A STUDY OF LOAD SUPPORT AND OTHER CRITERIA APPROPRIATE TO THE SELECTION OF INDUSTRIAL CONVEYOR BELTS

A thesis submitted in November 1976, by Niall Gordon Ramsden, to Aston University for the degree of Doctor of Philosophy.

SUMMARY

A study of conveying practice demonstrates that belt conveyors provide a versatile and much-used method of transporting bulk materials, but a review of belting manufacturers' design procedures shows that belt design and selection rules are often based on experience with all-cotton belts no longer in common use, and are not completely relevant to modern synthetic constructions. In particular, provision of the property "load support", which was not critical with cotton belts, is shown to determine the outcome of most belt selection exercises and lead to gross overspecification of other design properties in many cases. The results of an original experimental investigation into this property, carried out to determine the belt and conveyor parameters that affect it, show the major role that belt stiffness plays in its provision; the basis for a belt stiffness test relevant to service conditions is given.

A proposal for a more rational method of specifying load support data results from the work, but correlation of the test results with service performance is necessary before the absolute load support capability required from a belt for given working conditions can be quantified. A study to attain this correlation is the major proposal for future work resulting from the present investigation, but a full review of the literature on conveyor design and a study of present practice within the belting industry demonstrates other, less critical, factors that could profitably be investigated. It

is suggested that the most suitable method of studying these would be a rational data collection system to provide information on various facets of belt service behaviour; a basis for such a system is proposed.

In addition to the work above, proposals for simplifying the present belt selection methods are made and a strain transducer suitable for use in future experimental investigations is developed.

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N. G. RAMSDEN

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GLOSSARY OF CONVEYOR TERMS AND SYMBOLS USED IN THIS THESIS

At many points throughout the thesis terms peculiar to belt conveyor technology are used; for convenience, a glossary of these is presented here.

(a) Terminology related to the belt conveyor structure

Conveyor Centres Length: the distance between the two ends of the conveyor.

Conveyor Lift and Fall: these terms refer, respectively, to the vertical distance through which the conveyor lifts or lowers the load.

Drive Pulley/Drum: the driven pulley which transmits motion to the belt.

Snub Pulley: the pulley used to increase the arc of contact of the belt on the drive pulley.

Angle of Wrap: the angle subtended by the belt around the drive pulley.

Lagging: a covering, often of rubber, for the drive pulley to increase the co-efficient of friction between belt and pulley.

<u>Head End</u>:	the discharge end of the conveyor. For most conveyors the drive pulley is positioned at the head end.
<u>Tail End</u>:	the loading end of the conveyor.
<u>Idlers</u>:	rollers on which the conveyor belt runs. Normally three rollers in a line transverse to the belt are used.
<u>Wing/Side Rollers</u>:	the outer rollers of a set of idlers.
<u>Idler Angle</u>:	the angle at which the wing rollers of an idler set are inclined to the centre roller. This is done to make the belt assume a <u>troughed</u> shape and so increase the carrying capacity of the conveyor. If the idler angle is greater than or equal to 45°, the conveyor is said to be a <u>deep-troughing</u> installation.
<u>Idler Pitch</u>:	the distance between adjacent idler sets.
<u>Idler Gap</u>:	the small gap between the centre roller and the wing roller of an idler set. This is commonly ten millimetres.

<u>Slider Bed</u>: an alternative to idlers for support of a loaded belt. A slider bed is a smooth continuous wooden or steel table over which the belt is pulled.

<u>Skirt Board/Plate</u>: a device positioned at each side along a length of conveyor to prevent spillage of the load. Normally these are only present at the belt loading point.

<u>Take-Up</u>: a device for taking up slack and applying tension to the belt. It can be of either <u>screw</u> or <u>gravity</u> form. A screw take-up consists of a pulley, usually at the tail end of the conveyor, which can be moved backwards or forwards along tracks to vary the belt tension. A gravity take-up consists of a pulley around which the belt is looped. The pulley is free to move in a vertical direction and is loaded so as to provide the required belt tension.

<u>Transfer Points</u>: positions at which the conveyed load is transferred from one belt to another.

Tripper/Stacker: a device, either fixed or mounted on a moving carriage, which causes the conveyed material to be unloaded at a point along the length of the conveyor. If the tripper is mounted on a moving carriage it is said to be a travelling tripper and can discharge the load at any point along the conveyor.

(b) Terminology related to the belt

Carcase: the load bearing members of the belt. In modern belt constructions this normally consists of several plies of synthetic fabric separated by rubber interply. Alternative forms of reinforcement are

(i) Solid Woven which consists of a fabric woven as a single heavy web and

(ii) Steel-cord in which longitudinal steel cords form the carcase.

Warp: the longitudinal members of the belt reinforcement material.

Weft: the transverse members of the reinforcement material.

<u>Top Cover</u>: layer of elastomeric material <u>(compound)</u> to protect the carcase on the side of the belt that will carry the load.

<u>Bottom Cover</u>: protective layer, usually thinner than the top cover, on the side of the belt that will not carry the load.

<u>Interply</u>: the rubber compound present between adjacent fabric plies.

<u>Breaker</u>: fabric ply or plies under the top cover of the belt to prevent gouging and stripping of the top cover and to protect the carcase from impact.

(c) <u>Terminology related to the working conveyor</u>

<u>Loaded/Tight Strand</u>: the loaded belt running from the loading point to the discharge points.

<u>Empty/Return/Slack Strand</u>: the empty belt running from the discharge point to the loading point.

<u>Effective Tension</u>: the difference between the tensions in the tight and slack strands at the drive pulley.

Mechanical Fasteners:	devices for making a conveyor belt endless by joining the two ends together with metal staples or plates.
Vulcanised Splice:	an alternative and more efficient method of making a belt endless to mechanical fasteners. In this the belt ends are joined using only the materials already present in the belt construction.
Edge-Margin:	the unloaded strip of belt at each side of the material being carried.
Troughability:	the ability of a belt to form a troughed shape.
Load Support:	the ability to support the load without spillage on damage to the belt or conveyor structure.
Inter-Idler Sag:	the belt displacement below a straight line running between the top of the centre rollers of adjacent idler sets.
Safety Factor:	the ratio of belt tensile strength to maximum working tension.

(d) Symbols in common use throughout the thesis

Symbol		Units (unless stated otherwise)
T_1	The tension in the loaded belt strand at the drive drum.	kN/m
T_2	The tension in the return belt strand at the drive drum.	kN/m
T_m	The maximum tension occurring in the working belt. Often T_m is equal to T_1, depending on the conveyor configuration.	kN/m
T_e	The effective tension (see (c) above)	kN/m
K	Drive factor (see Section 3.3)	
θ	Angle of Wrap	o
μ	Coefficient of friction between belt and drive drum.	
b	Belt width	mm
t	Belt thickness	mm
L	Conveyor centres length	m

L_A	Length adjustment used in calculation of belt tension (see Section 3.3).	m
L_1	$L + L_A$	m
H	Conveyor lift or fall.	m
I	Idler pitch.	m
β	Idler angle.	o
S	Inter-idler belt sag.	m or mm
T	Belt capacity.	tonnes/hr
A	Cross-sectional load area.	m^2
V	Belt speed.	m/s
P	Weight of moving parts (see Section 3.3).	kg/m
B	Belt weight.	kg/m
M	Load weight $M = \frac{T}{V} \times 0.278$	kg/m
W	Total weight on belt ($W = B + M$)	kg/m

F_B, F_L Friction factors used in belt tension calculations as described in Section 3.3

INTRODUCTION

The Annual Abstract of Statistics, 1975, compiled by the Department of Industry shows that, in 1974, 107.5 million tons of coal, 3.6 million tons of iron ore and 348.4 million tons of minerals were mined in Britain. The gross output of the mining and quarrying industry in 1973 (the most recent year for which figures are available) was valued at £1,452 million. It has been said [1] that the cost of transportation can constitute up to 50% of the cost of winning a mineral and so the operating efficiency of the mining industry is very largely dependent on the efficiency of its material handling equipment. In fact, some industries would not be economically viable without efficient mechanical handling.

As conveying technology obviously represents a major factor in the production and economy of industry, it is important to ensure that the most suitable form of conveying machine is chosen for a particular task. In Chapter 1 a short review demonstrates that several different types are available and that the belt conveyor is often the final choice for a wide range of applications. It therefore follows that it is important to be able to find the optimum belt for a given conveying situation. Such a belt should be capable of withstanding the damaging factors that it will meet in service without overspecification of any design properties. The present thesis describes a study of the procedures by which a suitable belt for a particular task is selected and investigates how close the resulting belt constructions are to the optimum situation; the object of this study being to improve the selection techniques where necessary. The original terms of reference for the project were very wide so that a substantial part of the project was to look at the whole problem and logically identify and quantify the areas of most concern.

Much of modern conveyor belting design practice is based purely on the experience gained during the period when conveyor technology was in its infancy and installations were relatively short and lightly loaded. The extrapolation of design rules to present day all-synthetic material belts has caused some problems; evidence from the development history of belting, the scientific literature and the study of belt selection procedures discussed in the early chapters of the thesis suggested the major one was the adequate provision of a property called load support (defined in Chapter 1) because this could lead to over-specification of other design properties. Analyses of belts recommended for various conveyor installations confirmed this (Chapter 6) and demonstrated the need, on both technical and commercial grounds, for an experimental investigation of this property with the objective of providing a more rational approach to its assessment for a given conveyor situation. Such an investigation, described in Chapters 7 and 8, was therefore devised.

The work described in this thesis was sponsored by the Dunlop Angus Belting Group (see Appendix 1) and was carried out in an industrial manufacturing environment. It was therefore in competition with other projects being carried out by the sponsoring organisation for allocation of capital, manpower and machine time, so that each stage of the project had to be justified in terms of the possible benefits that it could provide. The project did, in fact, provide a useful contribution to the knowledge of belt selection techniques and it highlighted a number of areas which could profitably be examined in further investigations. In particular, it demonstrated the importance and provided a better understanding of the mechanism of load support, thus giving a basis for evaluation of the most economic methods by which this property can be achieved for any given combination of belt and conveyor structure.

CHAPTER 1

CONVEYING PRACTICE AND SOME ASSOCIATED PROBLEMS

The importance of conveying to the productivity of industry has been shown in the introduction above. It is obvious that to optimise the use of mechanical handling the conveyor design engineer must choose the most efficient conveying technique for the task being considered. This chapter discusses some of the factors that affect this choice and shows how, because of its versatility, the belt conveyor is very often the handling method finally used.

1.1 Selection Factors for Conveying Method

There are two levels of conveying: (a) in-plant handling such as that used in production lines and (b) bulk conveying used to transport materials from place to place. The work described in the present thesis is almost entirely associated with the second category, but with both there are seven basic questions that must be answered by the design engineer before choosing the conveying technique to be used:-

(1) What is the product or material to be handled?

(2) What are its physical characteristics? (Size, shape etc.,)

(3) What rates of flow are involved?

(4) What is the conveyor to accomplish? (Transportation, assembly, sorting etc.,)

(5) What expenditure is justified and how much money is available?

(6) Is the product fragile, or is noise, contamination or chemical reactivity a problem?

(7) Are there any restrictive factors to be considered such as available space or environmental or atmospheric conditions.

Having answered these questions it is then possible to study the advantages and disadvantages of the conveying systems available and select the type which best fits the requirements. For in-plant materials handling there are a great many different forms of conveyor available - one of which is, of course, the belt conveyor. The various types have been reviewed by Etheridge [2] and Brennan [3, 4]. A summary, presented by Etheridge, of their application is shown in Figure 1.1.1 and from this it can be seen that belt conveyors, although not the "best" for many applications, are very versatile in

CONVEYOR APPLICATION GUIDE CHART 1 = FAIR 2 = GOOD 3 = BEST	Transportation	Assembly	Processs	Accumulation	Sorting	Warehousing	Shipping Terminal	In-process Storage	Dispatching	Order Packing	Storage Handling	Live Storage	Horizontal Flexi-bility	Vertical Flexibil-ity
Hand-pushed monorail		1	1	1				1			1		2	
Powered travel monorail	2		1					1	1				2	
Overhead conveyor	2	2	2		1			1	2			2	3	3
Tow Conveyor	3	1	1	2		3	3	2	2	2		1	3	1
Roller Conveyor	1	1	1	2		1	1	2		1		2	1	
Live Roller Conveyor	2			1		2	1	3	1	2		1	2	
BELT CONVEYCR	3	2	1		1	2	2	2	1	1		1		2
Single strand chain on edge	3	3	3										2	2
Slat conveyor	3	3	1											1
Roller flight conveyor	1			3								2		
Sorter conveyor					3	2	3	2	3				1	1
Driverless vehicles	3					3	1	1	3	2			2	

FIGURE 1.1.1

Conveyor Application Guide Chart by Etheridge for In-plant Materials Handling that they will prove to be satisfactory for several different tasks.

Belt conveyors really become useful when dealing with movement of bulk materials and, as required transportation distances tend to become longer and longer, they very often represent the most suitable (and sometimes the only suitable) method of mechanical handling.

The basic classes of conveyors for bulk handling are, according to Buffington [5] :-

(a) Screw conveyors.

(b) Drag conveyors.

(c) Pan conveyors.

(d) En masse conveyors.

(e) Vibrating conveyors.

(f) Bucket conveyors.

(g) Belt conveyors.

One system not described in the review mentioned is that of (h) the Pneumatic conveyor.

A short description of each of these conveyor types and their advantages and disadvantages is given overleaf.

(a) Screw Conveyors

These consist of a steel helix mounted on a powered spindle. The moving parts are enclosed in some suitable casing so that any material fed into the device will be pushed along by the rotating helix. (Archimedes is reputed to have discovered this mechanism!)

The screw conveyor is a versatile and economical method of transporting many materials, although it is best suited to coarse materials such as clinker and coal [6]. It can operate at any angle of incline, has few moving parts and has a fairly low initial cost. It is ideal for handling systems requiring controlled continuous material flow and can easily be adapted to allow for any environmental conditions such as the need for vapour proofing. It is easy to clean and no load spillage can occur.

The capacity of the screw conveyor depends on the type of helix, its speed of revolution and the cross-sectional characteristics of the material. Free flowing materials will fill forty-five percent of the helical section whereas coarser, more abrasive materials will only fill fifteen percent [7]. Thus the screw conveyor tends to have a fairly low capacity and is relatively inefficient in terms of power used per tonne of material carried. (A typical conveying capacity is 30 m^3/h for a standard 305 mm diameter helix turning at 80 rpm.) The screw conveyor therefore tends not to be used when transportation distances are large, and, its length is typically limited to around fifteen metres. Abrasion occurs as the material is pushed over the casing of the helix, and this is one of the major faults of the screw conveyor. The problem can be overcome using special construction materials, but this, of course, increases the cost of the installation.

(b) Drag Conveyors

The drag conveyor consists of special wide chains which are dragged through a hard trough of iron, steel or concrete, thus pulling the load along with them. It is used for materials similar to those handled with screw conveyors and it tends to suffer from the same disadvantages of inefficient power-use and abrasion of the equipment by the load.

