aside, to solve the problem, we removed the centre roll of the "lower" set, to form a 4 - roll garland, which brought the belt lines to more-or-less coincidence. Not an entirely satisfactory situation, but a relatively elegant solution to a sticky problem.

Anglo American has issued a specification for garland idlers, (Figures 6 and 7) which specifies the boundary dimensions for the rolls and the idler set form. This was done to prevent a repetition of the scenario outlined above. Possibly a national dimensional standard could arise from this.

2.8 Picking Idlers

The same sort of confusion is prevalent with regard to picking idlers, with all sorts of dimensions and different profiles appearing. We are proposing a single dimensional standard, with individual performance standards stipulated by the project designers and consultants. (Figure 8)

2.9 Fixed Form Suspended Idlers

Anglo American has not specified a great deal of this type of idler. However, it is plain, from the various suppliers, that there is a need for some sort of dimensional standard. This will no doubt be addressed by the SABS or the CMA at some stage in the future.

* * * * * * *

3. VERTICAL CURVES - CONCAVE AND CONVEX

The subject of vertical curves has long been a source of irritation to operating personnel. In the case of concave curves, we often find the belt rising out of the idler bed, being totally unsupported for long lengths, with the attendant spillage and belt training problems. We see whole structures designed with the lift-out in mind, and various measures designed into the system to limit the lift-out of the conveyor.

Trippers

One of the worst offenders in the field of belt lift is the travelling tripper. We are all familiar with this machine. Usually designed with about 3,0m to 4,0m lift (sometimes considerably more), with the approach stringers at an optimistic 12°, or even a blistering 15°!. Now, let us consider a (fairly large) tripper with the lift of 4,0 m and a stringer approach angle of 12°. This gives us a length (on the slope) of

$$\begin{aligned} L &= 4/\sin(12°) \\ &= 19{,}239 \text{ m} \end{aligned}$$

Now, if we say that the tangent point of the curve is at the head pulley, (the most likely place for the upper tangent point to be, on a tripper) then the maximum radius that the curve can take will be given by

$$\begin{aligned} R &= L/\tan(12/2) \\ &= 183 \text{ m} \; ..... \; \text{not a very large value.} \end{aligned}$$

If we consider that the tripper is usually at the head of the conveyor, where the tensions are likely to be the greatest, then the curve determined above starts to look less and less attractive. Furthermore, the belt is **tangent** to the 12° line at the head pulley. This implies that the belt leaves the stringer immediately behind the head pulley, and that the belt is essentially unsupported from that point. With the higher tensions to be expected at the head end of the conveyor and, with the belt unsupported, we can expect to see the belt actually invert, under certain conditions, which creates a wonderful load-shedding area.

From this, we can see that the design of items such as travelling trippers needs much more careful consideration than has been the general practice up to now. I personally do not like trippers, and will always try to design some other form of materials handling system to avoid the installation of one. Systems such as back-to-back shuttles, shuttle chutes and so on are far preferable than the plain tripper, in most cases. The additional capital that the alternative systems may require is easily offset against the additional and continued attention that the travelling tripper requires.

3.1 Determining the Belt Radius - Concave

We normally analyse the curve for eleven conditions as shown in table 4 below.

TABLE 4	**CONDITIONS OF CURVE ANALYSIS**
CONDITION	DESCRIPTION
1	Loaded to lower tangent point - starting
2	Loaded to lower tangent point - running
3	Loaded to lower tangent point - stopped
4	Fully loaded at design capacity - starting
5	Fully loaded at design capacity - running
6	Fully loaded at design capacity - stopped
7	Empty belt - starting
8	Empty belt - running
9	Empty belt - stopped
10	Limiting centre tensions
11	Ensuring positive edge tension

For conditions 1 to 9, the radius of the curve may be found from the general formula

$$R = \frac{T_x \cdot 10^3 \cdot f_c \cdot f_b \cdot f_m \cdot f_s}{g \cdot B} \quad \mathrm{m} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (3.1)$$

The symbols are defined as follows :

- R = Radius of the curve m
- T_x = Tension at the lower tangent point for the specified condition of analysis. kN
- B = Belt mass kg/m
- g = Gravity constant = 9,81 m/s²

- f_c = Starting factor

  This is dependent on the type of starting control on the conveyor system.

  (i) For fluid couplings, the value of the factor lies between 1,8 and 1,2 depending on the type of coupling. The turbo type of coupling has a value generally around 1,3 to 1,25. The ordinary fluid traction coupling can have values as high as 1,8, depending on the degree of fill. An overfilled coupling will give higher values, while a coupling with too little oil will give very low values - but your conveyor won't start, either! For differential flow and scoop control type couplings, the value can be as low as 1,2.

  (ii) For electronic soft starts, the value is usually accepted as between 1,3 and 1,4 but lower values have been successful.

  (iii) For direct-on-line type couplings the value will be not less than 2,5. Direct-on-line couplings are not preferred where the belt has a concave vertical curve, for this very reason. The radius calculated will be much larger than would otherwise be required, because of the high starting factor.

- f_b = Curve "bedding" factor. This is effectively a factor of safety. It is simply a curve oversizing factor to ensure that the belt will lie in the curve. We normally assign the value of 1,1 to this factor, for normal curves. If the tension in the system is subject to unpredictable fluctuations, the factor can be as high as 1,2. For belts with unsupported lengths, such as the approach to a travelling tripper the factor should never be greater than 1,0.

- f_m = Mass loss factor. This is utilised to cater for the loss of cover and therefore loss of belt mass due to wear. The factor is usually taken as 1,1. Lower values can be taken, if the belt is designed for a short life, or if the curve is sized for geometric reasons.

- f_s = This is a "slope" factor, and recognizes the catenary function of the curve. It is really another "safety" factor. the factor is derived from

$$f_s = 1/\cos^2(\alpha) \quad \text{.......................(3.2)}$$

where α = curve subtended angle.

This all analyses the curve for the first 9 conditions enumerated above.

Condition 10

Here we want to limit the tension at the centre of the belt to a safe value. The radius to achieve this is given by

$$R = \frac{W \cdot (E \cdot W \cdot \sin \beta)}{f_d \cdot T_x \cdot 10^6} \quad m \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.3)$$

here W = belt width (mm)
β = Idler wing roll angle (°)
E = Belt modulus (kN/m)
f_d = Belt factor
= 4,5 for fabric belts
= 12 for steel cord belts

Values for the belt modulus are obtainable from the belting supplier.

