DRIVE CONFIGURATION EFFECTS ON CONVEYOR BELT COVERS

1 INTRODUCTION

The design of conveyor belts and the materials used in their construction have evolved over the years in order to cope with the ever increasing demands placed upon them. Cotton fabric gave way to polyamide and polyester materials for the tension member in the belting to increase the strength yet retain the degree of flexibility necessary for troughed conveyor belts. However, with the need for longer conveying distances and higher capacities, these materials were incapable of performing their required function due to both the excessive rigidity of the belting, caused by multi-strand thickness, and excessive elasticity of the resultant system.

Subsequently, higher powered conveyors utilised steel cord belting where the steel cords dramatically increased the tension characteristics of the belting whilst retaining belting flexibility. The successful introduction and use of such conveyors in British coal mines is well documented. In recent years however, belting manufacturers have been encouraged to develop solid woven fabric core belting beyond the proven Type 15 belting, ie 15000 lbf/in width, in order to eliminate the problems of susceptibility of steel cord belting to damage and hence corrosion of the steel cords. However, a successful design of fabric belting would have to compete with steel cord belting in terms of minimising stretch such that loop take-up lengths on long conveyors are minimised, and hence minimise the cost of expensive enlarged roadway cross sections.

This paper is concerned with two underground belt conveyor installations utilising Type 18 belting, ie 18000 lbf/in width, in British coal mines and the widely different results experienced at each site.

2 DEVELOPMENT OF THE TYPE 18 BELTING

The Type 18 belting, in a solid woven construction, was made possible due to the introduction of a completely new design. This allowed ease of manufacture and a reduction in the belting thickness normally associated with higher tensile belting. It also gave the low stretch parameters considered essential for such installations.

The stretch parameters specified for the Welbeck installation resulted in a design of belting with a particularly high modulus which, it is felt, unnecessarily compromised the performance of the belting on this design of drive. In the event, the installation could have accommodated belting with a lower modulus which would have improved its performance.

Accordingly, the Type 18 was redesigned with a lower modulus fabric than originally specified. This design of belting, whilst not satisfactory, was a distinct improvement on the earlier specifications.

3 TRIAL SITES

The first installation incorporating Type 18 belting was located at Gascoigne Wood in the Selby coal field. The conveyor was commissioned in March 1992. The conveyor operated very successfully meeting the requirements of the colliery for a low stretch textile carcase belt. The decision was then taken to widen the operational experience with the Type 18 belting by commissioning a second site at Welbeck colliery, in the Nottinghamshire coal field. The technical specifications of the respective installations are shown in Table 1.

DESIGN ASPECT	GASCOIGNE WOOD	WELBECK
Horizontal length (m)	2530	1420
Vertical lift (m)	110	200
Conveying capacity (t/hr)	2400	1000
Material density (kg/m^3)	880	880
Belt speed (m/s)	4	2.6
Belt width (m)	1.35	1.2
Belt thickness (mm)	22.5	22.5
Installed power (kW)	1490	895

TABLE 1: CONVEYOR BELT SPECIFICATIONS

4 OPERATIONAL EXPERIENCE

4.1 Gascoigne Wood Colliery

Initial problems were encountered with respect to tracking of the belt. The degree of belt drift was in the region of 200 mm. Preliminary investigations suggested that one joint had not been made correctly and was in effect not square. The joint was remade but the problem persisted. It was then thought that the spliced joints were too stiff and subsequently the joints were remade, again without success. Further detailed analysis revealed that the belt edge tensions were not uniform due to the way in which the polyester carcase had been produced. The belting manufacturer has subsequently corrected this aspect. The tracking problem has been resolved by installing a cable belt roller horizontally such that the pulley guides the belting back on line. The forces exerted by the pulley are very low, in the order of 700 N (160 lbf), and consequently does not cause any damage to the belting. Minor blistering was experienced on the carry side of the belting. The size of blisters were approximately 20 mm in diameter. The belting manufacturer resolved this particular problem by puncturing the blisters and injecting cement as required. The problem was put down to contamination occurring at the boundary between the PVC solid layer and the nitrile rubber cover during manufacture.

However, it must be recognised that, despite these initial problems, the conveyor has operated very satisfactorily, having conveyed a total of 11 million tonnes in the 3 years since it was first commissioned.

4.2 Welbeck Colliery

The conveyor was not fully operational until September 1992. Within two or three weeks the covers showed evidence of blistering. In the early stages, the blisters ranged in diameter from 15 mm to 50 mm, and standing up to 25 mm from the surface of the belt cover. The blisters were cut and cemented back to the carcase of the belting. Continued operation of the conveyor resulted in wearing away of the blistered areas, exposing the carcase of the belting.

The blistered surface also gave rise to problems in belt cleaning, and in the level of vibration experienced by the belt structure and drives caused by the passage of damaged covers through the system. Accordingly, additional belt cleaning arrangements were installed in the form of a belt washing system. By August 1993, approximately 80 to 90 % of the belting was showing signs of blistering and distress.

Concurrently whilst monitoring the problem underground the belting manufacturer was attempting to identify the cause or causes of the blistering in the form of manufacturing deficiencies or incompatibility with the drive configuration. The initial mode of failure was thought to be a failure in the bonding between the PVC solid layer and the PVC carcase. New sections of belting, with modifications to significantly improve the adhesion between coats (both PVC and rubber), were introduced into the conveyor. Unfortunately the new sections experienced the same problem of blistering as with the original specification, though not as severe. In this instance the cause was thought to be the result of contamination between the carcase and the covers resulting in PVC/Nitrile failure. The condition of the belting continued to deteriorate to such an extent that local delamination occurred in certain sections of the belting where a whole arm could be pushed underneath the cover. This was revealed to be the consequence of shear at the boundary of the solid PVC layer and the PVC impregnated fabric carcase. Even so, by February 1994, the general condition of the belting had not significantly deteriorated since January 1993.