(c) Pan Conveyors

A pan conveyor consists of large metal pans pulled by chains at either side of the structure and is best used for handling very hot and heavy loads. Conveying is not continuous but, for additional cost, systems can be designed which allow "stock-piling" of the pans en-route and thus effectively give a continuous flow of material arriving at the destination. The pans are usually made self-levelling so that vertical slopes can be traversed without spillage. The pan conveyor is expensive to install and maintain, so it tends only to be used for applications that demand the installation to be resistant to extreme conditions. (For example, one use of the pan conveyor is the transportation of molten glass.)

(d) En-masse Conveyors

The en-masse or continuous-flow conveyor transports material as a single mass inside a tube. This is achieved by a chain moving through the tube. At regular intervals along the chain,plates which almost fill the cross-section of the tube are attached to push the material along.

The en-masse conveyor is ideal if there are several loading and discharge points required in the system. The unit can be sealed to prevent contamination and very little maintenance is required. Motion in more than one plane is possible and any angle of inclination can be included in the system. Less space than that needed for belts and screws is required for this type of conveyor.

The main disadvantage of the en-masse conveyor is its very poor efficiency due to the great deal of power required to overcome the high frictional forces occurring as the material and plates move against the side of the enclosing tube. Also, only fairly low tonnage rates can be handled.

(e) Vibratory Conveyors

The principle of the vibratory conveyor is that when a plate is subjected to vibrations in a certain plane, granular or lumpy material upon it will be thrown forwards repeatedly, more or less in synchronism with the cycle of the pulsations [4]. This can result in an efficient, spillage-free and controllable transportation system for powders or granular materials. The machine has a long life expectancy because no working parts are in contact with the moving materials [8] and an installation can be entirely enclosed if dust prevention is required. A vibratory conveyor can be used for distributing material to various destinations by the use of diverting ploughs and several different chemicals can be handled at the same time by the machine if full length dividers are used to separate the carrying trough into a number of channels. Power requirements are low once the machine is in operation, but starting power requirements are fairly high. Initial costs of the equipment are high, and the system is limited to fairly low capacities. (50 tonnes per hour is the average figure, although capacities up to 500 tonnes per hour can be handled.) Vibratory conveyors rarely exceed fifty metres in length.

(f) <u>Bucket Conveyors</u>

The bucket conveyor is usually used where vertical motion is required in limited floor space. Essentially it is an extension of the chain conveyor or the belt conveyor (see below), because the buckets in which material is transported are mounted on a continuous chain or rubber/fabric belt. The conveyor gives smooth and quiet operation with good abrasion and corrosion resistance. Large material lumps can damage the equipment and excellent housekeeping and routine maintenance are required if serious damage to the conveyor structure is to be avoided. (For example, the bolts securing the buckets to the belt must be checked periodically.)

(g) Belt Conveyors

The belt conveyor is the most widely used system for transporting bulk solids. An example of a long belt conveyor installation carrying gravel can be seen in Figure 1.1.2. Liggins [9] describes it as "consisting, in its simplest form, of a tail pulley* and a head pulley* connected to a driving motor, with an endless belt travelling around the two pulleys." (See Figure 1.1.3a.) The belt itself is defined as "a number of load carrying members bonded together with polymeric compounds and protected from mechanical or chemical damage by elastomeric covers*." The load carrying members, commonly either of steel cords or textile plies, are known as the "carcase*" of the belt. The belt is usually used in a troughed form supported by rollers known as idlers* (See Figure 1.1.3b) which are, commonly, arranged equi-distant from each other along the conveyor length.

The belt conveyor is an extremely versatile piece of equipment and can be used to carry a great variety of materials ranging from sawdust to large fragments of granite, at temperatures ranging from sub zero to more than 150°C. A series of articles demonstrating the diversity of loads that can be handled has been published by Schultz [10, 11, 12] and several papers have been written giving the advantages of using belts for particular tasks. (For example, (a) Coal mining [13, 14] (b) Open Pit Mining [15], (c) Concrete placement [16, 17], (d) Transferring bulk material from ship to shore [6] and (e) Road construction [18].)

* See glossary of terms at the beginning of this thesis.

FIGURE 1.1.2

General view of belt conveyor carrying gravel

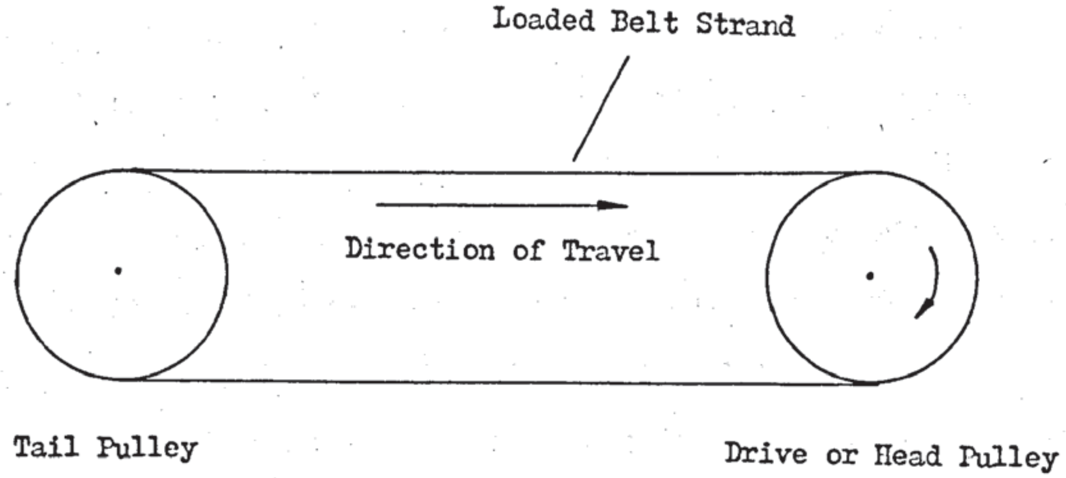


FIGURE 1.1.3a

The fundamentals of a belt conveyor

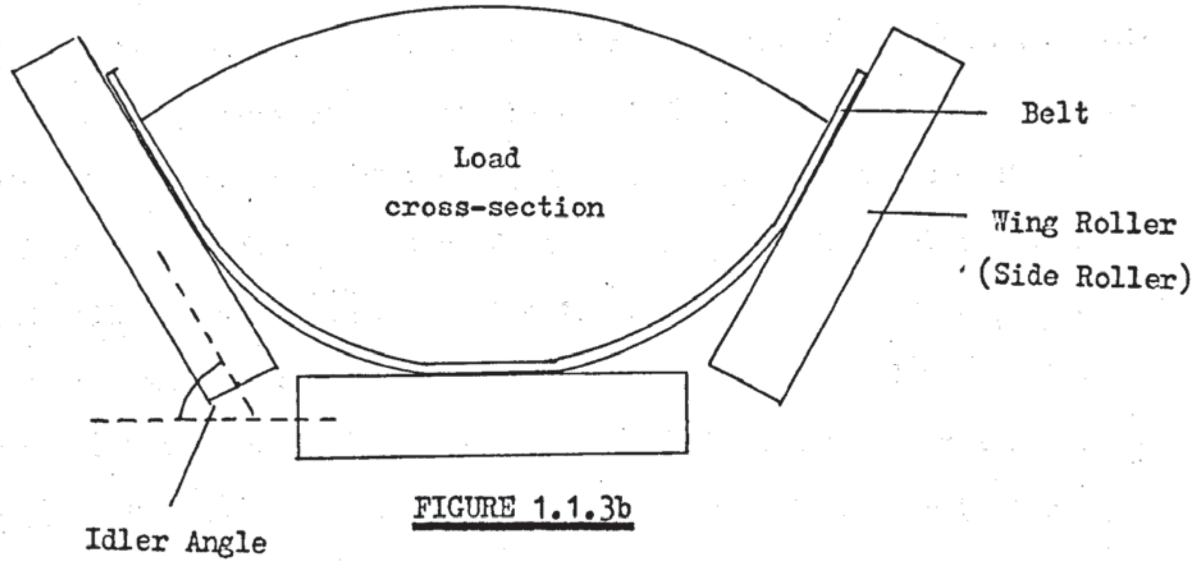


FIGURE 1.1.3b

Cross-section of a three-roll idler set

The wide variety of uses for the belt conveyor stems from the fact it can provide a continuous, economic, reliable and safe method of material transport. It can convey goods relatively quietly without causing any damage to the product being handled because the load does not touch any moving parts. (An experiment was once carried out to demonstrate this [13] :- Two dozen eggs in two standard cartons were placed on a coal laden belt and after travelling over four miles and going through eighteen belt transfer points, only five eggs were cracked in one carton and none at all were cracked in the other.) Capital costs are fairly low and maintenance is very cheap [19]. Power requirements are small once the conveyor is in motion. Capacities up to 2000 tonnes per hour are easily handled and loads greater than 10000 tonnes per hour can be carried.

The introduction of synthetic reinforcing materials has meant that single conveyor lengths of five kilometres or more are possible, and, if transfer points are used to transfer the load to another belt, the possible length of a conveyor system is virtually unlimited. When transportation lengths of such magnitude are required the belt conveyor provides the only feasible method of continuous conveying. (Although as movement of material by fluid flow develops this might provide an alternative.) Recent studies have shown that the cost of belt conveying for these long distances compares favourably with road and rail transport [1] (See Table 1.1.1.) Another analysis [20] which compared different methods of transporting ore from Walsum, West Germany to Rotterdam (206 kilometres) showed that, although not representing the cheapest method, a belt conveyor could give a better overall system than train or ship if other factors such as environmental effect and safety were considered. (The factors considered and their relative values were (a) Cost, 50%, (b) Technical conception, 5%, (c) Safety of the system, 5%, (d) Environmental effect, 25%, and (e) Development time required, 15%.)

The belt conveyor is, however, not without problems. The belt itself is susceptible to accidental damage if it becomes caught in some of the conveyor machinery. (The National Coal Board has carried out much work to try and find the parameters affecting the belt's resistance to this type of damage - See Section 2.6.) Cleaning sticky material from the belt can be troublesome. It is very difficult to make a belt conveyor travel through a horizontal radius and so the installation has to be almost perfectly straight. The introduction of synthetic materials such as nylon for belt carcase reinforcement has meant, as stated above, that longer and longer conveyor installations can be envisaged, but this has given rise to the need to consider belt design parameters that were previously of no concern. Unfortunately there is very little up-to-date information on the mechanism of material transportation by belt and so data on the design parameters affected has largely been extrapolated from experience with older types of belting. (That is, belting reinforced with natural materials.) Specific examples of the problems that this has led to are described later in this chapter.

Despite these problems it is likely that the use of belt conveyors will continue to increase because of the significant advantages that they have over other conveying systems.

HAULAGE METHOD	BELT CONVEYOR	RAIL-ROAD HAULAGE LOCOMOTIVE	TRAM LINE	DUMP TRUCK
Distance from mine to port (km)	6.50	14.00	6.50	10.80
Relative transportation cost per ton-km	1.00	0.58	2.29	1.30
Relative transportation cost per ton	1.00	1.26	2.29	2.16
Relative capital cost (Based on production of three million tonnes per year)	1.00	1.30	0.81	0.97

TABLE 1.1.1

Comparison of modes of transport by Rangarajan

(h) Pneumatic Conveyors

The most commonly used type of pneumatic conveyor is the type where material is transported through a pipe network by air pressure [4]. It is best used with powder or granular materials. The advantages of the system include simplicity of layout, ease of automatic control, economy of space, minimal maintenance, cleanliness and the ability to combine other functions such as drying, mixing, sifting and even chemical reaction [21]. Capacity rates up to 1500 tonnes per hour can easily be achieved. There are no associated moving parts such as belts or chains and any extreme temperatures that need to be considered can easily be dealt with.

The main problems with this type of system are the extremely high power costs required, the inability to handle sticky materials and the machinery wear that can occur with fine particles. The poor power efficiency obtained with pneumatic conveyors means that they are not usually installed if the transportation route is simple and straight. There are developments in pneumatic conveying (and other fluid-flow systems) which are intended to help overcome these problems, and so more use of this system might be made in the future.

It is obvious from the short descriptions given above of the different bulk material conveyors available that there are no clearly defined application areas where each type is to be used and that for most tasks any one of several different systems would prove to be satisfactory. It is therefore the responsibility of the conveyor design engineer to obtain answers to the questions given at the beginning of this chapter, select the conveying systems that will perform the task satisfactorily, consider the capital expenditure and running costs for these systems and then choose that which fulfils the requirements most economically. Probably the most effective method of carrying out this work would be to perform a "Use-value analysis" such as that mentioned above for the comparison of different ore transportation systems from West Germany to Rotterdam. (That is, relative importance values should be given to different design factors such as capital cost, maintenance cost, safety and material contamination. The overall most suitable system could then be selected.)

Buffington [5] has given two selection guides, presented in Figures 1.1.4 and 1.1.5, to make this procedure easier. As with in-plant material handling, it can be seen that the belt conveyor represents a system suitable for many different tasks, and so, because of this versatility, the belt conveyor is very often the transportation system finally selected. The large amounts of conveyor belting produced annually are shown by Table 1.1.2 reproduced from a report by Meersseman [22] . This table shows that in 1974, 335,000 tonnes of conveyor belting were produced in the world. The great increase in the usage of conveyor belting over the last few decades can be seen from figures quoted by Spain [23] ; In 1939 approximately 8,250 tonnes of belting were manufactured in the U. S. A. and by 1963 production had risen to 28,000 tonnes.

The discussion above has shown that belt conveying represents a very important contribution to industrial efficiency. It is therefore necessary to ensure that, having decided to install this type of system, the most suitable type of belt which will perform the task is

KEY:

* Conveyor suitable for task.

1. Special considerations required. Consult Engineering. Examples spec.steels spec.belts,enclosures etc.
2. In combination of inclined & horizontal units only.
3. Investigate max.operating angle for material handled.