Condition 11

Here we want to ensure that some tension remains in the outer edges of the belt, to prevent it from flopping open and shedding its load. I am sure we have all seen this occur, particularly on some older plants. It is a devilishly difficult situation to correct, without major alterations to the structure.

The radius to achieve the edge tension is given by

$$R = \frac{(E \cdot W \cdot \sin \beta)}{f_g \cdot [(0,1 \cdot T_b) - (T_x \cdot 10^3/W)} \quad m \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.4)$$

here T_b = belt class (kN/m)
f_g = Belt factor
= 9000 for fabric belts
= 4500 for steel cord belts

These formulae all look very imposing and off-putting. However, programming them into any computer makes their operation simple.

Choosing the Radius

Finally, we choose a radius that is larger than the largest value we determined. Bear in mind that the radius determined by condition 1 is usually the largest. The location of the curve with respect to the loading point and the degree of lift (if any) will help you to decide if you want to ignore that value. The condition (1) is a rare condition and may never occur in your system. If this is so, then you can ignore it completely. On the other hand, if the tangent point is only about 5 m from the loading point of your conveyor, then the possibility of this condition arising becomes more likely.

3.2 Laying Out the Curves in the Gantries or Stringers

We will all have seen that draughtsmen often seem to have great difficulty in laying out the curves to fit into the gantries or onto the stringers, in such a way that the gantries do indeed line up at their set out points. We then find stringers with steps, sudden changes in inclination, dams, weirs and so on - altogether a totally unsatisfactory situation.

We have developed a system where we recognise the relationship between the radius, the stringer length and the idler pitch and packing, and apply this to lay out the curve. The adoption of this system means that the conveyor designer will have to have prior knowledge of the intended stringer length or gantry length for the particular project. This is great, because it implies that the conveyor designer will have to communicate with the structural designer.

Applying a Simple Set of Rules

In order to simplify the layout and detail of gantry ends and stringers on conveyors with concave vertical curves, we need to apply the rules as outlined below. By the way, in the rules set out below, the words **stringer** and **gantry** are synonymous

Rule 1 : How Many Stringers ?

There must be a whole number of stringers spanning the subtended angle through which the belt curves. The terminal stringers are placed parallel to the belt at the terminal inclinations. The belt tangent point is located at one half of the stringer length at each end .

For example, consider a belt at 7°, curving through 8° up to 15°, with 6m long stringers. The first stringer will be inclined at 7°, with the tangent point located at a point 3m into the stringer (the first half of the belt being parallel to the stringer at 7°). The last stringer will be inclined at 15°, with the tangent point located at a point 3m into the stringer (the last half of the belt being parallel to the stringer at 15°) See figure 4.

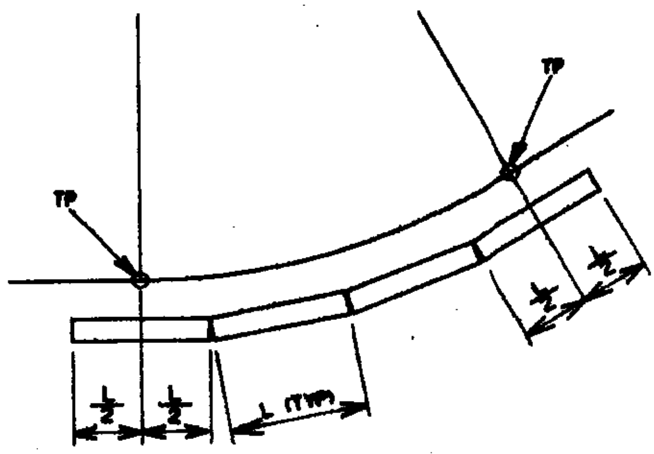


FIGURE 4

LAYOUT OF GANTRIES IN THE CURVE

Rule 2 : Naming the Variables

Total included angle .... θ°
Number of stringers .... n
Angle subtended by each full stringer ... $\alpha^\circ = (\theta/n)^\circ$
Full stringer length .... L m

Rule 3 : Determining the Radius

The radius of the belt is then a function of the included angle and the stringer length and is peculiar to that particular combination.

$$\text{Belt Radius} \quad R = \left[\{L/2\cdot\tan(\tfrac{1}{2}\cdot\alpha)\} - h\right] \text{ m} \quad ....(3.5)$$

here the variable h is defined as the vertical height of the idler set from the top of the stringer to the belt line. For idlers conforming to SABS 1313, the value will be

$h = 0{,}203$ m for stringers with no deck plates.
$h = 0{,}206$ for stringers with a 3mm deck plate.

Rule 4 : Locating the Stringers

The first half-stringer in the curve subtends an angle $(\frac{1}{2}\cdot\alpha)^\circ$. This stringer is at the same inclination as the approaching belt line. The next full stringer will be inclined at α°, with the next at $2 \cdot \alpha°$, and so on, until the last half-stringer is at slope $\theta°$. The inclinations are all with respect to the belt line at the start of the curve, of course.

3.3 Packs Under the Idlers

In order to allow the belt to more closely follow the curve profile and to avoid the situation where we have a mini lift-out, we need to pack the idlers on the stringers. These packs must be arranged in such a way that the belt is presented with a smooth idler bed. The pack height should not exceed 100mm for practical reasons. Larger packs may be used, provided that the stability of the idler set is not compromised.

The idler pitch is spaced along the line of the stringer in order to simplify setting-out and fabrication. The idlers are pitched symmetrically about the centre of the stringer bed, in such a way that the distance from the stringer end set-out point to the first idler on the bed is at one half of the standard idler pitch, at each end of the stringer. In this way, the idler pitch will remain reasonably constant and we won't have the variations at the gantry ends that we so often see.

Information and Calculation

The angle subtended by one idler pitch .... δ °
The idler pitch .... p m
The idler diameter .... d m

Then $$\delta = 2 \cdot \sin^{-1}\left[p/\{(2 \cdot R) + d\}\right]° \qquad \text{.....................(3.6)}$$

Location of the Packs - Odd Number

At the centre of the stringer length, no pack is required. The number of idlers on the stringer is given by $N = L/p$. We only use integers, no fractions, since we cannot have an half idler. If N is an odd number, there will be an idler at the stringer centre and no pack is required there. The pack height (y_1) required at the following and subsequent idlers, working outwards on the stringer length is given by

$$y_1 = R\{1 - \cos(m\delta)\} \quad \text{m} \qquad \text{....................(3.7)}$$

here m is the idler number, counted from the mid span.