The belting manufacturer also involved the British Rubber and Plastics Research Association (RAPRA) in the investigation. RAPRA conducted a finite element analysis of the design of the belting to confirm (or otherwise) whether the inherent characteristics of the materials used in the manufacture of the belting were the root cause of the problem. The analysis did not reveal any undue levels of stress in the material in the belting, or at the various sections through the belt, and was therefore discontinued.

Lengths of belting were then manufactured to a lower modulus design to increase elasticity, after being subjected to a series of tests on a conveyor belting test rig in order to verify the adequacy of changes and ensure that the problem had been resolved. The tests on the original belting and subsequent specifications had been carried out with 1.0 m diameter pulleys. In all cases the belting performed satisfactorily. The decision was then taken to reduce the pulley diameters to 0.8 m and to repeat the tests. Problems were then exhibited in the belting similar to that experienced at Welbeck. By June 1994, a redesigned belting had successfully completed the test programme and a forty metre length inserted into the conveyor. Unfortunately this belting also suffered from blistering on the drive side in operation in the same fashion as the previous specification. However, it was thought that the short length of belting could have influenced the performance of the new specification of belting. At this stage, the belting manufacturer and British Coal decided to commission Alex Harrison of Conveyor Technologies Ltd, Colorado, USA, to evaluate the two conveyor specifications with particular reference to the designs of their respective drives. The evaluation applied some fundamentally new concepts in visco-elasticity to this problem. The results of the analysis are presented in the following sections.

5 ANALYSIS OF CONVEYOR DRIVES

5.1 Drive Configuration

The two drive configurations deployed at Gascoigne Wood and Welbeck collieries are shown in Figure 1. As can be seen, both are dual pulley drives but the Welbeck colliery pulley arrangement is a tighter configuration than that at Gascoigne Wood. The essential design criteria for the respective drives is shown in Table 2.

DESIGN ASPECT	GASCOIGNE WOOD	WELBECK
Primary drive pulley diameter (mm)	1000	1250
Angle of wrap (degrees)	215	229
Primary snub pulley diameter (mm)	800	800
Angle of wrap (degrees)	215	208
Distance between snub pulleys (mm)	5100	4200
Secondary snub pulley diameter (mm)	800	800
Angle of wrap (degrees)	205	230
Secondary drive pulley diameter (mm)	1000	1250
Angle of wrap (degrees)	210	229

TABLE 2: DRIVE DESIGN CRITERIA

Computer modelling of the belt tensions was conducted utilising CEMA (the American Conveyor Equipment Manufacturers Association) and ISO 5048 based techniques. The calculated values of belt tensions for the respective conveyors based on CEMA and ISO models are shown in Table 3.

FACTOR	GASCOIGNE WOOD		WELBECK	
Tonnes/hr	0	2400	0	1000
Power (CEMA) kW	204	1332	75	715
Power (ISO) kW	225	1395	91	723
T_1 kN	91	383	105	358
T_2 kN	40	50	76	83
Slip factor k	0.23	0.77	0.14	0.43

TABLE 3: CALCULATED BELT TENSIONS

It was interesting to note that the Gascoigne Wood conveyor is closer to slip than Welbeck, slip being one of the established causes of excessive shear stresses in belting covers leading to delamination, particularly under dynamic conditions. However, delamination was not a problem at Gascoigne Wood. The absolute calculated belt tensions at full load in the two systems are very similar.

The slip factor "k" was calculated from:

$$A = \frac{T_1}{T_2}$$

$$B = e^{\mu\theta}$$

$$k = \frac{A}{B}$$

where μ is the coefficient of friction between the drive and the belt, and θ is the angle of wrap. Slip will occur where k is equal to, or greater than, 1.

The belting manufacturer measured a value for μ of 0.7 at static breakaway. It is reasonable to assume that a sliding or dynamic friction coefficient value of ⅔ of the static value, ie 0.46, would be achieved in practice which would result in k=0.52 for Gascoigne Wood and 0.38 for Welbeck. Neither system was close to slipping and Welbeck had a lower T_1/T_2 ratio effect than Gascoigne Wood. This further confirmed that the problem at Welbeck was not the consequence of slip. The next stage was to review the interaction between the belting and the drives for the two systems.

Traditionally, MHEA (the British Mechanical Handling Equipment Association) design guides for conveyors recommended a minimum distance between drive pulleys of 3.5 m or the distance travelled by the belting in 1 second, whichever was the greater. This was to permit the belting to recover from the elongation resulting from the difference in belt tension between the belting coming off the primary and going onto the secondary pulley.

However, this approach is simplistic as it does not take account of the effects of the snub pulleys on stresses induced in the belting.

It is important to consider the time taken for the belting to travel between the respective pulleys, ie the flight time, which is, of course, a function of belt speed and distance between the pulley. Table 4 shows these distances for the respective systems.