	Loading		Conveyor Profile: Angle of Incline					Complexity		Horizontal Carry			Vertical Lift				Material Properties: Material Size						Flowability ° of Repose				Temperature				Corrosive			Abrasive		
Conveyor Type	Controlled Feed	Uncontrolled Feed	Horizontal	10° Max.	18° Max.	30° Max.	Up to 45° Max.	Straight Line	Compound	Up to 100 Feet	Up to 250 Feet	Over 250 Feet	Up to 30 Feet	Up to 60 Feet	Up to 80 Feet	Over 80 Feet	Silt (under 250 M)	Very Fine (250 - 100 M)	Granule (100 M to 1/4")	Pebble (1/4" to 2")	Small Cobble (2" - 4")	Med. Cobble (4" - 6")	Flushing (0 - 15°)	Free Flowing (15 - 35°)	Sluggish (35 - 60°)	Sticky (60 - 90°)	Cold (below 32° F)	Ambient (32° - 150° F)	Hot (150° - 300° F)	Very Hot (300° - 900°F)	Non-Corr. (above 7 P.H.)	Mildly Corr. (5-8 P.H.)	Very Corr. (1-5 P.H.)	Non Abras. (1-3 MOH)	Mildly Abras. (3-5 MOH)	Very Abras. (ab. 5 MOH)
Apron-deep pan	*	*	*	*	*	*	1	*	2	*	*		*	*	*				*	*	*	*	*	*	*		*	*	*	*	*	*	1	*	*	*
Apron-hinged pan	*	*	*	*	*	1		*	2	*	*		*	*						*	*	*					*	*	*	*	*	*	1	*	*	
Apron-shallow pan	*	*	*	*	*	1		*	2	*	*		*	*					*	*	*	*		*	*		*	*	*	*	*	*	1	*	*	*
BELT-RUBBER BELT	*		*	*	3	3		*	2	*	*	*	*	*	*	*		1	*	*	*	*	1	*	*	*	*	*	1		*	*	*	*	*	*
BELT-SHUTTLE	*		*		*			*		*	*							1	*	*	*	*	1	*	*	*	*	*	1		*	*	*	*	*	*
BELT-WITH PLOWS	*		*	*	*			*	2	*	*	*	*	*	*	*			*	*				*	*		*	*	1		*	*	*	*	*	*
BELT + MOBILE TRIPPER	*		*		*			*		*	*	*	*	*	*	*		1	*	*	*	*	1	*	*	1	*	*	1		*	*	*	*	*	*
BELT + STATIC TRIPPER	*		*	*	3	3		*	2	*	*	*	*	*	*	*		1	*	*	*	*	1	*	*	1	*	*	1		*	*	*	*	*	*
Drag Chain	*	*	*	3	3		*	*	2	*	*		*				*	*	*	*	*			*			*	*	*	*	*	*	1	*	*	1
En Masse	*	*	*	*	*	*		*	*	*	*		*	*	*	*		*	*	*	*			*			*	*	*		*	1	1	*	*	
Flight	*		*	*	*	*	1	*	2	*	*		*	*	*				*	*	*	*		*	*		*	*	*		*	1	1	*	*	
Screw	*	*	*	*	1	1	1	*		*	*	1	*	1			*	*	*	*	*			*			*	*	*	*	*	1	1	*	*	1
Vibrating	*	*	*	*	1			*		*	*	*	*				1	1		*	*	*		*			*	*	*	*	*	*	*	*	*	1

FIGURE 1.1.4

Conveyor Selection Guide 'A' by Buffington

Type of equipment (* denotes suitability of equipment for task)	Material Characteristics									Path of Travel				
	Size			Flowability			Abrasiveness							
	Fine	Granular	Large lumps	Very free	Free	Sluggish	Non-ab.	Mildly ab.	Very ab.	Horizontal (H)	Inclined (I)	Verical (V)	Combined (HI)	Combined (HV)
BELT CONVEYORS	*	*	*	*	*	*	*	*	*	*	*		*	
Oscillating Conveyors	*	*	*	*	*		*	*	*	*				
Apron Conveyors	*	*	*		*	*	*	*	*	*	*		*	
Screw Conveyors	*	*		*	*	*	*	*		*	*	*	*	*
Flight Conveyors	*	*	*	*	*	*	*	*		*	*		*	
Wide Chain Drag Conveyors	*	*		*	*		*	*		*	*			
Bucket Elevators	*	*		*	*	*	*	*	*		*	*		
Skip Hoists	*	*	*	*	*	*	*	*	*		*	*		
Gravity-discharge	*	*	*	*	*		*	*				*	*	*

FIGURE 1.1.5

Selection guide 'B' for bulk handling equipment by Buffington

	Textile Reinforced Belts	Steel Reinforced Belts	P. V. C. Belts	TOTAL
U. S. A. and Canada	71,000	9,000	20,000	100,000
West Europe	45,000	30,000	20,000	95,000
Japan	38,000	12,000	-	50,000
Russia	26,000	4,000	-	30,000
Rest of World	52,000	8,000	-	60,000
TOTAL	232,000	63,000	40,000	335,000

TABLE 1.1.2

Worldwide conveyor belt production (tonnes) 1974 used. (The actual belt can represent thirty to fifty percent of the capital costs of an installation.) It is also important, of course, to ensure that the most suitable belt type is selected when a replacement is required for an existing conveyor.

Unfortunately belt design and selection suffers from the fact that belt conveying was a static technology for many years. During this period conveyors were relatively short and lightly loaded and the technical properties required from a belt were, more or less, met automatically by the constructions that were available. It can be seen from the production figures quoted above that the use of conveyor belting increased rapidly over three or four decades. This was mainly due to the introduction of synthetic fabrics enabling heavier loads to be carried over longer distances. However, the techniques of belt design and selection (reviewed in Chapter 3) did not keep pace with the changes occurring in the design of the conveyor installation itself. This has meant that belt selection procedures used at present are extrapolated from those which have proved to be adequate for older types of installations and the belt constructions used on them.

This has led to some problems, one of which has meant that, due to the nature of belt constructions, certain belt properties are grossly overspecified because of the need to provide satisfactory values of other properties. Before discussing this point more fully it is worthwhile to describe, in general terms, the present situation of belt conveying and belt construction selection.

1.2 Modern Belt Conveying Practice

A history of belt conveyor development is given in Appendix 2. This shows that the most significant event occurring in recent years is the introduction of synthetic fabrics as carcase reinforcement thus allowing the re-introduction of deep-troughing* and the use of longer and longer conveyor installations.

When initially put onto the market the main advantages claimed for all-synthetic carcase reinforcement were [24] :-

(a) Lighter belts. As the belt weight is involved in power calculations (See Section 3.3) it was claimed that less horsepower was required to drive synthetic belts.

(b) High flexibility meaning that the belts could be used on small diameter pulleys and that troughability* would cause no problems.

(c) Superior impact resistance due to the low modulus of the fabrics.

Deep troughing, too, was claimed to have several advantages [25] :-

(a) Increased carrying capacity.

* See glossary of terms at the beginning of this thesis.

(b) Reduction in power consumption due to elimination of skirt boards. Skirt boards* on 20° idlers* were used to prevent spillage and, as they rubbed against the belt, they increased the power requirements.

(c) Reduction in capital and maintenance costs as increased idler pitches* were possible because of less belt sag occurring between idlers due to the beam effect caused by the belt profile.

(d) Increase in permissible gradients. (With deep troughing the material is heaped into the belt centre, thus creating better internal material adhesion.)

It therefore appeared that the combination of deep troughing and all-synthetic belts would provide an excellent conveying system. This, however, was not completely the case. The advantages described above assumed that the methods of belt selection in common use for the then existing carcase constructions were valid for the new forms as well. In fact, the validity of the selection rules for the older constructions was not proven, but as conveyors tended to be fairly short and lightly loaded, this was not too critical.

The main criteria that influenced the choice of belt construction in the days of all-cotton carcases were that the belt had to have:-

(a) Sufficient width to carry the load at a practical speed.

* See glossary of terms at the beginning of this thesis.

(b) Sufficient tensile strength to withstand the maximum working tension to which it would be submitted. (The calculation for maximum working tension - see Section 3.3 - allowed for (i) minimum drive requirements, and (ii) limiting sag between adjacent idler sets.)

(c) Sufficient thickness and correct grade of cover compound.

(d) Sufficient lateral flexibility so that it would at least touch the centre roller of an idler set when empty. (See Figure 1.2.1 for an example of this occurring on a working conveyor.)

(e) Sufficient longitudinal flexibility to flex easily around the drive and tail pulleys.

Most of the criteria had no theoretical or experimentally proven foundation but were based purely on experience in service.

With cotton belts the large number of plies sometimes required to provide adequate tensile strength caused the carcase to be very thick and stiff, thus causing difficulty in troughing correctly even with small idler angles* of 20^{o} (See Figure 1.2.2). A simple test, known as the F/L test was introduced into the British Standard [26] to check that the belt would be flexible enough transversely and thus meet criterion (d). This test, discussed fully in Section 2.2, apparently proved sat-

* See glossary of terms at the beginning of this thesis.

isfactory in practice. The problem of insufficient longitudinal flexibility was usually overcome by using very large diameter end pulleys, thus reducing the strain occurring in the outer plies of the belt as it went round the pulley surface.

As mentioned above, the need for accuracy in the tension calculation for all-cotton belts was not too critical because of the nature of the conveyor installations. In any case, a 10:1 safety factor was used for the ratio of actual belt tensile strength to maximum working tension induced. This was intended to allow for [27] :-

(a) Additional loading caused by flexing of the belt on passing around the end pulleys.

(b) Loss of strength due to the plies not all bearing the load uniformly.

(c) Loss of strength from areas of shock.

(d) Loss of strength from ageing.

(e) Excess loading on starting as a result of acceleration of load, belt and rollers.

It is interesting to note that this reference demonstrated that, theoretically, a lower safety factor represented an optimum value. No records of this being verified experimentally are available.
(See Section 2.2.)

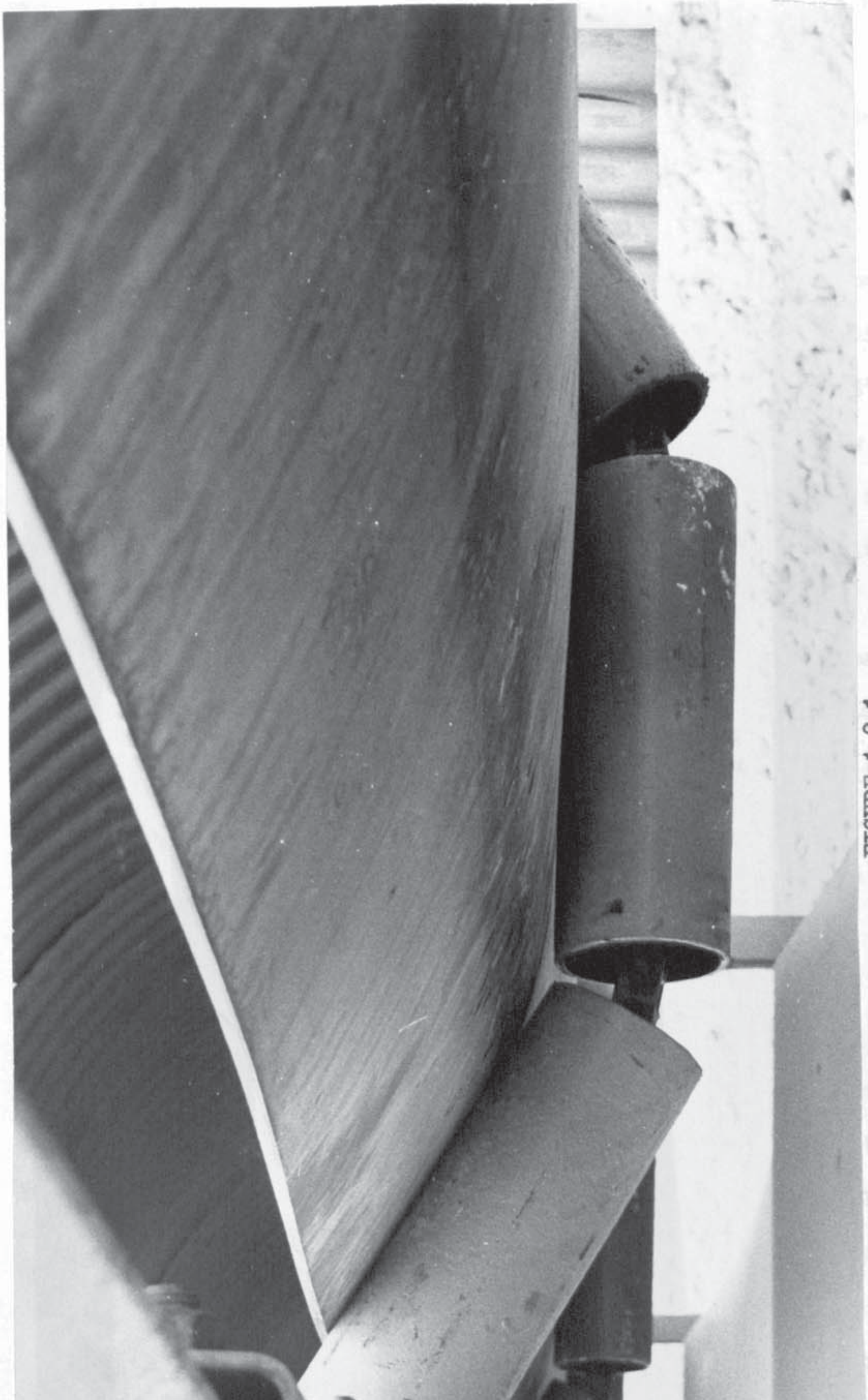


FIGURE 1.2.1

View of working belt exhibiting good troughability

Thus, as it was a straightforward matter to select the correct belt width and the required amount of cover (from rules again based on experience), it was very easy to select the belt construction design to be used for the application in hand.

When the same selection rules were used with the new types of belting a thin, very flexible carcase easily meeting troughability requirements resulted. When put into service these belts were often "nipped" between the rollers of an idler set. (See Figure 1.2.3.) This is the basic problem of "load support". Load support is defined, for the purpose of this thesis, as the belt's ability to carry the load without material spillage or damage to the belt or conveyor structure. Strictly, this property should be subdivided into transverse load support and longitudinal load support. Longitudinal load support is measured by the amount of sag that occurs between adjacent idler sets. (Permissible sag limits are discussed in Section 2.2.) As the provision of this longitudinal load support rarely presents problems, the term load support by itself implies transverse load support.

Another problem found with the all synthetic belts was the lack of resistance to impact. These two problems (load support and impact resistance) were overcome by using more plies than that required by tensile strength considerations and "bulking" out the belts further by increasing the amount of rubber sandwiched between the plies. This situation still exists and belt tensile strength requirement often must be overspecified in order to ensure adequate load support is provided.