Location of the Packs - Even Number

If N is an even number, there will be no idler at mid span. The packs required for the first idler (at ½ pitch) from mid span is given by

$$y_1 = R\{1 - \cos(\tfrac{1}{2}\delta)\} \quad \text{m} \qquad \text{....................(3.8)}$$

The subsequent idler packs are given by

$$y_1 \ldots y_m = R\{1 - \cos(m\delta + \tfrac{1}{2}\delta)\} \quad \text{m} \qquad \text{................(3.9)}$$

3.4 Belt Lift out of the Curve

Occasionally, we need to cater for the belt lifting out of the curve, such as the design that ignores the radius given under condition 1, or the approach belt to a tripper. There is also the possibility (and it is not a rare one) of the space available for the curve dictating the actual curve to be installed. In these cases, the belt will lift out of the idler bed and it is useful to be able to predict the amount by which the belt well lift. It is also useful to be able to plot any point on the lifted line, in order to design clearances, chutework, and so on.

More Simple Rules

Below are some simple formulae that we can apply to the known radius, in order to arrive at the results we need.

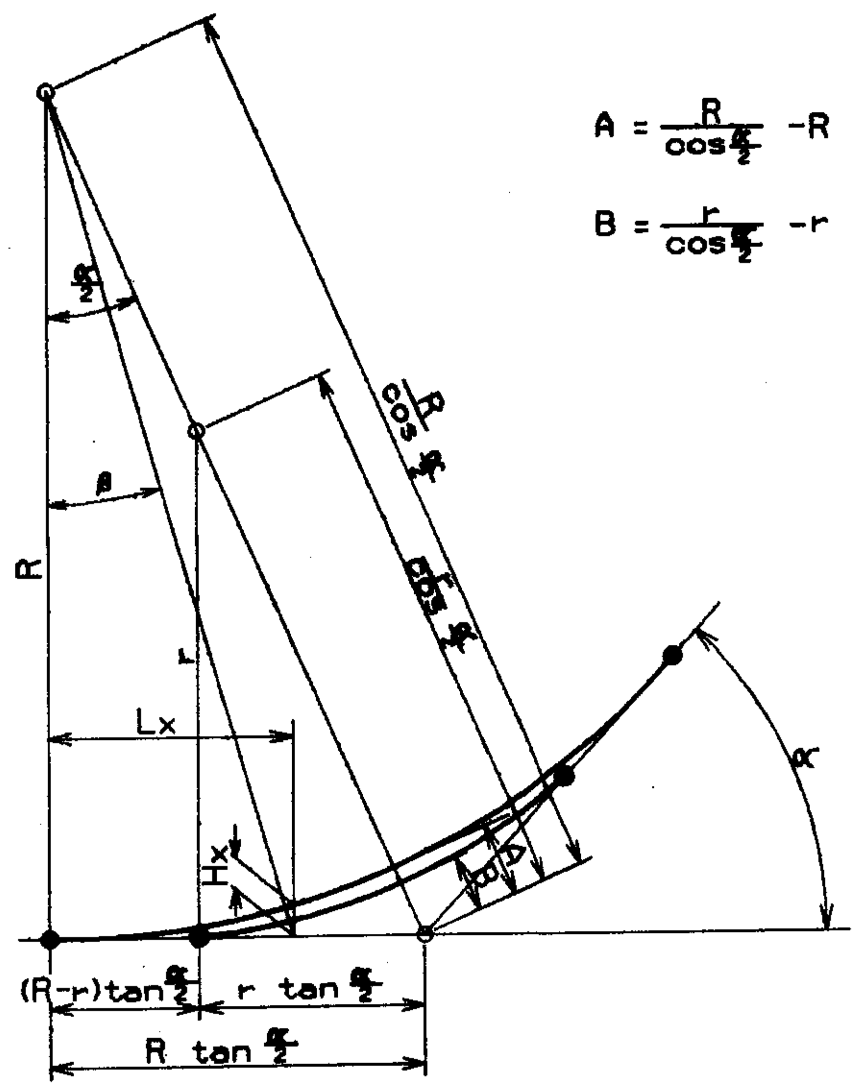


FIGURE 5

SKETCH OF THE BELT CURVE WITH THE LIFT RADIUS

Referring to the figure 5 above, the amount of lift out, measured at the centre of the curve, normal to the curve, is given by

$$H_1 = \frac{(R - r)\{1 - \cos(\alpha/2)\}}{\cos(\alpha/2)} \quad m \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.10)$$

A far more dangerous aspect of belt lift, and one that is most likely to be overlooked, is the fact that the tangent point on the belt moves back along the belt line, to satisfy the new increased radius. This has the possibility of the belt lifting into close chutework or other steelwork in the vicinity. If a belt weighmeter is located near the curve, the calibration of the unit becomes a near impossibility, because the belt rising out of the curve will reduce the load seen by the weigher. The amount that the tangent point moves back is given by

$$L_x = (R - r)\cdot\tan(\alpha/2) \quad m \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.11)$$

The belt rise at any point along the line can be determined by

$$H_x = R\cdot(1 - \cos\beta) \quad m \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.12)$$

Here, $\beta = \sin^{-1}(L_y/R)^\circ$, where L_y is the distance to the point to be considered. A common point to look at will be the difference between the tangent line for the two radii, namely equation 23.

Armed with these three formulae, we can adequately describe the vertical curve under lifting conditions.

3.5 Idlers in Convex Curves

We could develop a general case for the idlers and stringers in convex curves. The major difference is that, with a concave curve, we start with the tangent point at mid span of the stringer, packing to the outside, whereas, with the convex curve, we start with the tangent point at the stringer end, packing to the centre.

* * * * *

4. HOLDBACKS ON CONVEYORS

The holdback on the conveyor is probably the one unit which is most likely to be neglected form a maintenance point of view. To the uninitiated or untrained personnel, the holdback has apparently no effect on the running of the conveyor. This is largely true, in the forward condition, but the effects of the holdback failing are usually devastating.