BETWEEN	GASCOIGNE WOOD	WELBECK
T_1 Snub & Primary Drive	9.80 m	1.42 m
Primary Drive & T_3 Snub	1.16 m	0.92 m
T_3 Snub to T_3 Snub	5.10 m	4.20 m
T_3 Snub & Secondary Drive	1.16 m	0.316 m

TABLE 4: COMPARISON OF DRIVE LENGTHS

These values reinforced the original statement regarding the "tightness" of the Welbeck drive arrangement, in particular the distance between the T_3 snub and secondary drive pulleys of 0.316 m. A comparison of the belting lengths contained within the drive configurations for Gascoigne Wood and Welbeck gives 23.86 m and 14.91 m respectively, a ratio of 1.6. However, taking the reference points as "entry onto the primary drive drum" and "exit off the secondary drive drum", the respective belting lengths were 14.06 m and 13.49 m respectively, giving a much lower ratio of 1.04, ie the length of belting engaged in the drive was similar for both installations, contrary to the first visual impressions.

The "flight" times for the belting traversing the two systems is shown in Figure 2. These times were calculated using the stated design belt speeds. The critical "flight" times were obviously those between the snub pulleys and drive pulleys. The relatively short time periods experienced within the Welbeck drive might not be sufficient to permit the stresses induced in the belting, as it negotiated the pulleys, to reduce to satisfactory levels.

Accordingly, a representative sample of the Type 18 belting was subjected to a series of tests in order to evaluate its behaviour under dynamic shear stresses. A 20 mm by 20 mm block was held between two shear plates as a constant longitudinal force of 19.6 kN was applied. When the shear/strain approached a straight line, ie attained a stable condition, the load was removed. In this way it was possible to identify the shear creep and recovery characteristics for the belting. Figure 3 shows an example of the results of such tests.

The creep and recovery curves shown represent very long timescales in comparison to the belting "flight" times previously indicated in Figure 2. Therefore substantial levels of residual shear strain could be locked in the belting material as it started to negotiate the next pulley. Bearing in mind the short "flight" times, high speed tests were carried out to determine the initial elastic slopes. Table 5 shows the residual strains measured on a 20 mm by 20 mm sample after the removal of a 66.7 kN shear force.

ASPECT	SHEAR RECOVERY TIME (s)	% OF TOTAL RECOVERY
Elastic Component	0.06	30.0
Visco-elastic Component	0.12	36.6
	1.0	58.5
	2.0	63.4
	100.0	96.5

TABLE 5: SHEAR RECOVERY

These investigations provided basic data which enabled the shear recovery characteristics to be evaluated with the belting flight times through the respective drives. It was this aspect which was seen to be particularly significant bearing in mind the nature of failure, ie failure at the boundary between the solid PVC and the PVC impregnated fabric carcase.

It was now possible to estimate the residual stresses in the belting as it progressed through the drives. Referring to Figure 2, the Welbeck drive experienced a high tension bend with a flight time of 0.55 s at T_1. During this flight time, the belting would recover approximately 51%. The belting then traversed the drive drum for 0.96 s which should have permitted the shear stresses to decay to approximately 58%. It then experienced a flight time of 0.35 s before going into reverse bending around the first T_3 snub pulley. Bearing in mind that potentially 40% of the initial shear stresses were still locked in, the pulley side cover was being subjected to large accumulated strains. The shortest length of belting in the Gascoigne Wood system had a flight time of 0.29 s compared to 0.12 s at Welbeck. In this instance, the Welbeck belting would still be subject to 63% of the strains induced.

6 SITE TESTS

A number of measurements were taken underground at Welbeck colliery to establish actual belt speeds at various points in the conveying system, and drive power at empty, partial and full load conditions. The belt speeds measured at a nominal loading of 1000 tonne/hr are shown in Figure 4.

The velocity V_2 was measured 9 metres from the T_2 side of the drive. The motor currents were simultaneously monitored during these tests and averaged 26 amps and 31 amps for the primary and secondary drive motors respectively. These values were used to calculate the total effective input power to the system as:

$$P_e = \frac{(31+26)\ x\ 6600\ x\ 0.8\ x\ \sqrt{3}}{1000} = 521\ kW$$

$$T_e = \frac{P_e\ x\ \eta}{V} = 172\ kN$$

where η = motor, coupling and gearbox efficiency (assumed to be 90%)

V = nominal belt speed (2.7 m/s)

0.8 = power factor

The belt tension T_2 was calculated from the values recorded on the load cell incorporated into the loop take-up winch to be 94 kN.

The conveyor was unloaded and permitted to run in order that the belting could release stored stresses. The belting speeds were then measured at the T_1 and T_2 sides of the drive and found to be 2.66 m/s and 2.61 m/s respectively. The motor currents were simultaneously monitored and averaged 19 amps and 20 amps for the primary and secondary drive motors respectively. These values were used to calculate the total effective input power for the unloaded system as:

$$P_e = \frac{(19 + 20)\ x\ 0.8\ x\ 6600\ \sqrt{3}}{1000} = 357\ kW$$

$$T_e = \frac{P_e\ x\ \eta}{v} = 119\ kN$$

where η = motor, coupling and gearbox efficiency (assumed to be 90%)
v = nominal belt speed (2.7 m/s)

It was interesting to note that the difference in input power between loaded and unloaded conditions was 164 kW. Five stop/start tests were conducted during a 10 minute period after the belt had been unloaded. Each time the conveyor was started, the loop take-up winch initially paid out 450 mm and subsequently 300 mm until a total of 1.5 m had been released. On the fifth test no more belt was paid out. This test indicated that the belt was slowly tightening up over a 10 minute period, it was contracting from a stretched state. The length of belt put back was equivalent to 0.11% of the total length of belting.

The loop take-up controller was then disconnected. An estimated 1000 t/hr was then fed onto the belt and changes in motor current and T_2 belting tension recorded until the parameters had stabilised. The results are shown in Table 6.