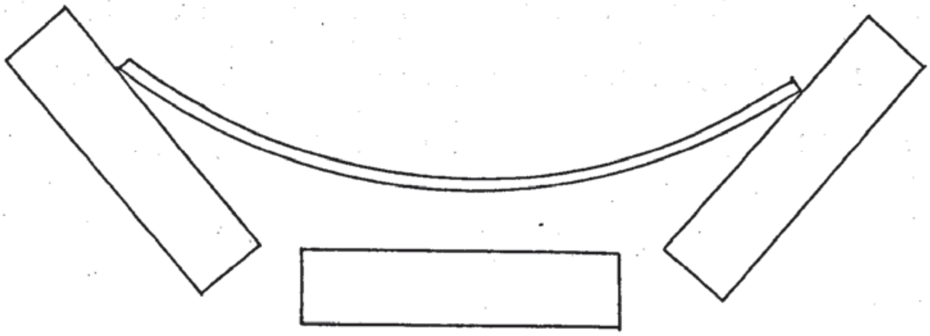


FIGURE 1.2.2

Inadequate belt troughability

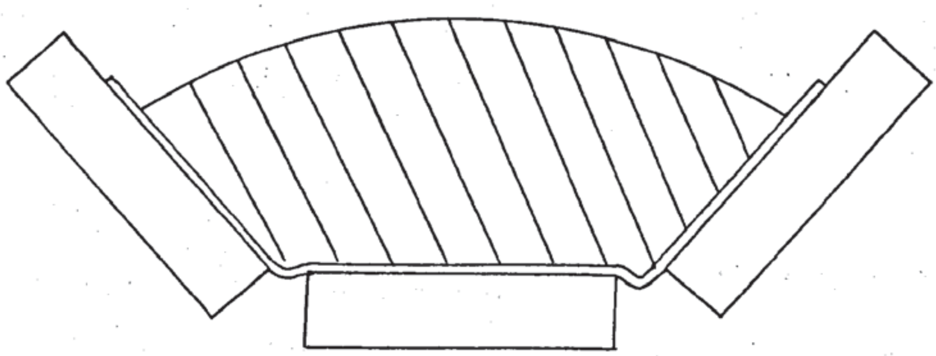


FIGURE 1.2.3

Inadequate belt transverse load support

(See Section 6.1 for a quantitative assessment of this overspecification.) It has been argued that this situation, although not representing an optimum combination of belt properties, is preferred by customers, anyway, because they associate very thin belts with inferior quality.

Thus, two more design criteria have had to be added to the list given above. These are that the belt must have:-

(a) Sufficient lateral stiffness so that it will not spill the material or get nipped between the rollers of an idler set.

(b) Sufficient impact resistance to withstand the impact at the loading point.

Another criterion, that of providing a belt cosmetically acceptable to the customer could be included, but a more useful, long term solution to this problem would be to demonstrate the superiority of a belt having the optimum combination of properties from the point of view of cost and service performance.

Criterion (b) is usually met by increasing the thickness of the belt top cover and introducing special cushioning rollers under the loading point. Developments in the methods of weaving the fabrics used have also helped.

Unfortunately, complaints about lack of load support still occur (see Section 7.6 for an example.) This is because there is neither a rational theoretical or empirically based method of calculating the amount of load support required by a given situation nor a means of assessing the load support capability of a given belt construction type. The reason that few complaints now occur is probably that the load support capability of a belt construction is underestimated and so overspecification of this property occurs.

Although load support is usually the only design criterion that demonstrates that the selection rules are inadequate, it is likely that the others are not entirely valid. This is not shown up in service, however, because of the safety factors incorporated into the selection procedure and the overspecification in some properties caused by the provision of load support. The area where this is most likely to be the case is that of the calculation of power requirements and necessary belt tensile strength (See Section 3.3). One of the advantages claimed for synthetic belts is their lower weight causing less tension to be induced in the working belt - thus reducing the power requirements of the conveyor. In March, 1976, a nylon reinforced belt was installed on a conveyor in a foundry which had previously always had belts of a cotton/nylon mixture. The customer immediately complained that his power consumption levels increased dramatically. This phenomenon was not consistent with the formulae for power and tension calculation and so showed that they were probably incorrect.

German work has shown that increasing idler angle increases power requirements [28]. This is not consistent with the claims of deep trough conveying given above.

It can be seen that the extrapolation of belt selection rules from the old cotton belts to the constructions used at present has not been entirely successful. This is probably because the selection rules used previously were not entirely correct, but the belt applications to which they were applied were not severe, and so no problems occurred. The development of deep trough conveying made possible by synthetic belts has thus led to difficulties in knowing exactly the design properties required from a belt for a given conveyor installation. In addition, there are difficulties in assessing some of these design properties in laboratory tests and supplying them without excessive overspecification of others. (See Chapter 4.) This statement is particularly true for load support. Therefore, as belting applications will undoubtedly become even more severe, it is essential that a more ideal system of belt selection is introduced. The attributes required of such a system are discussed below.

1.3 <u>Requirements of an "Ideal" Belt Design and Selection Procedure</u>

If, as required by an ideal situation, a belt is to perform as efficiently as possible on a conveyor installation, it is important that the belt must meet the needs of that conveyor with little or no overspecification of any design properties. (With the proviso that certain safety factors are required to allow for unforeseen circumstances such as excessive belt overloading.) If the ideal situation is to be achieved there are two basic "rules" that must apply:-

(1) The belt selection procedure used for that conveyor must assess exactly all the needs of the installation. (That is, the selection procedure must include valid rules for the determination of all belt properties which will affect service performance.)

(2) The actual belt properties concerned in the selection procedure must be known accurately from measurement.

These "rules" apply mainly when a replacement belt is to be found for an existing installation. If a completely new installation is to be built, then there should be liaison between the conveyor designer and the belt designer so that the optimum combination of belt and machine can be found. Although the Mechanical Handling Engineers Association publish a booklet "Recommended Practice for Troughed Belt Conveyors" [29] which was written with the help of belting manufacturers, there appears to be little co-operation when it comes to specific situations.

If the two "rules" mentioned above are to be met, the following features are required:-

(a) All belt and conveyor properties affecting service performance and life must be known.

(b) Quantitative assessment of the effect of these properties on belt performance must be available.

(c) A testing technique relevant to service needs must be used to measure the belt properties in (a).

Certain aspects of this are not, at present, available. If they are to be, then controlled field trials of different belts working under different conditions must be carried out. This would show the mechanism of belt breakdown and hence would lead to a knowledge of the factors that affect it. Comparison of the results obtained from field trials and small scale laboratory belt testing would establish quantitative correlation (if any exists) between belt properties and belt performance. It should be remembered that belt properties might be affected by the differences in the raw materials or in the manufacturing process. It can be seen that the information collected in any field trial must therefore be very comprehensive in that it must include full raw material details, manufacturing details, finished belt properties, installation details and load properties. Ensuring that the testing techniques reproduce, as nearly as possible, service conditions would make it easier to establish any correlation that might exist.

An ideal belt selection and design procedure can be summarised as one (a) having a proven relationship to belt service performance and (b) using values of belt design properties assessed from service related testing methods. If such a system was available it would be important to apply the procedure rigorously in order to check continually its validity. If belting or conveyor developments reduced its applicability then it would be necessary to review the system, but the basic data concerning belt performance would still be available from the original field trials.

The present belt selection procedures, as has already been described, do not always demonstrate the features of an ideal system. Belt testing to the various applicable international standards can, in general only be regarded as control testing and has little relevance to field performance. It is therefore vital that work intended to help approach an ideal system more nearly is started as soon as possible.

The next chapter reviews the previous work which has been carried out on the various facets of belt selection and design discussed above and subsequent chapters establish the area most urgently in need of research and describe the experimental work carried out to study it.

CHAPTER 2

REVIEW OF SCIENTIFIC LITERATURE RELATING TO CONVEYOR BELT DESIGN AND SELECTION PROCEDURES

Two major publications provide excellent descriptive information concerning the design, manufacture and use of belt conveyors. These are (a) "Belt Conveyors and Belt Elevators" [30] by Hetzel and Albright and (b) "Belt Conveyors for Bulk Materials" [31] published by the Conveyor Equipment Manufacturers' Association (CEMA). However, if a more detailed approach is required, it is necessary to consult papers and articles presented in the trade journals of industries relying on efficient material transportation. As mentioned above at the end of Chapter 1, the published literature on the various features that make up an ideal belt design and selection procedure is reviewed here. Many articles covering the whole spectrum of belt conveying have been reviewed during the course of this research, and so, complementary to this chapter, a full bibliography of these is presented in Appendix 3.

The following aspects of belt conveying are dealt with here: (1) Belt Selection Procedures; (2) Derivation of Selection Rules and Assessment of Belt Design; (3) Laboratory Testing of Belt Properties; (4) Service Information Collection; (5) Field Trials and (6) Correlation of Laboratory Testing and Service Information. There are no distinct borders between some of these design aspects and so the work of some researchers falls into several of these categories. More detailed references to specific aspects of the research work described in the present thesis are included in the relevant section. (For example, some idler loading literature is used in Section 7.2 to help design a special idler set for experimental work.)

2.1 Belt Selection Procedures

Several manufacturers publish manuals in which their methods of belt selection are described. These are critically examined in Chapter 3. In addition to these manuals, several papers have been published outlining the general principles of belt selection which are as described above in Section 1.2. Examples of such articles are those by Kyle [32] and Bullough and Routledge [33] . Both of these papers give worked examples of how belt selection is carried out and describe the factors that decide the final outcome, but they do not attempt to examine the basis of the various selection rules involved. Several of the articles of this type deal with the mechanical parts of the conveyor as well as the belt.

A more complete work is that of Liggins [9] in which he divides belt selection into two parts:-

(1) The selection of the most suitable belt cover elastomers and the associated compound formulations.

(2) The selection of the type and strength of the carcase. He stresses the importance of determining the minimum economical strength of the belt carcase required to give a good field performance. This is essential because the load bearing members invariably constitute

the major component of the raw materials costs.

Liggins advises that, although most manufacturers' selection techniques are basically the same, great care is required when comparing one manual with another. This is made necessary because of the confusion that can result from the different definitions of certain selection factors used by the various manufacturers (See Chapter 3). The major contribution that Liggins' paper makes is its attempt to show how some of the belt design data is derived whereas the papers mentioned previously only tabulate this data. Liggins also describes some of the factors that decide the choice of fabric and shows how the various fabric weaves available can affect the final belt properties.

Another general, but comprehensive, article is that of Grant [34]. In his paper a summary is presented of the more important conveyor belt design criteria and not only those mentioned in Chapter 1 are included but also others which tend not to be included in published selection procedures are listed. (The reason for not including certain criteria in selection manuals is because they represent required features that are built into all the different belt construction types available.) The full list of properties given by Grant is:-

Load support - Longitudinal
- Transverse

Elongation at operating tension

Resistance to growth

Safety factor

Modulus

Flex life

Impact resistance

Rip and tear resistance

Fastener holding ability

Vulcanised splice fabrication

Property retention when wet

Property retention at high temperature

Chemical attack resistance

Weight

Grant does not discuss all of these criteria but points out that some of them require a "more objective assessment when considering new materials". The two particularly mentioned are those of elongation at operating tension and safety factor. The elongation at operating tension dictates the amount of take-up* travel necessary in any conveyor system. Safety factor, in this case, is the ratio of belt tensile strength to maximum permissible operating tension and it is intended to allow for additional stresses as described in Section 1.2. Grant suggests that possibilities for substantial reductions in safety factors exist by the more efficient application and processing of both the newer and the more familiar carcase materials. He states that "safety factor as a design criterion could then be more objectively assessed in terms of other required properties such as flex life".

* See glossary of terms at the beginning of this thesis.

The National Coal Board, who, as is shown below, have carried out a great deal of work on nearly all aspects of conveyor belt use, issue an Information Bulletin [35] to "help managerial or engineering staff to avoid such errors as:-

(a) Using a belt which is too small or too slow to carry its load.

(b) Installing a motor which is likely to be overloaded.

(c) Installing too powerful a motor which may cause damage to the belting or its joints.

(d) Extending a belt to such a distance that there is undue risk of breakdown.

(e) Using belting on drums of unsuitable diameter.

(f) Using unsuitable belts."

The bulletin therefore describes a simplified method of selecting the correct belt to use under the conditions at hand. The stages of this particular selection procedure require no knowledge or understanding of belt design from the user. It is interesting to note how the process can be reduced to such a simple task by the use of tables and nomographs but it is unfortunate that it is probably based on incomplete and sometimes invalid selection rules and belt data. More recently the National Coal Board has issued a manual [36] that describes a similar procedure to that in the Information Bulletin but also attempts to show the basis of some of the tables and formulae used in belt selection.

The selection of suitable belt covers is a much more straightforward, but no less important, task than the choice of carcase. As mentioned above, Liggins and several other authors describe some of the types of cover compound available. Buettner [37] provides a full review of those in common use and describes some of the information required by the belt designer to help in the selection of the correct cover. This is:-

(1) Types of oils and chemicals present and their temperatures and concentrations.

(2) Heat - is it dry, moist, constant, intermittent, enclosed or open?

(3) Describe fully the loading conditions, size and nature of the product.

(4) Is there moisture?

(5) Is a food grade compound required? Must it be non-marking or of a special colour?

(6) Is stickiness or release of material a problem?

(7) Is the belt supported on the carrying side by flat or troughed idlers or is it a slider* bed?

(8) How has any previous belt cover responded to the service conditions; what grade of compound was it?

(9) Are there any ploughs, deflectors or skirt boards in contact with the belt cover?

Buettner then presents a guide to the areas of application of the commonly available compounds (See Figure 2.1.1). This guide is intended for relatively light duty applications and does not mention fire resistance - the property required from belts to be used in coal mining. Hallam [38] has published a comparison of some cover compounds which meet this requirement.

One other aspect of belt design and selection which has been singled out as subject matter for an article is that of selection of the correct type of mechanical fasteners*. Mechanical fasteners represent an alternative to vulcanised* splicing for making a conveyor belt endless. The article, "Selection and Installation of belt fasteners" [39], gives some useful guidelines to help ensure that the correct type is chosen:-

(a) Overall belt thickness governs the size of fastener used. Make certain the fastener selected will accommodate the belt.

* See glossary of terms at the beginning of this thesis.

KEY E - Excellent S - Satisfactory F - Fair D - Doubtful U - Unsatisfactory	Natural (NR)	Gen. Purpose Synth. (GRS)	Nitrile (NBR)	Neoprene (CR)	Polyvinylchloride (PVC)	Silicone	Teflon (TFE)	Polyurethane (PU)	Butyl (HR)	E P D M
Acids - Dilute	S	S	S	S	E	S	E	U	E	E
Ageing	S	S	S	E	E	S	S	S	E	E
Abrasion - Ambient Temp.	S	S	S	S	S	D	U	E	F	F
Abrasion - High Temp.	U	U	D	F	U	D	U	U	S	S
Cutting and Gouging	S	F	D	S	S	D	U	E	S	S
Oil : Animal	U	U	E	S	E	U	S	E	S	D
Vegetable	U	U	S	S	E	U	S	E	E	S
Mineral	U	U	E	F	E	U	S	E	U	U
Ozone	F	F	S	S	E	E	S	S	S	S
Release	U	U	D	D	F	E	E	F	S	S
Sunlight	F	F	F	E	E	S	S	S	E	E
Tear	S	F	F	S	F	D	D	E	S	F
Temperature : High	U	F	F	S	U	E	E	D	E	E
Low	E	F	F	D	D	E	S	S	S	S
Water	S	S	S	S	E	S	S	F	S	S

FIGURE 2.1.1

Comparative resistance of belt cover compounds (by Buettner)

(b) Where fasteners are to be recessed, measurement should be taken after the belt cover has been removed in the splice area.

(c) Of the basic type of mechanical fasteners available:
(1) Separable fasteners are recommended for portable equipment where the belt is to be moved or extended;
(2) Solid plate fasteners, as a general rule, are used for permanent installations on heavy duty bulk conveyors and elevators.

(d) Higher belt tensions require compression or semi-compression fasteners, preferably with a number of staggered points of attachment to the belt.