Bad Experiences

We have had the experience where the low speed holdback, of the roller ramp and oil lubricated type, was fitted to a conveyor. The conveyor is about 400m long with a lift of around 70m. There came a day when the holdback oil seals failed, probably as a result of overfilling and high pressure wash down. The unit was filled with grease and there was no apparent effect on the conveyor. About three months later, the conveyor was subject to a full load power trip. Unfortunately, by that time, the grease in the holdback had hardened to the extent where the rollers were trapped off of the ramp. Hence no holdback, and about 8 tons of rock deposited over the tail pulley.

What do the Suppliers say ? The First Approach

Most holdback suppliers will tell you that the sizing of the unit is dependent on the installed power of the system. They then proceed to factor the power by 250%, to cater for emergencies. While there is nothing inherently wrong with this approach, there is the danger of significantly oversizing the units, especially if rationalisation has taken place. The oversizing then can lead to overpricing, which is not healthy at all.

When Do I Need to Install a Holdback ?

Traditionally, the requirement for a holdback on a conveyor was given by ; "When the power required to lift the load is greater than one half of the power required to move the load horizontally, then a holdback is required". Translating this, we can say if

$$T_L > 0,5(T_e - T_L - T_{a1}) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(4.1)$$

then a holdback is required.

Here T_L = Lift tension kN
T_e = Total Effective tension kN
T_{a1} = Material acceleration tension at loading point. kN

Full Belt Requirement - The Second Approach

Using the statement above, we can determine what will be probably the minimum requirement for the holdback.

The holdback rating will then be :

$$Q_1 = \frac{\{(1,5 \cdot T_L) - 0,5(T_e + T_{a1})\} \cdot D_P}{2000} \text{ kNm} \quad \ldots(4.2)$$

Here D_P = Pulley diameter m

Getting Closer - The Third Approach

Another (and probably more realistic) approach would be to consider the tension generated at the holdback station with only the inclined portion of the conveyor loaded, and the rest of the conveyor empty. The empty portion is considered as having a reduced friction factor as well, in order to maximise the holdback rating. Using this approach, the empty belt tension (T_{eb}) of the system is calculated using a friction factor reduced to a suggested value of **0,016**. The empty belt tension then becomes

$$T_{ebx} = 0{,}016 \cdot T_{eb}/C \quad \text{kN} \quad \ldots\ldots(4.3)$$

where C is the friction factor utilised in the main design.

The holdback rating will then be :

$$Q_2 = \frac{\{(1{,}5 \cdot T_L) - 0{,}5(T_{ebx} + T_{a1})\} \cdot D_p}{2000} \quad \text{kNm} \quad \ldots(4.4)$$

The Maximum Rating - The Fourth Approach

There have been reported incidents of conveyors "spragged" at the tail, or at some loading point along the belt. This occurs when an object (usually a loose liner plate, a jumper bar or even an rogue lump) becomes jammed between the belt an the chutework or the conveyor supporting structure. The conveyor power pack continues to pump power into the system, with the top strand stationary. At some point, one or more of a few things occurs.

Either (i) the belt breaks, or

(ii) the motor trips, or

(iii) the belt comes to a halt, with the motor still running, and the driving pulley simply slipping in the belt, or

(iv) The fusible plug on the fluid coupling blows.

If the first condition occurs, the belt is going to run away anyway, and the holdback will have little or no stopping effect on the system, depending on the location of the break.

If any one of the other conditions occur, the belt will have a tremendous amount of stored energy and the holdback will have to be sized in such a way that it will neither slip nor explode.

The belt spragged condition can be expressed as an oversizing factor, which is applied to the belt effective tension T_e, in order to establish the holdback rating. This oversizing factor is directly dependent on the **total** angle of wrap in the drive, as well as the standing or slack side tension T_2.

Since T_2 is applied externally, by means of the take-up mass, it may be considered constant under steady-state conditions. Any additional tension applied to the system is then limited to a maximum value. This value is determined when the driving pulley, or pulleys, are on the point of slip, or when the maximum transmissible torque of the fluid coupling is reached (or it blows it's fusible plug), or when the motor trips. The least of these values is then used to determine the holdback rating.

4.1 Determination of the Oversizing Factor

Remember that $T_1 = T_2 + T_e$ ....(4.5)

that is, the "tight side" tension (T_1) in the conveyor is the sum of the effective tension (T_e) and the standing tension (T_2).

For overtensioning as a result of the spragged condition,

$$T_{1x} = T_2 + T_e + T_x \quad \text{..........................(4.6)}$$

where T_x is the additional tension applied to the system.

Since $T_1/T_2 = e^{\mu\theta}$

$$(T_2 + T_e + T_x)/T_2 = e^{\mu\theta}$$

ie $$(T_2 + T_e + T_x) = T_2 \cdot e^{\mu}$$

$$(T_e + T_x) = T_2 \cdot e^{\mu\theta} - T_2$$

$$= T_2 \cdot (e^{\mu\theta} - 1)$$

Then the oversizing factor $f_o = (T_e + T_x)/T_e$

$$f_o = \{T_2 \cdot (e^{\mu\theta} - 1)\}/T_e \quad \text{.....(4.7)}$$

Here μ = Coefficient of friction, belt and driving pulley.
θ = Angle of wrap on the pulley, in radians.

Then the holdback maximum,rating becomes

$$Q_3 = f_o \cdot T_e \cdot D_p/2000 \quad \text{kNm}$$

ie $$Q_3 = \{T_2 \cdot (e^{\mu\theta} - 1)\} \cdot Dp/2000 \quad \text{kNm ....(4.8)}$$

Note that the angle of wrap (θ radians) is taken as the total actual angle of wrap in the drive. We do not assume a drive wrap here, even though we may have used one in the main design, since we want to ascertain the worst case.

The maximum value for f_o = 2,5 ....(4.9)

The holdback rating so determined is usually the largest of the values determined, excluding the first case - sizing on motor power, which can be greater, especially for a single drive.

Choosing a Holdback Unit

The holdback rating we have determined is referred to the pulley shaft speed and is thus the low speed unit rating. Armed with the information we have just determined, we will be in a position to make more than a passing guess at the suitable holdback. Most reputable manufacturers of holdbacks base their catalogue ratings on a safety factor. This factor is usually between 1,5 and 2,0. If the maximum rating of the holdback falls below the catalogue rating, but is satisfied by the maximum rating with a safety factor of 1,0 then the unit will probably be suitable. Remember that the spragged condition is one that is not likely to occur frequently, if it occurs at all.