TIME	PRIMARY MOTOR		SECONDARY MOTOR		T_2	% LOADED
Minutes	Amps	kW	Amps	kW	kN	
0	19	183	20	192	31	0
1.25	20	192	22	212	29	15
2.50	22	212	23	221	27	30
3.75	23	221	25	240	26	45
5.00	25	240	28	269	24	60
6.25	27	260	31	298	22	75
7.50	32	308	37	356	20	90
8.75	38	365	42	404	18	100
18.00	26	250	31	298	31	100
20.00	26	250	31	298	31	100

TABLE 6: EFFECT OF BELT LOAD WITH RESPECT TO BELT STRETCH

As expected, the conveyor replicated the same pattern of belt stretch for the first 10 minutes as the load was increased to its maximum. After 8.75 minutes had elapsed, T_2 dropped to a point of slip to 18 kN whilst the combined motor currents rose to 80 amps. This was equivalent to a belt line power of 732 kW giving an effective belt tension for T_e of 244 kN. The modelled values for T_e were 295 kN.

However, thereafter the power required to drive the conveyor reduced over a period of 10 minutes whilst the value for T_2 rose back to 31 kN. It must be remembered that the loop take-up was not functioning at this time. Therefore, these results indicated that the belting was contracting as the duty imposed increased. This could be the result of such contraction taking place as a consequence of rises in belting temperature as it worked or, alternatively, the belting was recovering as it progressed down the return line of the conveyor and thereby increasing the strain in the T_2 area of the conveyor.

7 ANALYSIS OF RESULTS

7.1 Analysis of Drive Speeds and Powers

Figure 1, previously referred to, shows the Welbeck drive configuration. The relationship between the respective belting speeds were:

$$\frac{V_a}{V_3} = \frac{r_a}{(r_a + t)} \qquad (1)$$

where V_3 was the measured belting cover speed at the secondary snub (2.799 m/s), r_a was the pulley radius (0.4 m) and t the belt thickness (0.0225 m).

From equation (1):

$$V_a = \frac{0.4}{(0.44 + 0.0225)} \; x \; (2.799) \; m/s$$

Using the relation between the secondary drive speeds as a ratio:

$$\frac{(V_s - V_a)}{(V_3 - V_a)} = R \qquad (2)$$

and

$$V_s = 2.65 + (0.1491 \; x \; R) \qquad (3)$$

where R was the degree of recovery from shear induced by bending around the pulleys, and the value R = 1 represents no recovery and R = 0 represents 100 % recovery.

From the measured value for the secondary drive motor full load current I_s of 52 amps at 1480 rpm, and the synchronous speed current at 1500 rpm of 7.82 amps, the speed of the motor during the test with a current of 30 amps was calculated to be :

$$V_a = 1500 - \frac{(30 - 7.82)}{2.209} \; rpm$$

which, allowing for 1 % slip at the coupling, gave an input speed to the drive drum of:

$$0.99 \; x \; \frac{1489.96}{35.57} = 41.4692 \; rpm$$

Therefore, for a 1.25 m drum diameter, the belt speed V_s was 2.7141 m/s, which could be substituted into equation (3) to give:

$$R = 0.4302$$

This indicated that the belting would only recover approximately 43 % from the stresses induced from bending around the snub drum before reverse bending onto the secondary drive drum. This calculated value of R was close to the predictions based on measured visco-elastic decay of the belt sample in Table 5. This process identified a retention of strain that was not relaxed before reverse bending, thereby creating a mechanism of high additional strains on the carry and pulley covers.

The same procedure was used to determine the primary drive belt speeds. These were determined to be 2.7178 m/s (measured) for the pulley cover entering the drive and 2.7667 m/s for the carry cover on exit. The difference in cover speeds could be expressed as a flight time lag of the pulley cover in the order of:

$$\frac{2.7667 - 2.7178}{0.316} = 2.055 \; ms$$

7.2 Differential Elongation of Covers

Based on the mechanical configurations of the Gascoigne Wood and Welbeck drives and a belt thickness of 22.5 mm, the differential elongation of the belting covers were calculated to be:

Between	GASCOIGNE WOOD	WELBECK	COVER
Primary Drive	84.43	89.9	CARRY
Primary Snub	84.43	82.46	PULLEY

Between			
Secondary Snub	80.50	90.32	PULLEY
Secondary Drive	82.47	89.93	CARRY

These values were obtained by treating the respective covers completely independently from each other, ie no interaction. In terms of stored shear strain, the amount of strain due to primary pulley bending would only decay by approximately 43 % between leaving the primary drive pulley and entering the first snub pulley. By analysing the drive in this way the locked-in cover elongations could be determined at various locations around the drive as shown in Tables 7 and 8.

LOCATION	PULLEY COVER ELONGATION mm	CARRY COVER ELONGATION mm	DIFFERENTIAL CHANGE
Exiting Primary Drive	0	84.43	+ 84.43
Entering Primary Snub	0	0.58 x 84.43 = 48.97	+ 48.97
Exiting Primary Snub	48.97 + 84.43 =133.40	48.97	- 84.43
Entering Second Snub	0.40 x 133.4 = 53.36	0.40 x 48.97 = 19.59	- 33.77
Exiting Second Snub	53.36 + 80.50 = 133.86	19.59	- 114.27
Entering Second Drive	133.86 x 0.58 = 77.48	19.59 * 0.58 =11.36	- 66.12
Exiting Second Drive	77.48	11.36 + 82.47 = 93.83	+ 16.35