(e) Fastener selection is restricted by the diameter of the smallest pulley over which the belt has at least a 90° wrap*. Where operating tension is lower than the belt rating, the minimum pulley diameter may be reduced proportionately.

(f) The kind of material being conveyed also has a direct bearing on fastener choice.

The type of article discussed above is really aimed at making the general principles of belt selection understood to the conveyor user. They do not pretend to make any contribution to the development of

* See glossary of terms at the beginning of this thesis.

belt design, but they do represent an important section of the literature available on conveyor belting. They also, of course, perform a useful task in that they show the belt user that the specification of belting is not completely straightforward and that several factors must be considered. Bernardov [40] has shown that incorrect belt selection caused by not obeying the selection rules can lead to gross overspecification thus increasing costs significantly. (It should be remembered that the commonly used selection rules lead to overspecification of certain properties anyway.) The literature reviewed below concerning more specific facets of belt design and selection tends to represent a more scientific contribution to conveyor belt technology.

2.2 Derivation of Selection Rules and Assessment of Belt Design Parameters

The conditions under which belt conveyors work are so extremely diverse that it is very difficult to reduce belt design requirements to a simple formula; there have been attempts to do this, however, for some of the more basic belt design parameters. It is unfortunately the case that, having found that these parameters can be theoretically derived, the formulae are often elaborated so much that they lose their usefulness to the belt user. A typical example of this is given by the formulae available to calculate the cross-sectional load area on a belt. (A parameter required in order to estimate the carrying capacity of the conveyor.)

It has been mentioned above that Liggins [9] has attempted to show how some aspects of the belt selection procedure were derived. For example, rather than just present a table of data for the carrying capacity of a belt under different conditions of belt width, troughing angle, belt speed and material density, Liggins quotes a formula to derive the cross-sectional load area.

It is interesting to compare the formulae quoted with those derived by other workers and organisations. Liggins' formula, which is empirically based, is:-

$$A = C \; (0.9b - 0.05)^2 \qquad (2.2.1)$$

Where A = Cross-sectional area (m^2)

C = Empirical constant depending on troughing angle (C is found from a table presented by Liggins)

b = Belt width (m)

The formula recommended by the British Standards Institution is [41]:-

$$A = 10^{-6} \; S \sin\beta\,(W + S\cos\beta) + 0.5R^2(2\alpha - \sin 2\alpha)\; m^2 \qquad (2.2.2)$$

This formula is based on the assumption that the cross-section of a bulk material on a belt is formed by a trapezium and a circle segment, and in it:-

$$S = 0.5\,(b - W) - (0.05\,b + 25) \qquad (2.2.3)$$

and

$$R = \frac{0.5\,W + S\cos\beta}{\sin\alpha} \qquad (2.2.4)$$

In this case S, R, W and b are in millimetres. The angle α must be in radians (a fact not emphasised in the British Standard). The meaning of the symbols used in these formulae can be seen from Figure 2.2.1.

It is obvious that the formula presented by Liggins is much easier to apply than that of the British Standards Institution. Other workers have also studied the design parameter of cross-sectional area. Koval'chuck [42] has investigated the optimum shape and position of rollers on an idler set to give maximum load area. He arrives at a complicated configuration which would almost certainly be uneconomic to produce. Cartwright [43] has considered the effect on load cross-section of increasing the idler angle of a standard three roller idler set and has shown that the optimum idler angle when carrying a load with a surcharge angle of 20° is approximately 58°. These pieces of work, although useful in that they demonstrate optimum conditions that should be striven for, show how the work carried out on one particular design property can, at the expense of others, assume much more importance than the property merits. For example, Koval'chuck does not discuss the effect of his optimum idler set on other design considerations such as power consumption, troughability and belt transverse stress. Cartwright does this, however, but unfortunately one of his major claims of the advantages of deep troughing - that of reduced power requirements - has been proved incorrect (See later in the section).

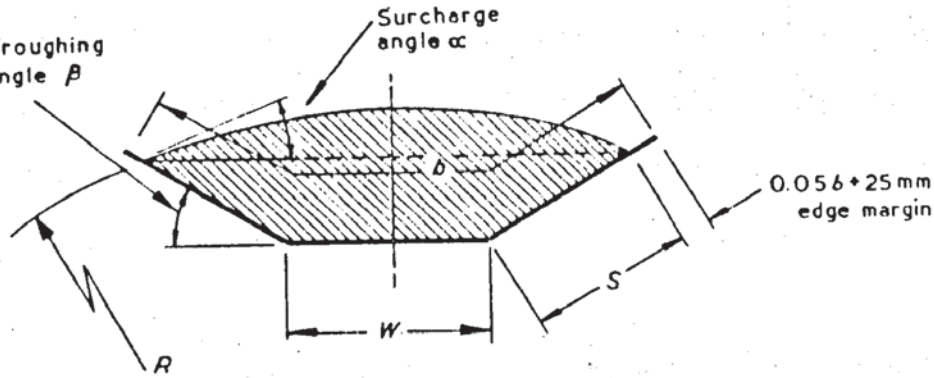


FIGURE 2.2.1

Nominal cross-section of load stream
showing load area parameters used in British Standard 2890

With belt conveying, as in any technology, it is important to ensure that any design rules are valid from service measurements and it is useful for belt development purposes if the rules can have some theoretical basis. It is, however, also important to make the rules as simple as possible to apply (with the proviso, of course, that the rules must still be valid within acceptable limits of accuracy). When considering the cross-sectional load area of a belt conveyor it is unlikely that, in practice, the theoretical capacity will be attained. Instead, this parameter will be affected by such factors as the lump size of the material.

Grierson points this out in his article "The Capacity Potential of Belt Conveyor" [44] and advises that, "as cross-section is conditioned by so many variables, it needs to be determined by measurement in each case". He also points out that conveyor manufacturers often recommend a 2:1 loading factor thus specifying a conveyor having a potential capacity twice that of the normal loading rate to allow for flood loading. From the experience gained during the work described in his article, Grierson recommends the use of empirically based formulae, such as that quoted above from Liggins.

Grierson, during the 1960's, published several articles concerned with the fundamentals of belt conveyor design. In his monograph "The Design and Use of Belt Conveyors in Mines" [45] he mentions, amongst comprehensive information on all aspects of belt design, some tests carried out to establish the formulae used in the calculations to find the maximum belt operating tension and resulting power consumption. (See Section 3.3.) These tests are described fully in another article [46] "Some Aspects of Belt Conveyor Design" published in 1963.

His work involved carrying out a series of tests on a horizontal conveyor, 60 inches wide, carrying gravel. In all, ten tests were made with the conveyor varying in length from 200 to 1800 feet. Horsepower measurements were taken when the conveyor was running empty and when the belt carried a regulated load of 1500 tons per hour.

It was found that, allowing for slight experimental error, the relationship between conveyor length and horsepower was linear. If horsepower was taken as the ordinate and conveyor length as abscissa, the line representing the test results did not pass through the origin, (See Figure 2.2.2) and so the relationship was in the form:-

$$y = m x + c \tag{2.2.5}$$

Where y = horsepower consumption

m = rate of change of power with conveyor length

x = length of conveyor

c = power consumed by conveyor independent of its length

From this work it was possible to establish empirical formulae to be used in a belt selection procedure. The methods for doing this are described by Grierson in his article.

In Section 3.3 it will be seen that all the manufacturers still use the basic rule of horsepower consumption being linearly dependent on belt length with an additional component, independent of length, which is considered to be due to the frictional effects of the end pulleys.

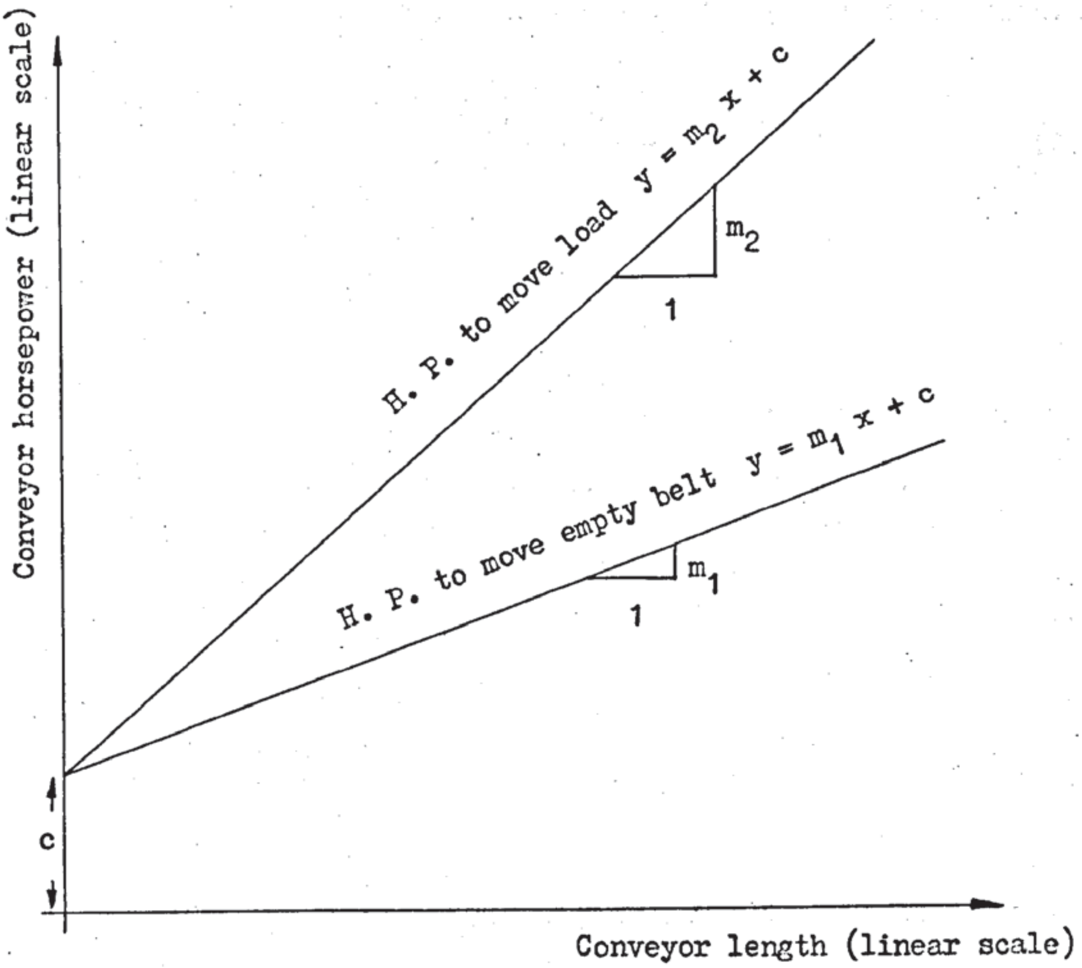


FIGURE 2.2.2

Linear dependence of conveyor power consumption on conveyor length as demonstrated by Grierson

Once the power consumption of a conveyor has been measured it is a simple matter to calculate the effective tension in the system. This is the tension difference, T_e, between the strands of the conveyor belt before (T_1) and after (T_2) the drive pulley (See Figure 2.2.3). T_e is directly proportional to the power consumption.

$$T_e = \frac{\text{Power at drive pulley (kW) x 1000}}{\text{V (m/s)}} \text{ N} \tag{2.2.6}$$

Where V = Belt Speed

The basic laws of belt drives are then used to find the maximum tension T_1, induced in the belt during operation [47].

$$T_1 - T_2 = T_e \tag{2.2.7}$$

$$\text{and } \frac{T_1}{T_2} = e^{\mu\theta} \tag{2.2.8}$$

Where μ = co-efficient of friction between belt and pulley surface

θ = angle of wrap of belt around pulley

It will be seen from Chapter 3 that in belt selection procedures this process is carried out in a slightly different order.

Other researchers have carried out work similar to that of Grierson. Asman [48] in fact carried out some less comprehensive work earlier than Grierson. In his article (published in 1959), Asman showed that the horsepower formulae used by different manufacturers did not give results consistent with each other. He therefore carried out work to establish the conveyor parameters affecting power consumption and belt working tension. He demonstrated that these parameters were load weight, type of idler bearing, belt speed, belt tension, belt type, idler spacing and load cross-section. He did not establish any quantitative rules to allow for the effect of these, although a formula based on the work is quoted by Dennehy [49] in the article "Overseas Observations on Belt Conveyor Design".

Dennehy also discusses the work carried out in the Hannover Technical High School under the direction of Professor A. Vierling. This institution appears to be the one to have published most scientific research work on belt conveyors. The work, carried out at approximately the same time as that by Grierson, showed that belt driving tension could be split into several components as follows:-

(1) Belt tension required to turn terminal and bend pulleys.

(2) Belt tension required to turn idlers.

(3) Belt tension required to move belt and material on loaded strand of conveyor.

(4) Belt tension required to move belt on return strand.

(5) Belt tension required to elevate material.

(6) Belt tension required to elevate belt.

Dennehy emphasises that great care is required in the choice of empirical factors to be used for each component in tension formulae if accurate results are to be obtained. Thormann [50] has carried out a comparison, supervised at the Hannover Technical High School, of the different techniques of tension calculation. He concluded that none of the methods then available was completely accurate,but suggested that for short conveyors a method splitting the calculations into several different components should be used and that with longer conveyors a calculation method should be chosen which has proven validity for the conditions which will be met in service. The references given in Thormann's thesis are intended to make this choice easier.

More experimental work dealing with power consumption of conveyors was published from the Hannover High School by Behrens [28] in his doctorate thesis in 1967. He established and quantified several important facts:-

(1) Power consumption is very strongly dependent on belt tension.

(2) Large idler pitches increase power consumption, but this effect is decreased as belt tension increases.

(3) Power consumption is influenced by troughing angle.

(4) Belt speed has little effect on the total power consumption per unit weight of material carried.

One very important conclusion was that conveyors with 30° idler angle were more economic than deep troughing installations of 45°. Although deep troughing was found to increase the possible load cross-section in Behrens' experiments he concluded that a more efficient method of obtaining the higher capacity would have been to increase the belt speed. This feature is not considered in the selection rules generally used for belt conveyors. If it was, then the trend towards deep troughing installations might be reversed. Behrens would appear to have been the first researcher to have studied the effects of all the different features of a conveyor on the power consumption. His work was carried out mainly using steel cord reinforced belts and was not continued to include the all-synthetic fabric constructions. It is felt by the present author that this work should now be carried out to see if the various power calculation factors which have been extrapolated from old types of belting to the new constructions are valid. Asman, as described above, found that belt type affected power consumption, so it is likely that they will be incorrect. The example quoted in Section 1.2 would also suggest this.