(As an aside, if you are experiencing frequent sprags, then you ought to be looking at the cause of those sprags, and at ways of eliminating them. Each time a conveyor is spragged, the loads imposed on the structure, on the belting and pulleys, in fact on the whole system, are quite large.)

Where do I Put the Holdback ?

The holdback can be either a low speed or a high speed unit. The low speed units are typically between 10 to 12 time more expensive than the high speed units. This is often the motivating factor behind the choice of high speed units in preference to the low speed units.

The high speed units have a reduced torque, which is determined by the reduction ratio of the reducer, at the point where the unit is mounted. If it is mounted on the input shaft of the reducer, as with worm boxes, then the rating is reduced by the full reducer ratio.

Helical and bevel helical reducers can have the unit mounted on either the input shaft, or one of the intermediate shafts. The units are mounted either completely internally, or on an extension of one of the intermediate stage shafts, as a semi-external unit. The holdback rating is then reduced by the ratio up to that point, working back from the output shaft.

The low speed unit is usually mounted on the driving pulley shaft, or on the secondary drive pulley shaft in the case of a dual drive.

4.2 High Speed Versus Low Speed

The choice of high speed or low speed units depends on your budget, or on the conveyor application which requires the holdback. It is fair to say that the majority of catastrophic mechanical failures on conveyors in this country occur in the drive train. These failures usually occur when the conveyor is fully loaded.

Now, if the conveyor is in a holdback situation, with the problem being at the drive, it is very difficult to remove the drive without some form of external sprag on the conveyor. The alternative is to manually lash the burden off the conveyor, depending on its size. All this means that we will need some form of external holdback anyway.

Should the low speed coupling fail, then there will be no holdback anyway, since the reducer will be effectively removed from the system.

In the case of multiple drives, if each reducer is fitted with a holdback, it must be sized to withstand the full torque, unless load sharing units are installed. This is so, since the reducers are not likely to have the same amount of backlash. This in turn implies that one of the units will engage before the others, requiring that it be capable of withstanding the full torque. This in turn implies that each reducer itself will be capable of withstanding the full torque, without failing.

A point to bear in mind, is that the design lubricant for the internally mounted holdback must be compatible with the reducers lubricant. There have been reported cases of internal holdbacks slipping because the reducer was filled with synthetic or EP oil.

For these reasons, my personal preference is to use the low speed units, They are not dependent on the drive for their operation and can be installed in such a way that they may be easily maintained. If necessary, load sharing can be achieved, either by means of static hydraulic linking or by mechanical linking.

<u>Selecting the Holdback Pulley</u>

The normal runback tension, which we can call T_h, is essentially an effective tension in reverse. The slack side tension is determined by T_2.

Then, **in the runback condition**, the "tight" side tension (T_{1h}) is given by:

$$T_{1h} = T_h + T_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.10)$$

and $\quad T_{1h}/T_2 = e^{\mu\theta}$

thus $\quad \boldsymbol{\theta = \ln(T_{1h}/T_2)\cdot(180/\mu\pi)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.11)$

This is the minimum wrap required, under normal conditions only, to prevent the belt slipping back over the pulley fitted with the holdback. This allows the designer to position the holdback unit remote from the drive pulley, or to select the pulley to be equipped with the holdback.

<u>4.3 Controlled Release of the Holdback</u>

Some years ago, the British National Coal Board, now called The British Coal Corporation, reported at least two separate incidents of conveyor sprag. The holdback units operated efficiently in both cases, successfully holding the belt from reversal. When the sprags were being manually removed, that is, the coal was being lashed off the belt at the tail end, the situation was reached where the sprag was suddenly cleared. The sudden release of all that stored energy resulted in the workers, in both reported incidents, being killed. A similar incident was recently reported in the journal of the Association of Mine Resident Engineers in this country. This time, the sprag was reported as occurring on a gold mine. So we can see that this sort of unfortunate thing is not the exclusive domain of the coal mines.

The reported incidents in the U.K. led the National Coal Board to specify controlled release on all new installations requiring a holdback. The argument is that the stored tension in the system is released in a controlled manner, allowing the sprag to be cleared in safety.

There are at least two holdback manufacturers that supply units that have a controlled release facility. Both systems operate in essentially the same way. The unit is basically a high speed holdback, fitted with an additional friction plate clutch arrangement. The clutch is held by means of springs or Belville washers on bolts. By releasing the pressure on the friction medium, the holdback is allowed to slip backwards, until the tension in the system is evened out.

The major disadvantage of the systems is that they are currently only available as high speed units. This effectively excludes the multiple drive situation, as described above. Secondly, the units are not self resetting, and we can thus have the situation where the conveyor could be restarted with the holdback units still disengaged - a potentially dangerous situation.

Alternative to Controlled Release Holdbacks

A simple means of controlled release of the stored energy in a conveyor is to provide the take-up with a means of release. In this way, the T_1/T_2 tension ratio is destroyed and the belt will slip back over the held pulley. The advantage of this system is that the conveyor will not start unless the take-up unit is reset. Of course, there is also the advantage of cost. The addition of a facility to release the take-up will cost no more than a few minutes of the draughtsman's time in the original design. The facility is also easily retro-fitted to existing units.

* * * * *

5. VIBRATIONS IN THE RETURN STRAND.

The return strand of a conveyor is probably the one area most neglected by conveyor designers and operators. For example, how many times has a conveyor been designed with a perfectly adequate vertical concave curve, a far as the top strand is concerned, only to have the return strand run into the underside of the steelwork ? The designer failed to appreciate the significance of the fact that he has placed the drive internally, remote from the head. This gives rise to a return strand tension **before the drive**, greater than the "tight side" tension T_1. The belt has only its own mass to keep it down, and it loses the battle, of course. Using the approach outlined in section 3 above, we can establish the radius of the conveyor on the return strand at any point.

The area I wish to address is the pitching of the return strand idlers, with reference to the vibrations in the return strand.

We have seen in section 2 above that we can establish the pitch for the return strand, based on the bearing life, the shaft parameters and the sag of the return strand. There remains a further parameter to investigate, namely the pitch of the idler to minimise vibrations, or belt flap.