TABLE 7: PROGRESSIVE DIFFERENTIAL COVER ELONGATIONS - GASCOIGNE WOOD

LOCATION	PULLEY COVER ELONGATION mm	CARRY COVER ELONGATION mm	DIFFERENTIAL CHANGE
Exiting Primary Drive	0	89.9	+ 89.9
Entering Primary Snub	0	0.57 * 89.9 = 51.24	+51.24
Exiting Primary Snub	51.24 + 82.46 =133.70	51.24	- 82.46
Entering Second Snub	0.366 * 133.7 = 48.93	0.366 * 51.24 = 18.75	- 30.18
Exiting Second Snub	48.93 + 90.32 = 139.25	18.75	- 120.50
Entering Second Drive	139.25 * 0.634 = 88.28	18.75 * 0.634 =11.89	- 76.39
Exiting Second Drive	88.28	11.89 + 89.93 = 101.82	+13.54

TABLE 8: PROGRESSIVE DIFFERENTIAL COVER ELONGATIONS - WELBECK

The differential elongations between the two covers for the two conveying systems are shown in graphical form in Figure 5 and Figure 6 with respect to "time" and "distance travelled" respectively. As can be seen, the differential elongations ranged widely as the belting passed through the drives. However, the two systems are remarkably similar.

The most significant differential elongation was experienced on the Welbeck drive as the belting came onto the secondary drive drum. It must be noted that the real stress levels experienced would be dependant upon the combined axial and bending forces imposed and the degree of interaction across the belt thickness. Even so, it was recognised that the bending stress in the solid PVC could be significantly higher than in the impregnated fabric.

Therefore any bending induced stress would be expected to be distributed across the belting thickness at a varying gradient. The nitrile covers are comparatively elastic and consequently the overall composite effect of bending and relaxation was better understood by referring back to the shear test work.

8 CONCLUSIONS

The source of belting deterioration was the result of two factors: manufacturing problems associated with the belting and the configuration of the drive.

8.1 Belting

The problem of cover delamination revealed a need to develop a better adhesion system at the boundary of the nitrile rubber cover and the solid PVC cover of the composite construction.

Having resolved the cover interface problem, the problem of delamination was then transferred to the next weakest region, namely the solid PVC to PVC impregnated carcase. These failures were internal to the belting and only became apparent as large areas of cover bubbled outwards.

Any changes in the construction of the belting will be compounded by the need to use low stretch materials incorporating high strength.

The belting must be designed to maximise the rate of shear recovery.

The resultant thickness of belting is an important factor as this will determine the levels of differential elongation as the belting negotiates the drive.

8.2 Drive Configuration

The drive systems at Gascoigne Wood and Welbeck visually appear to be different, with the increased distances between respective drums at Gascoigne Wood indicating that this configuration would treat the belting more kindly. For example, the snub pulleys on the Welbeck drive are much closer than those at Gascoigne Wood. Consequently, there is insufficient time available as the belting passes from the primary drive drum to the primary snub pulley and from the secondary snub pulley and secondary drive drum.

However, the widely different belt speeds, 4 m/s and 2.6 m/s at Gascoigne Wood and Welbeck respectively, result in very similar belting stress patterns, with the sole exception for the Secondary Snub/Drive drums. In this instance the Welbeck drive is by far the worst, as suspected in the early stages of the investigation. Therefore, it is considered that the Gascoigne Wood drive is on the borderline with regard to successful application of the Type 18 belting.

The mechanical requirements of a drive are, in part, dependent upon the properties of the belting. To resolve the problem at Welbeck Colliery, it would be necessary to increase the distances between the snub and drive pulleys. In this instance this was not considered to be a feasible solution as it would locate the snub pulleys outside the existing drive modules.

9 RECOMMENDATIONS

For the future, conveyor belting and drive manufacturers must work in close collaboration for high powered conveyors incorporating solid woven core belting to be applied successfully. The minimum distance between snub pulleys and their respective drive drums must be considered in conjunction with the shear recovery characteristics of the specified belting. The wrap angle of the drive drums could be reduced, and in so doing reduce the bending induced shear on the covers and the carcase.