Once the maximum tension to be induced in a belt during operation has been found it is next necessary to establish a safety factor for the ratio of belt tensile strength to operating tension. This is required to take account of any uncertainty involved in the design calculations and of any unintentional extreme loading conditions that might occur. One particular feature that the safety factor is intended to allow for is that of the increased stress in the outer plies of the belt as it passes around the end pulleys (see Section 1.2) and some theoretical studies have been carried out to investigate this additional loading. Frenzel and Rothe [51] demonstrated in 1958 that the additional stress caused by bending in very thick belts could be in the order of the tensile strength of the belt. They suggested that the recommended safety factors from the German Standards Organisation [52] could lead to overbuilding of belts. Further work of Frenzel and Rothe was published in another article [27] and they demonstrated that the recommended safety factors did not necessarily represent the optimum values. This was because cotton belts were still in common use and a high specified safety factor resulted in the need for many fabric plies which thus gave a very thick belt and caused corresponding large additional stresses in it when passing around a pulley. Frenzel and Rothe demonstrated that, with certain pulley diameters, it would, theoretically, be better to reduce the safety factor thus allowing a thinner belt and decreasing the additional stress dramatically. It would appear that this theory was not tested and the high safety factors remained in common use. In more recent years Tabaddor has produced a more elaborate theoretical model for the bending of a belt around a drum [53] but this has not been incorporated into any published belt tension calculation methods.

The National Coal Board carried out an experimental exercise to study the effect on a belt as it passed around a pulley [54, 55]. In this work strain gauges were positioned in between the different plies of a belt sample. The object of these tests was not really to verify any theory but they did demonstrate that speculations made by Swift [56] as early as 1928 were correct. He had suggested that an arc of constant strain exists as a belt runs onto a pulley and that this is

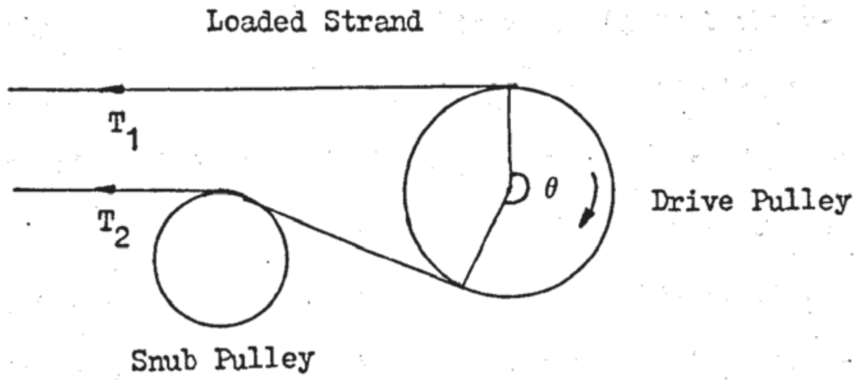


FIGURE 2.2.3

Tension change around the drive pulley

of a belt conveyor

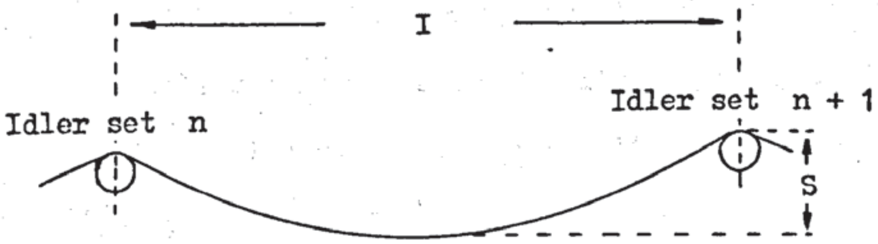


FIGURE 2.2.4

Inter-idler belt sag

followed by one of changing strain over which the tension changes according to the classical formulae of belt drives. (A full theoretical study of this is presented by Sanders [57].) The National Coal Board work contributed a great deal to the knowledge of induced tension in belt drives and also demonstrated that the nature of the cover and interply compounds used for the belt had little effect on the magnitude of the induced flexing strains at normal ambient temperature. This had been the original object of the work.

One other part of belt selection procedures that has been the subject of a theoretical study is that of limiting belt sag between idlers. This sag is usually expressed as the ratio of sag to idler pitch (that is, S divided by I in Figure 2.2.4). Published acceptable limits range from 1.5 to 3% for this ratio. It is generally assumed that this sag is determined by tension alone and so tension calculations in the belt selection procedure take this factor into account (See Section 3.3). The belt profile between the adjacent idler sets is taken as being a catenary, the shape taken up by a completely flexible string hanging under its own weight. It is easy to derive the tension required in the belt to limit the sag to any desired limit (see Appendix 4).

Van Leyen [58] has made a comprehensive study of inter-idler sag and has derived a formula for it that includes the effect of belt stiffness:-

$$\frac{S}{I} = \frac{W\,I}{8\,T}\left(1 - \frac{4}{I}\sqrt{\frac{E\,J}{T}}\right) \qquad (2.2.9)$$

Where S = sag (cm)

I = idler pitch (cm)

W = Total loading (Belt and load weight) (kg/cm)

T = Belt tension (kgf)

E = Elastic modulus (kgf/cm^2)

J = Second moment of area of belt cross-section (cm^4)

Van Leyen found that the factor $\frac{4}{I}\sqrt{\frac{EJ}{T}}$ was, in practice, much less than unity and so could be ignored. He therefore considered that the formula could be reduced to:-

$$\frac{S}{I} = \frac{WI}{8T} \qquad (2.2.10)$$

This is the formula derived from the catenary (See Appendix 4). It would appear from van Leyen's work that the assumption that belt stiffness has little bearing on the amount of sag is correct. Behrens' work [28], as mentioned above, showed that drive requirements were strongly dependent on sag and the effect of the idler angle on this parameter (see Figure 2.2.5). Behrens found that actual sag values differed slightly from those calculated by the catenary formula and that at 45° idler angle they were slightly less than at 30°. (This would be expected from an increased beam effect but does not agree with experimental results described in Section 8.2 of the present work.) However, for practical purposes, Behrens considered that the catenary formula would provide satisfactory results and the effect of idler angle could be considered to be negligible.

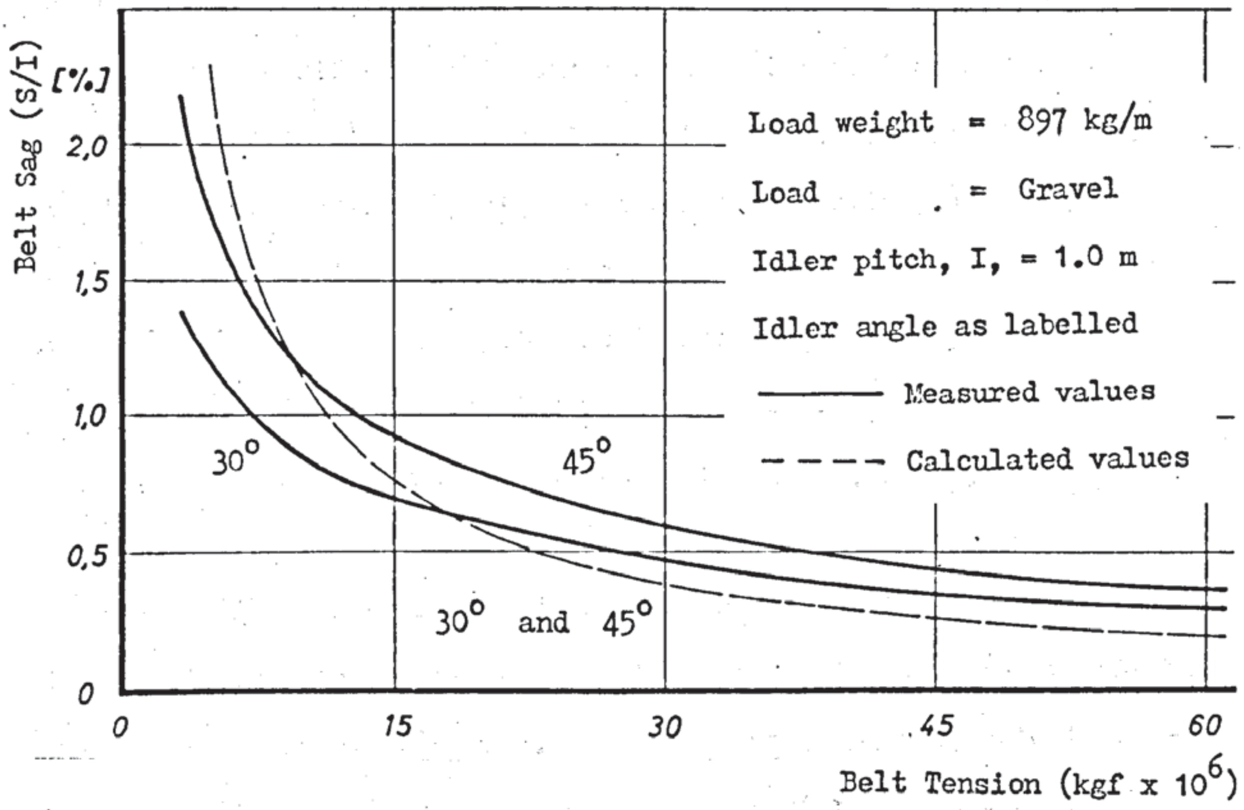


FIGURE 2.2.5

The dependence of inter-idler sag on belt tension measured by Behrens

More recent work by Oehmen and Alles [59] (again carried out at the Hannover Technical High School) showed that when belt constructions have very different transverse stiffness characteristics, there is an appreciable difference in sag values. (The belt constructions compared were textile reinforced and steel cord reinforced - steel cord belts have extremely high transverse flexibility.) The work also showed that the amount of sag approaches the theoretical value more closely as the idler pitch is increased. Some experimental work on inter-idler sag was included as part of the research for the present thesis and is described in Chapters 7 and 8.

Oehmen and Alles also included some basic work on the mechanism of belt resistance to impact and this is reported in the same article as their work on inter-idler sag. They were not, however, able to propose quantitive criteria as to how much impact resistance should be built into a belt working under certain conditions and so this design property can still only be assessed in comparative tests. Some impact test procedures are discussed in Section 2.3 and proven correlation of one with service performance is mentioned in Section 2.6.

The only other belt design property that has been studied theoretically and experimentally in order to determine a design rule is that of troughability. According to Ellington [60] the minimum level of belt/idler contact for an unloaded belt should be:-

Side rollers	:	One tenth of roller length
Centre roller	:	Point contact

As stated above in Chapter 1, troughability was a problem with all-cotton belts. A test method for this parameter was therefore required and that which was eventually accepted as a standard consisted of hanging a full width belt sample under its own weight as shown in Figure 2.2.6. The ratio of the vertical displacement of the sample centre, F, to the total width of the belt, L, has then to be greater than a given limit which varies according to idler angle. For example, if the idler angle on a particular installation is 35°, the minimum acceptable value of F/L is 0.11. Ellington described tests which showed the relationship between this ratio, F/L, and the maximum permissible troughing angle. He claimed that for F/L less than 0.35 the belt shape could be described by a parabola, and for F/L greater than 0.35 the shape was best described by a catenaric formula. The ratios accepted by the British Standard Institution [26] bear little relation to the theoretically derived ones, but are, from service experience, such as to ensure the minimum acceptable contact between belt and rollers will occur under all conditions.

A more thorough analysis by Belostotskii [61] showed that for adequate troughability,

$$\frac{F}{L} \geq 0.078\ (1 + 0.09/\cos\beta)\ \ell_{n}\ \left((1 + \sin\beta)/1-\sin\beta\right) \qquad (2.2.11)$$

Where β = idler angle ($^{\circ}$)

This was based on the assumption that the shape of the belt sample during an F/L test was that of a parabola, and the work confirmed that the F/L values quoted in the British and International Standards met this requirement. The derivation of this formula, although an interesting exercise, has no real practical purpose for modern belting because the F/L test has become virtually obsolete due to the excellent troughability characteristics of all-synthetic conveyor belts.

One company which has not accepted the F/L test is the Japanese firm Bridgestone Tyre Company Limited. This firm has developed their own test to measure troughability [62] in which a small belt sample is made to form a cantilever as shown in Figure 2.2.7. The deflection of the sample is measured. Bridgestone present a full analysis of this test using inapplicable small deformation simple bending theory. From this they claim to have identified satisfactory minimum deformations which must occur during the test if service troughability is to be adequate. In the same handbook, Bridgestone have also published some experimental results which demonstrate the changes in troughability co-efficient which occur with changes in type of fabric, strength of fabric, thickness of rubber cover, number of plies and temperature. This information forms a useful data base showing how belt stiffness can be affected by these parameters.

The only other rationally derived design rules for conveyor belt constructions result from the comprehensive work carried out by the National Coal Board on all aspects of belt service performance [54]. Field trials (see Section 2.5) established that belt resistance to longitudinal tearing was an important design parameter and that, for the specialised application of belt conveyors in underground coal mines, electrical and fire resistance were required for safety reasons. The National Coal Board then decided to establish a specification for

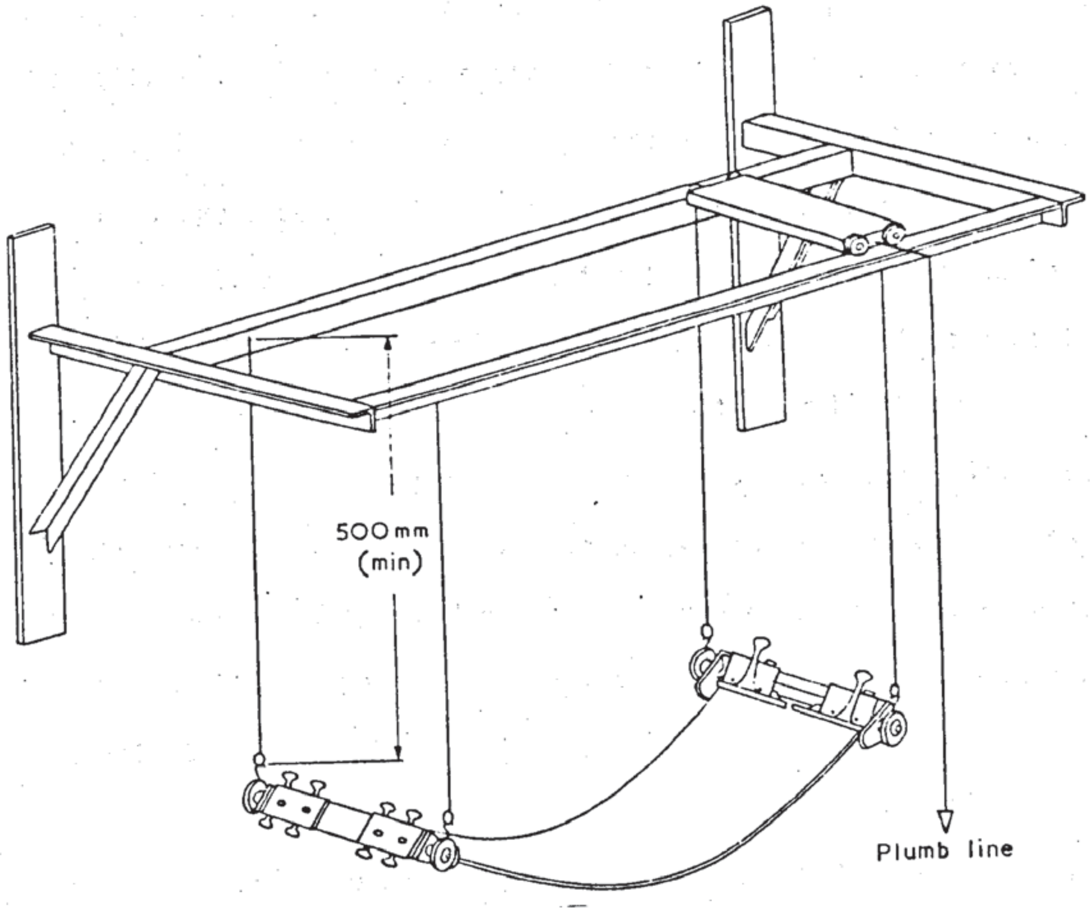


FIGURE 2.2.6

British Standard Institution Troughability Test (F/L Test)

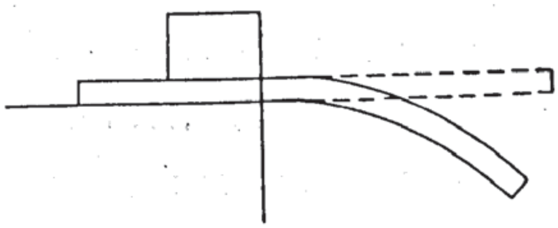


FIGURE 2.2.7

Bridgestone Tyre Company Troughability Test belts to be used in such conditions [63]. This specification (which is still in use) included tests with minimum values to be obtained for the three properties of tear, fire and electrical resistance. The test for fire resistance is particularly relevant to service conditions as it represents a situation that has been known to lead to outbreaks of fire. It involves turning a pulley against a stationary belt sample under tension and the belt must break up before ignition occurs due to the heat generated from friction. Watts [64] describes this test as "an excellent example of design of test methods to simulate end-use performance" and so it represents an ideal situation which should be striven for in all tests concerned with belt design and selection.