What is Belt Flap?

The return belt "flaps" for a number of reasons. The most important reason, given that there are no untoward dynamic problems, is as a result of excitation at or near to the natural frequency of vibration of the conveyor. The belt displays an up and down motion which can have a magnitude great enough to cause premature idler failure. The interaction between the tensioned belt and the supporting idlers is important in system design. Failure to recognise the potential problems and to deal with them at the design stage, can turn an otherwise good design into one that causes the operating personnel headaches.

The Belt and a Piece of String

If the tensioned return strand is treated as a tensioned string, it will have a natural frequency of vibration that varies with the tension.

This frequency is given by:

$$f_n = \frac{1}{2 \cdot P_r} \left[\frac{T_x}{B} \right]^{\frac{1}{2}} \quad \text{Hz} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5.1)$$

Here, T_x = Tension at the point under consideration kN
P_r = Return idler pitch m
B = Belt mass kg/m

5.1 Determining the Imposed Frequency

If the tensioned "string" is excited at, or near, its natural frequency of vibration, it will tend to vibrate with maximum amplitude, or **resonate.** Designers of belt rappers have long used this approach to optimally locate the belt rapper with respect to the squeezer, in order to achieve the maximum amplitude, and therefore the maximum cleaning effect. Actually, a section of return strand that is resonating cleans the belt rather well! The imposed frequency is generated by any rotating part of the system, in contact with the belt. That effectively includes all the return idlers, bend pulleys, take-up pulley, driving pulleys and so on. The frequency is given by the rotational speed of the particular item, with respect to the belt speed.
The imposed frequency is given by

$$f_i = \left[\frac{S}{\pi \cdot D_x} \right] \text{ Hz} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5.2)$$

here S = Belt speed m/s
D_x = diameter of pulley or idler at the point under consideration m

Points to Consider

Armed with the information above, we can proceed to compare the natural frequency of vibration of the belt (the frequency is continuously changing, as the tension changes), with the imposed frequency at each station. That means checking every idler pitch.

The worst case is when $f_i = f_n$, when resonance will occur. Ideally, we try to keep the value of f_i between $0{,}7 \cdot k \cdot f_n$ and $1{,}3 \cdot k \cdot f_n$. The variable **k** is an integer, being a multiple or sub-multiple of the natural frequency of vibration.

If we find that f_i does fall inside these limits, then we must adjust the idler pitch in the vicinity of the offending vibration. Adjusting the idler pitch alters the natural frequency of vibration and can take the section of belting out of resonance.

Manipulating the Formulae

From equations 5.1 and 5.2 above,

When fi = fn , (the worst case)

$$\frac{S}{\pi \cdot D_x} = \frac{1}{2 \cdot P_r} \left[\frac{T_x}{B} \right]^{\frac{1}{2}}$$

thus

$$P_x = \frac{\pi \cdot D_x}{2 \cdot S} \left[\frac{T_x}{B} \right]^{\frac{1}{2}} \text{ m} \quad \ldots\ldots\ldots\ldots\ldots\ldots(5.3)$$

and

$$T_x = B \cdot \left[2 \cdot P_x \cdot f_i \right]^2 \text{ kN} \quad \ldots\ldots\ldots\ldots\ldots\ldots(5.4)$$

If the spread of the natural frequencies across the tension in the return strand is such that f_n approaches f_i at some point, it will be necessary to establish the tension and location of that tension where the resonance will occur. Once this position is established, the idler pitch must be adjusted in this area, such that

$$f_i > 0{,}7 \cdot k \cdot f_n \quad \ldots\ldots\ldots\ldots\ldots\ldots(5.5)$$

and

$$f_i < 1{,}3 \cdot k \cdot f_n \quad \ldots\ldots\ldots\ldots\ldots\ldots(5.6)$$

These manipulations allow the designer to more easily locate potential trouble spots and to take the necessary action to rectify the predicted resonance.

* * * * * * *

6. ACKNOWLEDGEMENTS

The author wishes to thank the Anglo American Corporation for permission to publish this paper and to take part in this important workshop.

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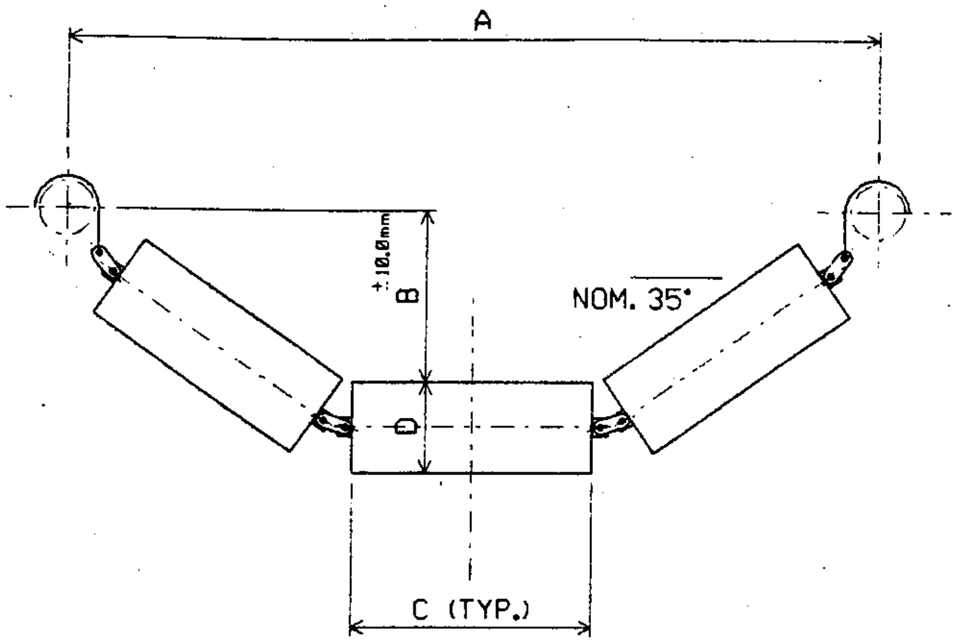

BELT WIDTH	A	B	C	D
600	920	175	240	127
750	1050	200	290	127
900	1180	230	340	127
1050	1310	260	390	127
1200	1470	295	450	127