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27 March 1995

REFERENCES

1 CTL Report No 1060 of Conveyor Technologies Ltd: A Harrison 1060\1994

2 CTL Report No 1069 of Conveyor Technologies Ltd: A Harrison 1069\1995

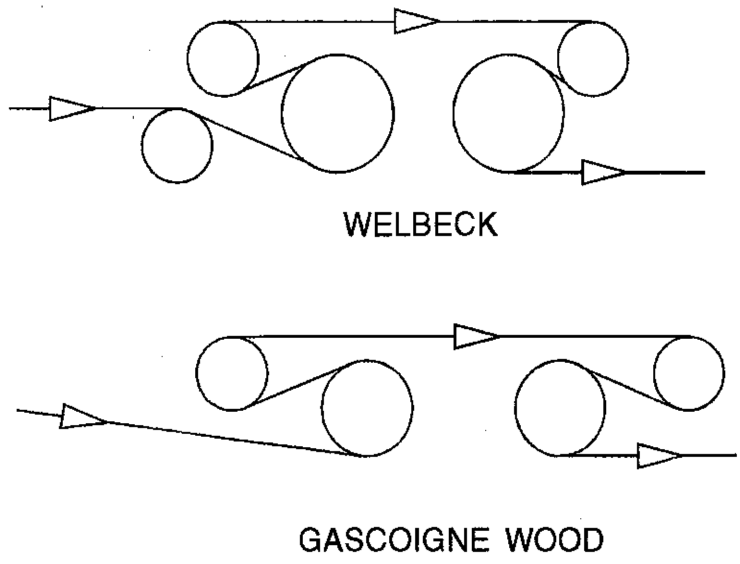

FIGURE 1: DRIVE CONFIGURATIONS

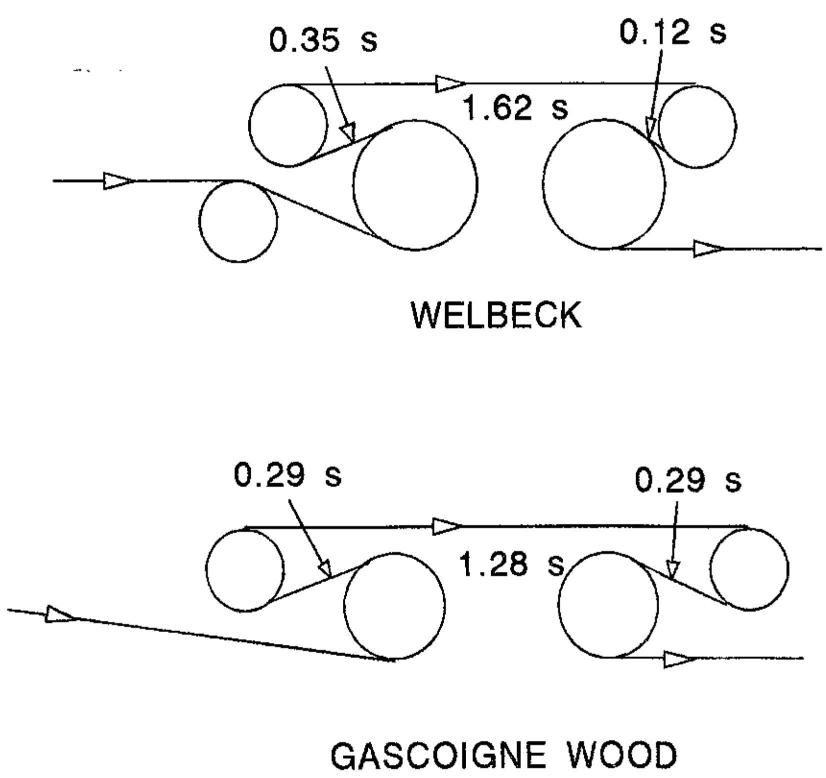

FIGURE 2: FLIGHT TIMES FOR DRIVE CONFIGURATIONS

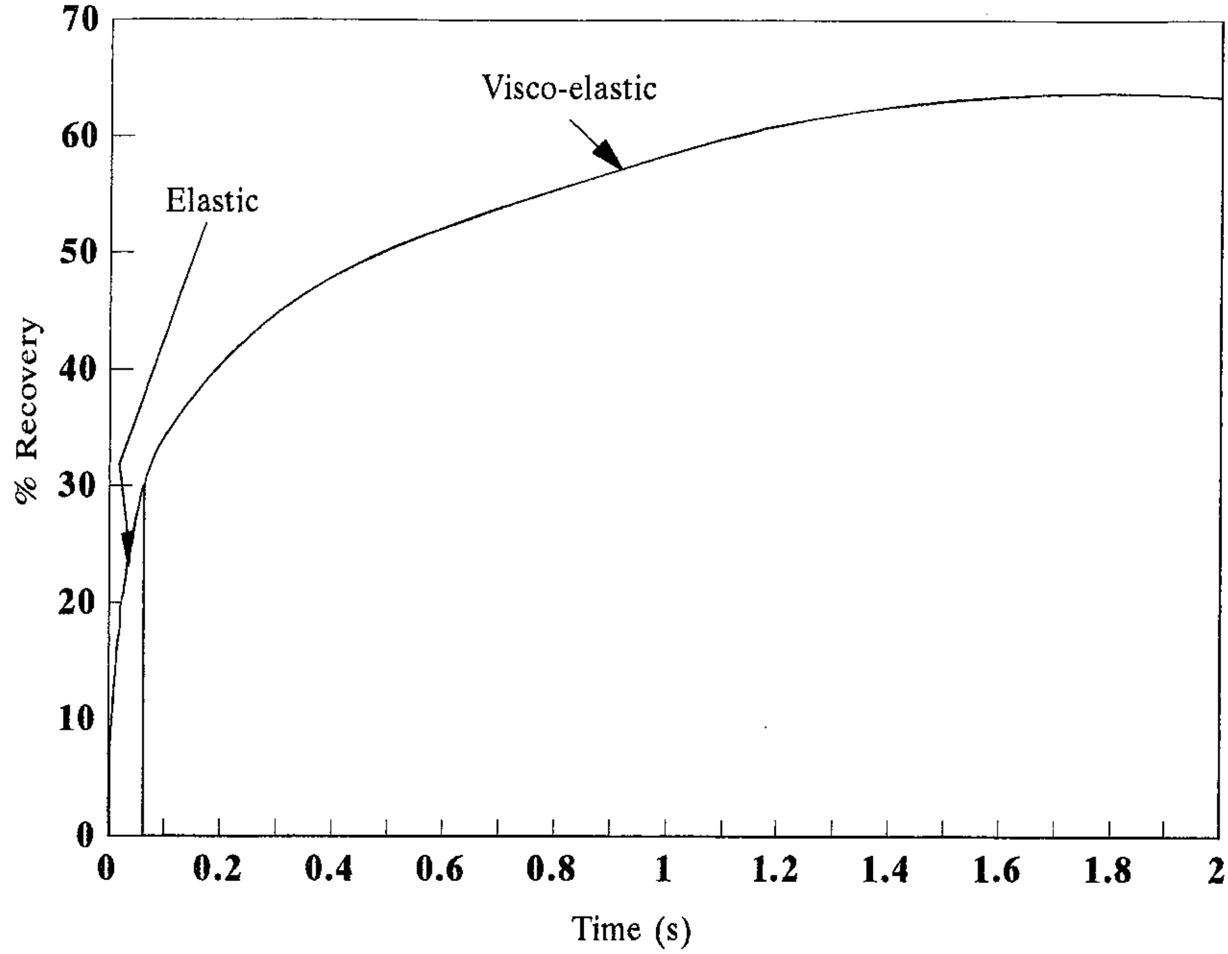

FIGURE 3: SHEAR RECOVERY