The review above has described the only published work available on the derivation of belt selection rules and the assessment of belt design properties. As far as the present author is aware there has been no literature published on the other parameters which appear in the list of belt properties to be considered (see Section 2.1). In particular there are no useful published studies on transverse load support (although Spain [23] has described a test which attempts to reproduce the damage that insufficient transverse load support can cause). This is a very important omission as this property fixes the necessary construction in many cases of belt selection, as will be seen in Section 6.1. It should be noted that a theoretical study of this parameter by Zolutukhina and Kozhushko [65] produced no useful conclusions.

The design of the conveyor structure itself is a technology which tends to be regarded as separate from that of belt design. As mentioned previously, it is unfortunate that this is the case, but the two technologies do, at least, meet in some minor areas in the consideration of transition distances, vertical curves and idler pitches.* The transition distance is the minimum distance required to avoid excessive tension in the belt edges between the flat belt at the horizontal end pulley and the troughed form at the first set of idlers. Vertical curves occur when there are changes in the gradient of the conveyor. Both belt and conveyor properties need to be considered for transition distances and vertical curves. The design rules for them are simple to apply and are described in Section 3.8.

The selection of idler pitch is, however, not so easy to carry out. Kolobov and Koval'chuck [66] have carried out some theoretical work on idler loading and propose that idler pitches should be increased with the increasing belt tension which occurs as the conveyor head end is approached. Thormann [67] has also put forward a case for this, basing his opinion on the fact that as tension increases, belt sag decreases and so belt sag could still be contained within acceptable limits even if idler pitches were increased above normal values. Grimmer [68] has considered the effect of increased idler pitch on the life of roller bearings and demonstrated that several factors must be considered when deciding loading limits for individual idler sets. Shakmeister, Dmitriyev and Miagkov [69] carried this type of study one stage further when they considered several factors to establish the optimum idler pitch. These included effect on (a) power

* See glossary of terms at the beginning of this thesis.

costs, (b) capital costs and (c) maintenance and repair costs; the exercise was valuable in that it made an attempt to optimise several conveyor design considerations. It did not, however, consider such factors as reduction in the belt/conveyor transverse load support capability with increasing idler pitch. (The experimental investigation of the present work - see Chapter 8 - has established this.) The work of Shakmeister and his colleagues has therefore provided a useful example of how several factors must be considered when establishing rules for most aspects of belt and conveyor design and selection. The discussion above has shown that this has not been done for many aspects of belt conveyor design

2.3 Laboratory Testing of Belt Properties

Some of the work described above can, of course, be regarded as laboratory testing. This section is intended to discuss testing procedures that have been (and, in some cases, still are) used on conveyor belts but cannot be regarded as work to establish belt design and selection rules. Some of the work described could be used as a comparative method of determining the resistance of different belt constructions to various damaging factors that will be met in service, but, as such tests have not yet led to belt design rules, they have not been included in the previous section.

British and other national standards specify several small scale tests that must be carried out on belting samples. These include:-

(1) Cover tensile strength.

(2) Cover elongation at break.

(3) Whole-belt tensile strength.

(4) Elongation of whole belt at break.

(5) Elongation of whole belt at working tension.

(6) Ply to ply and cover to ply adhesion.

There are also, of course, acceptable results specified for these tests. Hallam [70] maintained in 1960 that, although, certain of these properties will affect belt performance (see Section 2.6), the test results can only be regarded as quality control checks as they have no proven relationship with service conditions. This, unfortunately, is still the case.

There have, however, been attempts to introduce into conveyor belt standards, tests which more closely show the belt's ability to withstand service conditions. Bulgin [71] has described the study of abrasion resistance tests carried out by several belt manufacturers to see if any of the test methods available would be suitable for inclusion in the British Standard. It was found that none of them gave reproducible results and so the work was discontinued.

Another property which has proved difficult to quantify is that of impact resistance. The field trials of the National Coal Board (described in Section 2.5) showed that it is a very important parameter to consider and various methods of measuring it have been proposed. Hallam, in the article mentioned above, described a simple pendulum machine which would give a single blow to a belt sample and a more elaborate machine which would deliver repeated blows. A more sophisticated pendulum machine devised by Dunlop Physical Research Department [72] was designed to give impacts of various input energies, with different impacting tool shapes, to a sample which could be held under various measurable tensions. The energy returned to the pendulum after impact could also be measured so that a simple calculation gave energy absorbed, but no criterion of acceptability for the results from this was adopted.

The National Coal Board also devised an impact test which allowed a heavy cube-corner to fall on the belt sample [54] . The loss in belt strength after several impacts was then measured. Although results of this test gave good correlation with service experience it was not introduced as part of the National Coal Board published specification.

British Nylon Spinners devised two tests [73] : one allowed a blunt chisel head to strike the belt sample under tension and the number of blows required to penetrate the belt was recorded; in the second test, a small strip of belt was placed on a metal anvil, a bar was allowed to fall on the sample and the strength of unimpacted and impacted belt samples were compared. These tests were used to compare different belt constructions at the time when synthetic fabrics were being introduced. A recent article by Ezhova [74] shows that such tests are still used to provide comparisons between different types of belting.

He illustrates the impact resistance of a single ply belt of a certain fabric as comparable to that of 4 ply belts of one other type of fabric and 6 ply belts of another but does not attempt to define "acceptable" results. Thus there is still no absolute standard by which impact resistance can be measured. The apparent lack of interest shown in this by manufacturers is probably because it no longer represents a problem with present day belt constructions but this does not mean that any future belt developments will not require such a standard.

A great many properties of belt cover compounds in addition to those listed at the beginning of this section can be measured and most of these have been considered during attempts to correlate laboratory tests with service performance. (Any proven correlations are described in Section 2.6.) Hannoyer [75] studied compound hardness, density, tensile strength, extension at break, resistance to tear and resistance to abrasion. Similar properties have been studied by the Malaysian Rubber Producers' Research Association [76, 77, 78], and this organisation also considered property changes after oven ageing at 100°C. (Oven ageing is an accepted method of accelerating property changes in rubber compounds.) Spain [23] has explained that weathering, oil and chemical resistance can also be easily evaluated in the laboratory.

Flexural rigidity is a "whole-belt" property that has been studied in some detail. This has been done by the National Coal Board [54] because it was found that belt flexibility had a considerable effect upon conveyor driving characteristics. Several methods of measuring belt flexibility were examined, but none were introduced into any belting standard or specification. The National Coal Board do, however, still test the flexibility of all belts to be used underground even though they do not yet publish any limits that have to be met. Belt stiffness proved to be an important belt characteristic in the provision of transverse load support during the experimental investigation in the present thesis, (Chapters 7 and 8).

Two standard test methods to establish the efficiency of mechanical joints in conveyor belting are available [79] ; one is static and involves a straightforward tensile test, whilst the other is dynamic and involves flexing the joint quickly around several pulleys of small diameter and thus attempts to accelerate service conditions. Unfortunately, although the methods of carrying out the test and reporting the results are described in the standard, acceptable results are not proposed and so the test can only give comparative results and does not provide an absolute standard that must be attained.

One other piece of literature should also be mentioned. Bobeth and Heger [80] have studied the effect of changes in the manufacturing conditions on the laboratory stress-strain characteristics of the final product. Their research demonstrated that it is not only raw material properties that affect final whole-belt properties.

The discussion above has shown that many cover and carcase test properties can be (and have been) measured in laboratory tests. It is likely that some of these properties will affect the final performance of the belt in service. How they will do this cannot be determined unless a great deal of information from routine service and controlled field trials is available. The next two sections describe some of the work that has already been done in these areas.

2.4 Service Information

Some of the work described in previous sections could be regarded as the collection of service information. For example, Grierson carried out his power consumption studies on working conveyors. However, for the present work, the term service information is used to describe any information that is obtained about working conveyors that are not being used for specific studies. Thus, for example, details of complaints from belt users about inadequate performance would provide a contribution to service information.

There is very little literature published on this fairly general aspect of belt conveying. It is, however, very important because collection of service information could eventually lead to a detailed knowledge of the performance characteristics of different belt constructions operating under various conditions. In addition a complete picture of the belt conveying market could be built up and thus enable an optimum range of belt constructions fulfilling market requirements to be devised. (See Section 6.3.)

One example of the use of service information has been presented by Wilcox, Robertson, Tideswell and Jones [81]. These researchers studied the causes of seventy belt fires occurring underground between 1940 and 1948. The study resulted in recommendations for changes in belt machinery that helped to decrease the risk of further fires and thus the work demonstrated the value of studying service information to improve techniques of belt usage.

Unfortunately, the work described above would appear to be the only published use of service information. Hanson [82], however, has described data sheets which would form a basis for logical collection of such information. The sheets are mainly intended for details of conveyor equipment but Dunlop Australia [83] has issued similar sheets more concerned with belt performance details. If these were used in conjunction with the specification sheets used by various manufacturers in their belt selection procedures, they would provide an excellent method of obtaining details of the conveyor belting market. Further suggestions for the type of data that should be collected from service are discussed in Chapter 4.

2.5 Field Trials

A field trial is an exercise carried out to evaluate quantitatively the performance of a belt on a conveyor installation. If such an exercise is to be of any real value, thorough details of the operating conditions of the belt are required and so full records of running hours, tonnage carried and belt failures must be kept. The National Coal Board demonstrated how a large scale operation of this type including several belts could be set up. This has been described in the report "Field Trials of Conveyor Belting" [84]. The trials were started as part of the development of fire resistant belting and the information collected from them was used for four purposes:-

(a) To compare the wearing properties of the different types of plastic belting and to compare them with those of rubber belting;

(b) To investigate the causes of the removal of belting so as to assist manufacturers in overcoming the defects in non-inflammable materials;

(c) To compare the performance of belting under different conveying conditions and thus, perhaps, to suggest improvements in conveying practice;

(d) To obtain data for use in developing laboratory tests so that the National Coal Board might introduce consumer testing of conveyor belting.

One difficulty was that of the long time that could elapse before all the original belt on an installation was replaced. This meant that belt life, the absolute assessment of belt performance, had to be predicted by analytic means if conclusions from the trials could be drawn within a reasonable time period. The methods used to do this have been described by Norvall [85] and Hems [86], and consisted basically of extrapolating a curve of length of original belt remaining against time in service so that a time when, theoretically, no original belt would remain was obtained. The need for this is caused by the fact that belting, although a continuous band, behaves, in practice, as though it were made up of a number of elements of length, each element giving a different period of service. This is because when a belt is damaged only the damaged area need be removed and a new piece of the same or of a compatible construction is simply spliced into the original. As random damage often causes failure, the new piece of belting need not remain in service for the same time as its predecessor, a process shown diagrammatically in Figure 2.5.1.

In addition to the field trials, the National Coal Board carried out a laboratory examination of the belt types used. Any correlation found between the test results and service is discussed in the next section of the present thesis. Several of the laboratory tests were also carried out on pieces of belt removed during the trials thus providing useful knowledge of the changes that occur in belting during service. Vulkan [87] has described this work in relation to the following properties:-

(a) Co-efficient of friction.

(b) Tensile strength.

(c) Ply adhesion.

(d) Electrical resistance.

(e) Flexural rigidity.