FIGURE 6

NOTE: (1) DIMENSION 'A' SHALL BE TOLERANCED TO ±1,0mm FOR DIMENSIONAL TEST PURPOSES ONLY

(2) DIMENSIONS 'C' & 'D' ARE IN ACCORDANCE WITH SABS 1313

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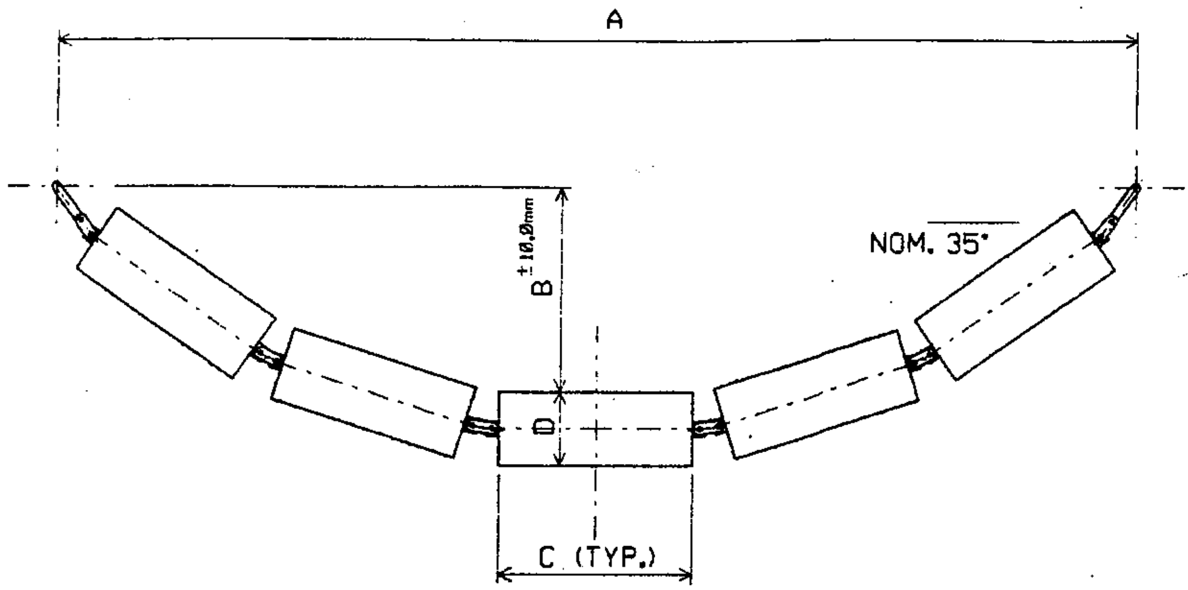


BELT WIDTH	A	B	C	D
1350	1865	355	300	127
1500	2050	390	340	127
1650	2185	415	370	127
1800	2320	440	400	127
2100	2660	510	475	127
2400	2940	560	536	127

FIGURE 7

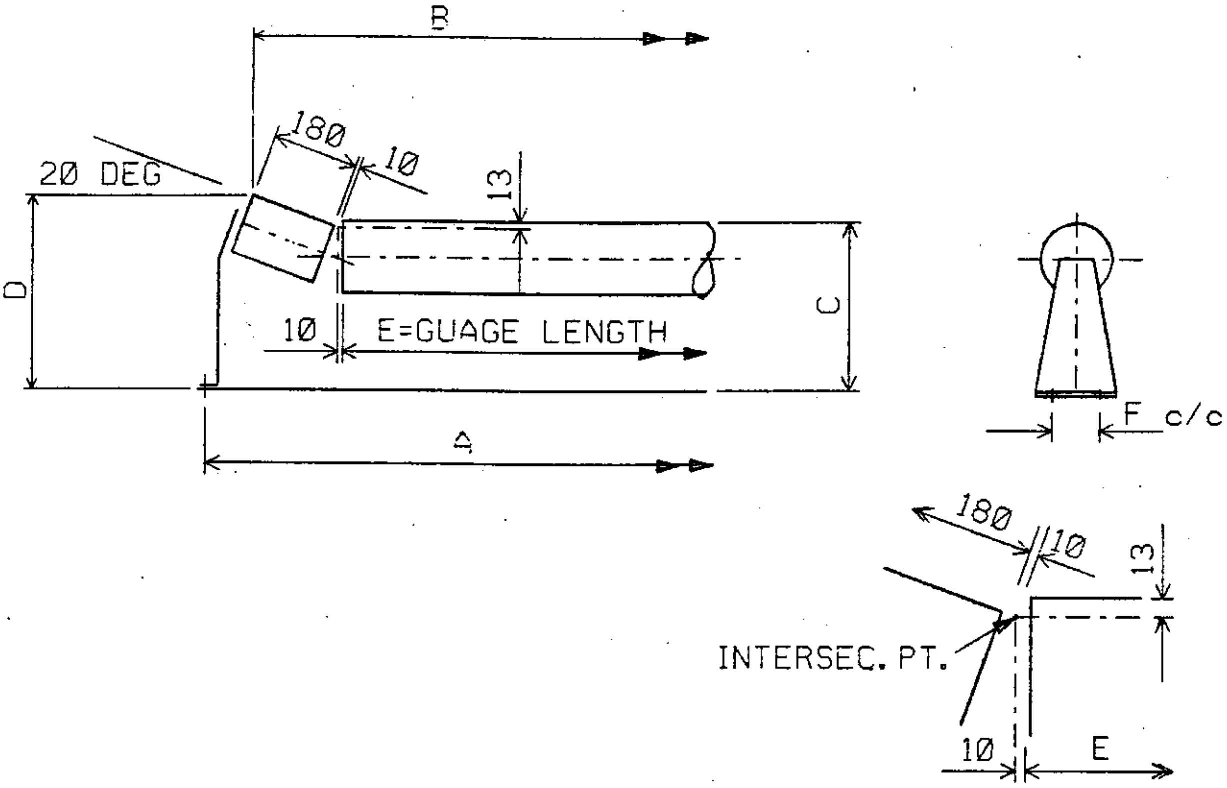
NOTE: (1) DIMENSION 'A' SHALL BE TOLERANCED TO ±1,0mm FOR DIMENSIONAL TEST PURPOSES ONLY