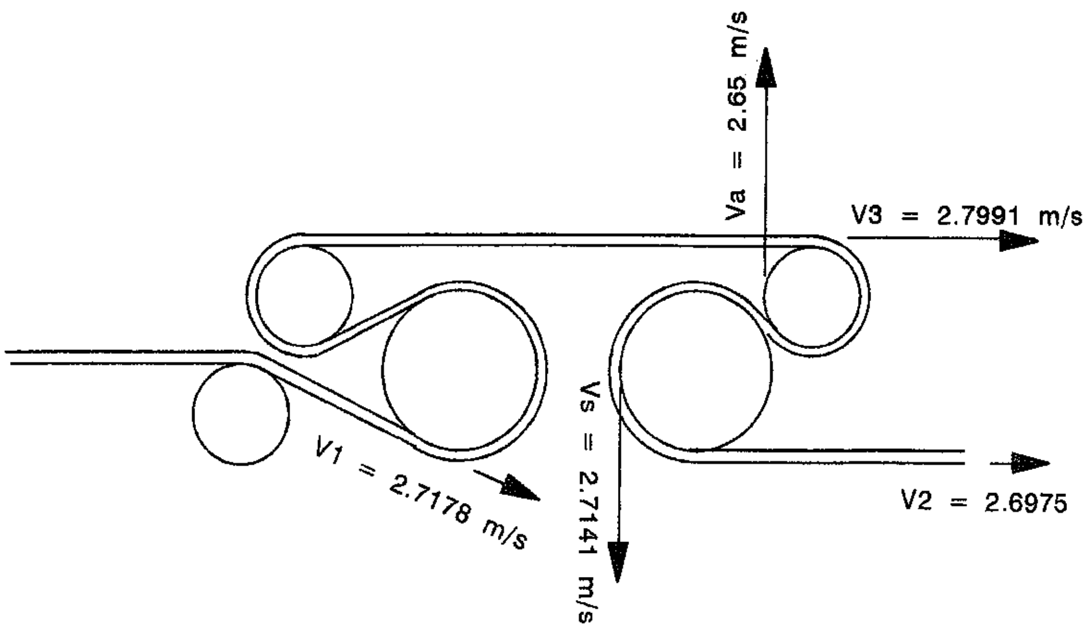

FIGURE 4: BELT SPEEDS THROUGH WELBECK DRIVE

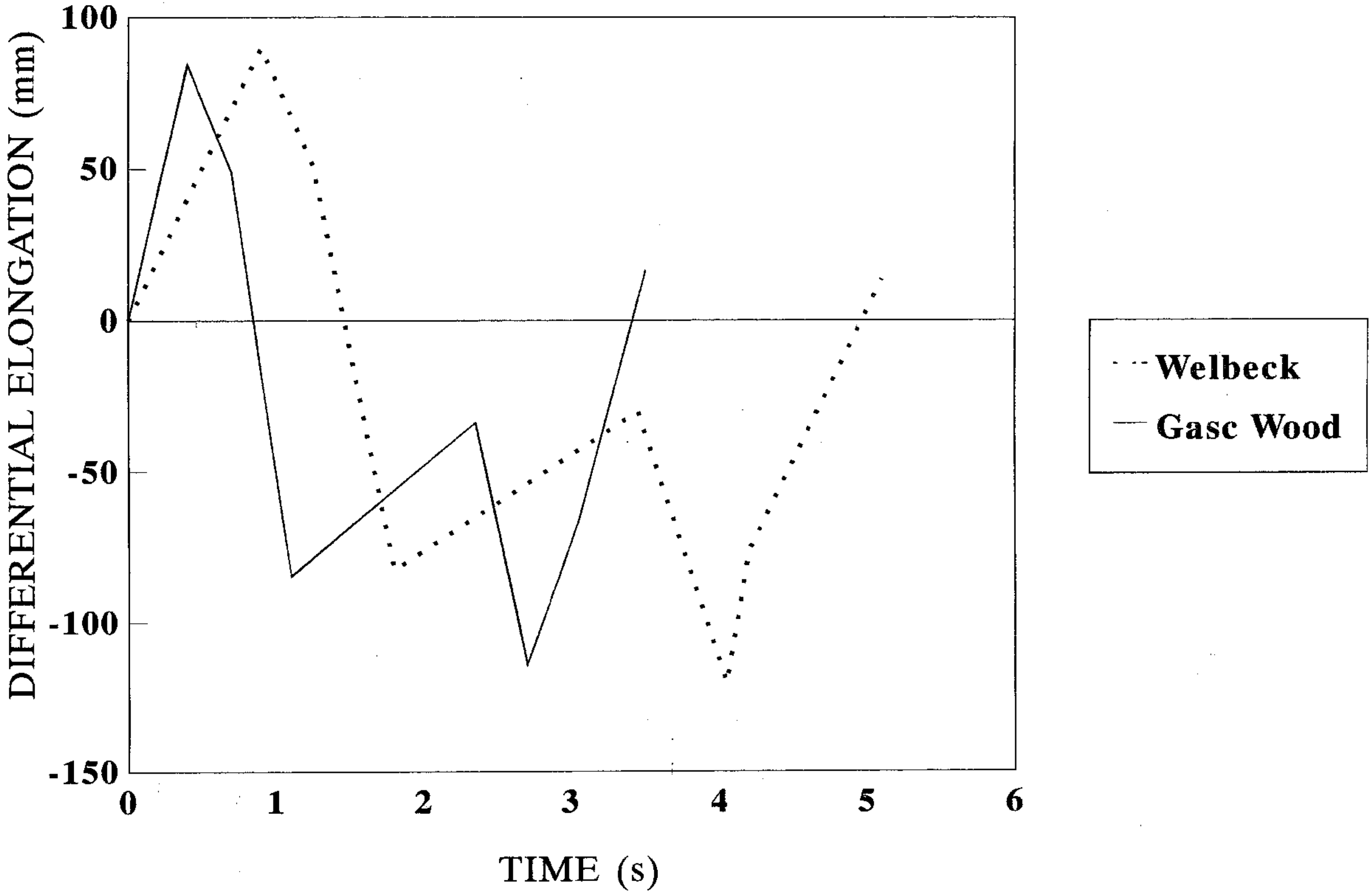

FIGURE 5: DIFFERENTIAL ELONGATION WRT TIME

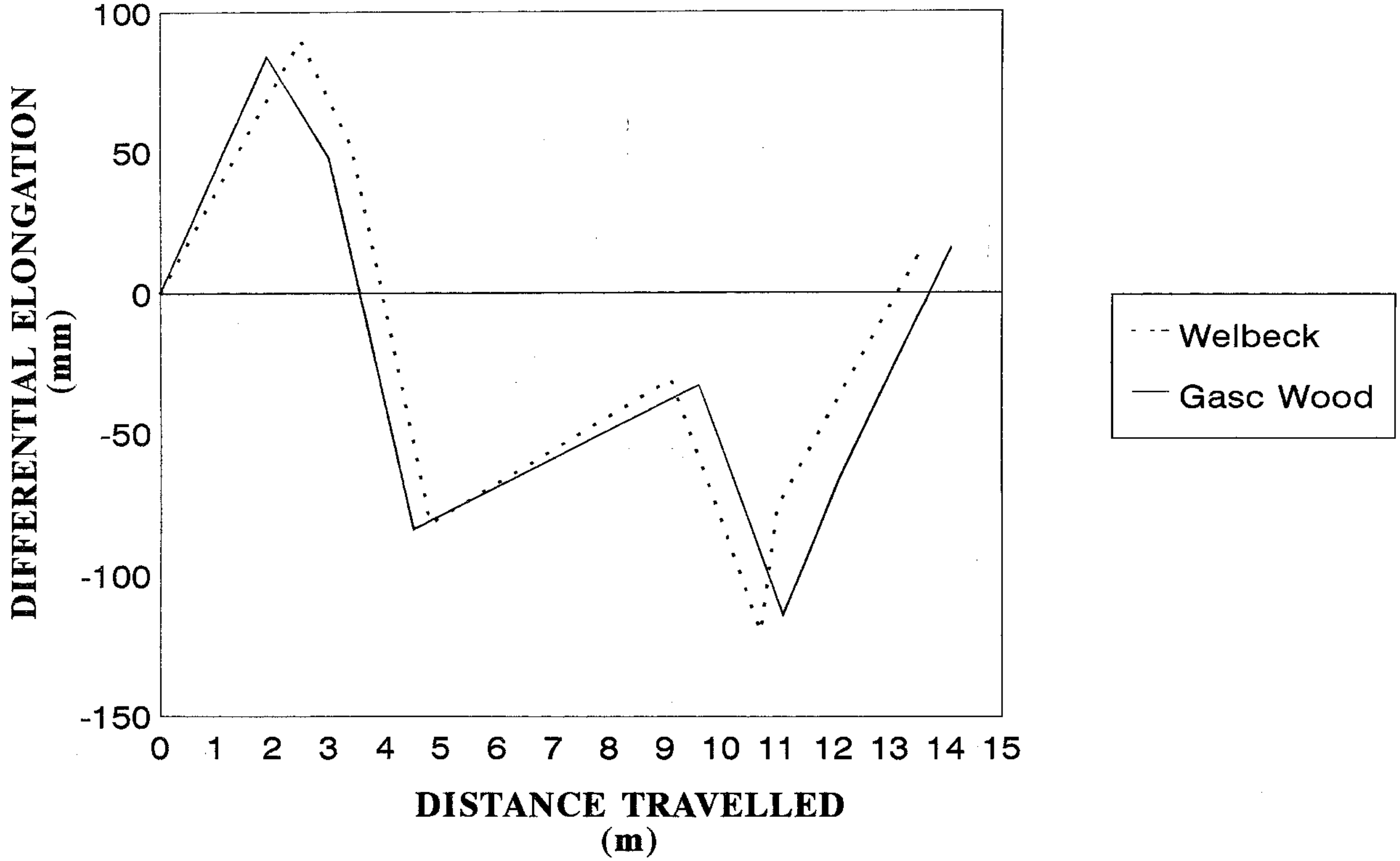

FIGURE 6: DIFFERENTIAL ELONGATION WRT DISTANCE