(2) DIMENSIONS 'C' & 'D' ARE IN ACCORDANCE WITH SABS 1313

rg No Y1890 ~ Sheet 1 of 1

IN-LINE TYPE
PICKING IDLERS

PROPOSED STANDARD DIMENSIONS



BELT WIDTH W	A	B	SLOTTED HOLES	C	D	E	F	CLEARANCE BELT EDGE TO ROLL EDGE
750	990	922	14X25	265	315	546	100	93 APPROX
900	1144	1074	14X25	265	315	698	100	94 APPROX
1050	1296	1226	14X25	265	315	850	100	95 APPROX
1200	1448	1380	14X25	265	315	1004	100	97 APPROX
1350	1600	1532	14X25	265	315	1156	100	98 APPROX
1500	1752	1684	18X30	285	335	1308	150	99 APPROX
1650	1904	1836	18X30	285	335	1460	150	100 APPROX
1800	2058	1988	18X30	285	335	1612	150	101 APPROX
2100	2362	2294	18X30	285	335	1918	240	102 APPROX
2400	2668	2596	18X30	285	335	2222	240	103 APPROX

FIGURE 8

WING ROLLS 125 NOM. DIA. SERIES 25
CENTRE ROLL 150 NOM. DIA. SERIES 30
WITH SPHERICAL ROLLER BEARINGS
(WHICH MAY BE SPECIFIED AS A RUBBER
IMPACT ROLL. IN WHICH CASE DIM. C, AS MAX
WILL INCREASE BY 10mm)
E=[1.017(W-300)+87.14] APPROX
FOR TOLERANCES SEE SABS 1313

PART OF DRAWING NUMBER

BEL508AS

BELTCON WORKSHOP PAPER ENTITLED

"CONVEYOR DYNAMICS"

By A. Surtees
Pr Eng.
September 1991

Table of contents:

1. INTRODUCTION

This papers concerns practical aspects about transient belt tensions. These are the tensions which occur in the belt during the transition period - From stationary to running condition or vice versa. i.e. Starting and stopping.

These tensions are usually the largest which can ever occur in a conveyor system, and if not taken into account, can cause severe damage to the belting, tensioning devices and associated structures.

This paper deals only with transient tensions which occur during starting a conveyor. Transient tensions during stopping are not discussed but obvious similarities exist.

Videos and a simple working model, analagous to a conveyor drive, are used to show the different tensions and the two main factors which contribute to undesirably high transient tensions.

A simple rule is then introduced to allow high transient tensions to be avoided at the design stage.

Two worked examples are then discussed showing the existence of undesirable tensions and steps taken to overcome them.

Finally, before concluding, site measurements and observations of transient tensions are briefly discussed.

2. VIDEO

The video shows the destruction of the Tacoma Narrow Bridge by excitation of its natural frequency, as well as other pertinent material.

3. SIMPLE WORKING MODEL

This demonstration of the simple model clarifies the various tensions: i.e. T_1, T_2, T_E and accelerating tensions:- Rate of introduction into belt and limitation of magnitude.

4. SIMPLE RULE TO AVOID UNDESIRABLE TRANSIENT TENSIONS

This concerns the first main factor contributing to high tensions:- rate of introduction of tension into the belt to accelerate the system.

Simple Rule: The ramping time from zero accelerating tension to maximum accelerating tension must be at least 5 x the time it takes for a disturbance to travel from the head to the tail in the return belt. (Ref: Various papers by Dr. Funke, Hannover University)

i.e. $t_{RAMP} > 5 \times t_{BELT}$

$$\text{WHERE } t_{BELT} = \frac{\text{CONVEYOR CENTRES (L)}}{\text{WAVE ADVANCE VELOCITY (v)}}$$

Values for the wave advance velocity v are dependent on the belt construction and formulae to calculate them are available in various publications. (Funke, Harrison, Nordell Et Al.)

To allow ease of calculation (without losing too much accuracy), the following values may be used.

Steel cord belts:	1800 m/s
EP belts:	1200 m/s
Solid woven belts:	900 m/s

The results are summarized in graph no. 1 in the appendix. It is obvious from this graph that for the same length conveyor, the minimum ramp time of the solid woven must be twice as long as that for a steel cord belt.

By ensuring that the acceleration tension stays below the critical line (refer to graph no.2), the existence or effect of additional undesirable tensions can be ignored. i.e. For belt strength calculations, only the maximum magnitude of the accelerating tension needs to be allowed for (i.e. starting factor as a %age of T_E).

5. WORKED EXAMPLES

5.1 Rate of tension introduction into belt exceeding the critical line.

(See calculations and graphs in the appendix).

5.2 Maximum acceleration tension exceeding that value allowed for in the design

(See calculations and graphs in the appendix).

6. SIMPLE SITE OBSERVATIONS AND MEASUREMENTS TO INVESTIGATE THE EXISTENCE OF DYNAMIC TENSIONS

The following observations or measurements can be made:

- Starting and stopping times
- Take up weight displacement
- Winch loadcell tension
- Motor start up sequence
- Peak motor currents
- Belt over and underspeeding

Simple formula that can be very useful are as follows:

$$t_{STOP} = \frac{mv}{T_E}$$

$$t_{START} = \frac{mv}{T_S}$$

Whereas;

t_{START} = START TIME FROM "BREAKAWAY"

T_S = ACCELERATION TENSION

m = SYSTEM MASS

v = FULL BELT SPEED

With the first formula, given the stopping time, and knowing the mass of the conveyor system and the full belt speed, the effective tension can be calculated. This can give a good idea of the power requirements for the conveyor. (Compared to what is actually installed).

Knowing the stopping and starting time for the belt under similar loaded conditions;

$$\frac{T_S}{T_E} = \frac{t_{STOP}}{t_{START}}$$

Where T_S = Average acceleration tension.
This formula allows checking of the torque limitation or starting factor for the conveyor.

7. CONCLUSIONS

If the rate of tension introduction during acceleration of the conveyor is less than the critical value, and the peak acceleration tension value is taken into account at the design stage, all aspects of conveyor dynamics can be ignored. The same applies to braking.

Note that this is a general statement and does not consider take up types and geometry, tail or intermediate drives (tail or booster). It has however proved to be a useful tool in overcoming existing problems and when used in new designs, has obviated individual dynamic analyses for medium length conveyors:- up to 6 km for steel cord belts and 3 km for solid woven belts.

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