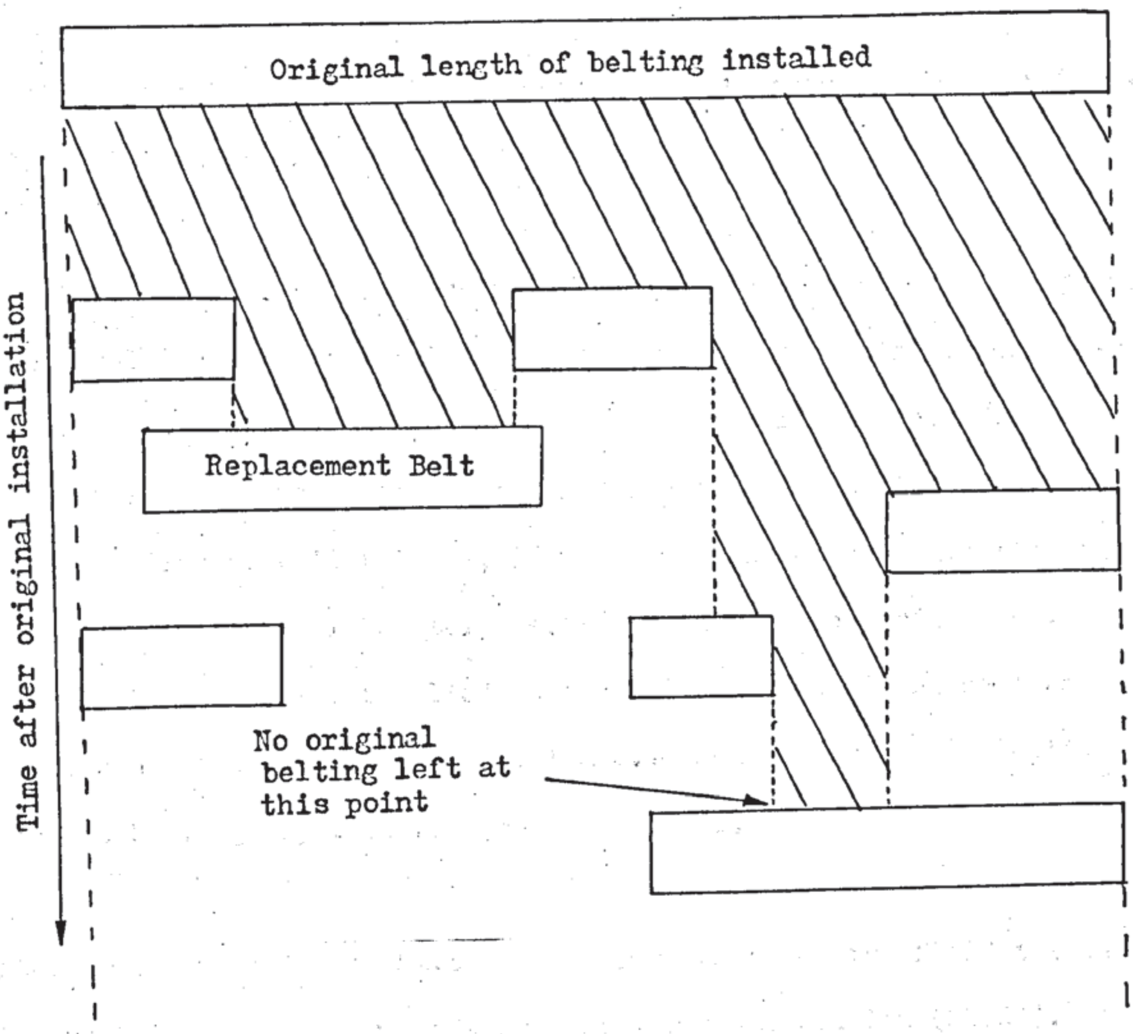


FIGURE 2.5.1

Diagrammatic representation of
discontinuous removal of
belting from an installation

Some of these properties feature in belt selection procedures, so any changes that occur in them must be allowed for in the data used. It would appear that Vulkan is the only worker to have published information of this kind.

The National Coal Board exercise, although admittedly benefitting from the availability of several conveyors working under similar conditions, demonstrated that many aspects of belt performance could be studied fully by using field trials. The trials resulted in a better understanding of belt failure, improvements in conveyor practice and some correlation of laboratory testing with service performance. This last achievement resulted in the National Coal Board setting up their own specification (see Section 2.1) and an "Approvals Section" to test belting supplied for use underground.

Another organisation to have carried out comprehensive field trials is the Malaysian Rubber Producers' Research Association (MRPRA). In particular, this association has studied cover compounds. Three trials were specifically designed to evaluate (1) compounds for carrying cold materials [76], (2) natural rubber and synthetic rubber in heat-resistant belting [77] and (3) the effect of antioxidants on wear [78]. In each case the belts tested had several different compounds vulcanised onto a common carcase and the wear resistance of each was assessed by periodically measuring the belt thickness during service. The results were recorded graphically and it appeared that the rate of wear decreased with time. It was thought that this could be due to slight structural changes in the belt carcase causing an increase in measured thickness: average rates of wear were therefore calculated giving greater weight to early measurements. Tensile strength, elongation at break, tension stress at 300% elongation, hardness and tear resistance of the compounds were measured in the laboratory, but no attempts to correlate service wear with these propertios were made.

Similar work to that of the MRPRA has been reported by Hannoyer [75] who also carried out field trials in which several cover compounds were vulcanised to a common carcase. Belt thickness was measured before and after the trial at twenty-four different points in each compound zone. A special template was made to ensure that the position of each point was exactly the same for both sets of results. Laboratory evaluation of compound hardness, density, tensile strength, elongation at break, resistance to tear and resistance to abrasion was made and a study to find any correlation between these properties and service performance was carried out. The results of this are given in the next section.

More recently, the Russian researchers Polunin and Gulenko [88] have published details of field trials to establish the mechanism of belt cover wear, but no details of how the test results were measured are given in the article and no useful conclusions about how compounds could be improved resulted.

It can be seen from the discussion above that the National Coal Board is the only organisation to have studied the complete mechanism of belt performance and ultimate failure by using field trials. Other work has only been concerned with fairly specialised aspects of belt performance. The field trial work of the Dunlop belting organisation is discussed in Section 4.3.

2.6 Correlation of Laboratory Testing and Service Information

As might be expected from the thoroughness of their field trials procedure, the National Coal Board continued their studies to establish laboratory tests which would correlate with belt service performance characteristics. The most obvious result of this work was the introduction of the drum-friction test into a specification for fire-resistant conveyor belting [63]. As mentioned previously, this represented an example of the design of a test to reproduce end-use performance. Of more general use were the tests to establish a belt's ability to withstand random incidental damage: it had been found from the field trials that belting was most often removed from a conveyor because of this sort of damage rather than because it had worn out. (This fact means that the relevance of some of the field trials discussed in the previous section is questionable.) It was found that most occurences of incidental damage could be categorised into the following forms [55] :-

(1) Damage inflicted by impact; for example, from falling material at transfer points.

(2) Damage to the edge, due to the belt rubbing against or catching in parts of the conveyor framework or obstructions of various kinds, causing narrowing and sometimes tearing of the belt.

(3) Damage resulting from the belt folding on itself transversely usually because of the belt moving off centre

and pushing against the side of the conveyor structure. This form of damage was known as a "turnover".

Tests were devised which reproduced each of these forms of damage. An impact test (see Section 2.4) gave results correlating extremely well with the operational trials and a machine designed to test resistance to edgewear gave fairly good correlation with service. It was found that a belt's resistance to turnover damage was dependent on its physical properties, and so it was considered that no new form of test appeared to be necessary for this property. The main properties found to affect turnover damage and, to a lesser extent, edgewear were weft elongation at break and weft tensile strength. This was the first proven correlation between the results of laboratory small scale test methods and service performance. A full report of this has been given by Weinberg [89].

No other correlation between belt physical properties and belt performance have been found, although Hallam [70] in 1960 listed some service factors and the belt properties that might influence them. These were:-

(a) Power requirements:-

(1) Warp strength.

(2) Warp elongation at working load.

(3) Co-efficient of friction.

(4) Longitudinal flexibility.

(5) Weft strength
(6) Weft elongation at break } If mechanical fasteners are used.

(b) Troughability and load support:-

(1) Lateral flexibility.

(2) Weft elongation at break.

(c) Flexing around pulleys:-

(1) Longitudinal flexibility.

(2) Adhesion, cover to ply and ply to ply.

(d) Belt stretch:-

(1) Warp elongation at working load.

(2) Warp elongation at break.

(3) Fatigue.

(e) Ageing:-

(1) Oven ageing - tensile and elongation losses on covers.

(2) Oven ageing - loss in ply adhesion.

(f) Abrasion:-

(1) Cover compound abrasion test.

(g) Mechanical Damage:-

(1) Warp tensile strength.

(2) Warp elongation at break.

(3) Weft tensile strength.

(4) Weft elongation at break.

(5) Longitudinal tear strength.

(h) Impact:-

(1) Warp tensile strength.

(2) Warp elongation at break.

(3) Weft elongation at break.

(4) Fabric construction.

(5) Ply insulation.

(6) Cover.

Hallam pointed out that several of the property tests mentioned above had not been established. These were:-

Along the belts:-

Longitudinal flexibility.

Flexing fatigue.

Impact resistance.

Across the belt:-

Maximum lateral flexibility (load support).

Covers:-

Cover compound abrasion.

Co-efficient of friction.

It is interesting to note that these tests have still not been introduced into the main international standards. It would appear that no work carried out has yet established the influence of any of these belt properties on service performance, and so even the tests that are established can still only be regarded as quality control tests. There have, however, been attempts to combine the results of several small scale tests into one "belt performance index" carried out in the U. S. A. and reported to the International Standards Organisation in 1964 [90]. The index took into consideration the stress/strain properties of the cover compounds as well as abrasion resistance characteristics. It was claimed that this index correctly graded the surface capabilities of belt cover materials. It was decided that the abrasion test used was not reproducible and so the index was rejected but it was resolved that the ISO working committee "considers it necessary to carry out studies on the <u>correlation</u> between the wear of belt covers in <u>actual</u> service and the results of laboratory tests which may relate to such wear resistance (if possible based on a single test or if not possible, on a formula combining the results of a minimum number of tests)."

Although some researchers, as mentioned above, have studied the mechanism of belt wear in field trials, no positive correlation between belt properties and wear has yet been achieved. Hannoyer [75] concluded from his work that there is no correlation between service life and compound tensile strength, elongation at break or resistance to tear. He did find that some correlation exists between length of service and abrasion resistance but his main conclusions were that "the comparison of service life and laboratory characteristics of conveyor belts is somewhat negative, in the sense that the laboratory tests most often used do not give the users an idea of the service life compared to other compounds" and "the index of abrasion appears to be the only one to show a partial correlation with service life, but it does not give enough certainty for control tests".

Polunin and Gulenko [88] have claimed to have established correlation both experimentally and theoretically, between belt cover wear and certain cover properties. These include modulus and co-efficient of friction. The conclusions from the work, given here in full, that "wear of belt covers is mainly by a friction contact fatigue mechanism and testing of cover compounds should be carried out on a metal mesh with specific geometric parameters and on a test bench" show that the results were not too significant.

The only other published details of service/laboratory correlation are given by Spain [23]. He has claimed that flex life of a belt under actual service conditions can be estimated by applying the following formula:-

$$\frac{\text{Flex life}_x}{\text{Flex life}_a} = \left(\frac{D_x}{D_a}\right)^{5.35} \left(\frac{T_a}{T_x}\right)^{4.12} \left(\frac{V_a}{V_x}\right)^{0.5} \left(\frac{t_a}{t_x}\right)^{6.27} \left(\frac{L_x}{L_a}\right) \qquad (2.6.1)$$

Where:-

D	= Pulley Diameter			
T	= Belt Tension			
V	= Belt Speed		x	= Desired conditions
t	= Belt Thickness		a	= Test conditions
L	= Belt Length			

This formula was derived from the results of a large number of tests involving controlled variations of each of the influencing factors. Spain pointed out that the primary use of this formula is for power transmission belt and its projection to conveyor conditions usually indicates flex life to be of little concern because the belt will, in practice, be removed from service for one or more of a variety of other reasons.

This demonstrates that, apart from the work of the National Coal Board, very little correlation has been achieved between belt service performance and laboratory tests to establish the belt's resistance to the factors that will ultimately bring about its failure.

2.7 Conclusions from Scientific Literature Review

It can be seen from the above review that there has been considerable time and effort devoted to studying certain aspects of conveyor belt design and selection. Unfortunately much of this work was carried out approximately twenty years ago and significant changes in belting materials and conveyor structures have occurred since. The work has not been repeated to see if the results are still valid for modern practice. In any case, the need for this repetition of earlier work is not as urgent as the need for original studies of design parameters that were previously of no concern. Evidence accumulated during the present investigation shows that the most pressing need is for an investigation into the provision of the necessary transverse load support in a belt. It would appear that no experimental work whatsoever has been published on this design property and yet, as shown in Section 6.1, it fixes the final outcome of the majority of belt selections and its satisfaction causes overspecification of other design properties. It is generally accepted that belt lateral stiffness plays a major role in the provision of transverse load support, but there is still no standard test for this, even though there is an urgent requirement for one. This is shown by a recent request for such a test from the Mechanical Handling Engineers' Association to the British Rubber Manufacturers' Association to which the reply was [91] :-

"The flat belting technical committee of the B. R. M. A. have carefully considered the M. H. E. A. request for a test method to determine the maximum flexibility of conveyor belting. Whilst it is realised that in British Standard 490 : Part 1 : 1972 the test provided for in clause 16 (the F/L test) determines the minimum acceptable flexibility for suitable troughing and that the note appended thereto states the maximum flexibility may require consideration, nevertheless the Committee have been unable to arrive at a test method to quantify this parameter. Certain members use various methods to fix arbitrary limits for their own constructions, but it has proved impossible to evolve a method acceptable to all constructions.

We would therefore suggest that the note in BS.490 : Part 1 : 1972 is amended by deleting the last sentence and replacing it by:-

Whilst the above minimum values ensure that the belt will confirm satisfactorily to the troughing idlers, it has not been possible to fix a "maximum" value as determined by this particular test. Consideration should therefore be given to the belt selected to ensure that it has sufficient transverse rigidity to resist undue sagging and/or trapping in the junction of the idler rollers. This information can be obtained from the belting manufacturers".

The National Coal Board has demonstrated the method by which such a test could be evolved. Field trials and service information could be used to show the transverse load support requirement of particular load/conveyor combinations. A service related laboratory small scale test should then be devised so that the load support capability of any belt construction can be assessed quickly. As field trials are expensive and lengthy it is sometimes necessary to reproduce service conditions as nearly as possible on a large scale test rig. A series of controlled experiments carried out on such a rig to study the factors affecting load support is described in Chapters 7 and 8 of the present thesis.

The fact that one particular design criterion is most important at present does not mean that it should be studied at the total expense of all the others. Instead, constant work on the interaction of the various belt and conveyor design parameters should be carried out and belting manufacturers should learn, from previous experience, that future developments might mean that other design properties become important.

Service information on all aspects of belt behaviour should be collected and if any field trials are carried out the opportunity should be taken to study the changes that occur in belt properties during service. Such information should eventually lead to a series of criteria by which it is possible to judge any present or future belt construction's resistance to the damaging factors that it will meet in operation. This is, of course, one of the requirements of the "ideal" belt design and selection procedure discussed in Chapter 1.

CHAPTER 3

COMPARATIVE STUDY OF BELT SELECTION PROCEDURES OF VARIOUS ORGANISATIONS

The general design parameters which need to be considered in a belt selection procedure have been outlined in Section 1.2; and a full list of them compiled by Grant [34] has been given in Section 2.1. In this chapter the belt selection procedures of various organisations are reviewed to show how they ensure that these required design properties are present in the belt construction finally chosen for the task in hand.

It has been mentioned in Chapter 2 that certain of the design properties in Grant's comprehensive list are not incorporated into belt manufacturers' published selection procedures because they are present in all the types of belt construction available or because the manufacturer has no means of quantifying a particular property requirement for a given conveying situation. The necessary belt properties that <u>do</u> feature in the selection procedures are that the belt must have:-

(a) Sufficient width to carry the load at the running speed of the conveyor.

(b) Sufficient tensile strength to withstand the tension occurring in the working belt.

(c) Sufficient transverse load support to prevent the belt from catching in the idler gap.

(d) Sufficient lateral flexibility to provide adequate troughability.

(e) Sufficient longitudinal flexibility to pass around the conveyor end pulleys.

(f) Sufficient impact resistance.

(g) "Elastic modulus" compatible with the requirements of the conveyor structure.

(h) Permanent elongation characteristics compatible with the requirements of the conveyor.

(i) Suitable covers of the correct type to withstand abrasion and any other damaging factors.

Not all the manufacturers necessarily consider all of these properties - as will be seen below - and they do not always agree about how the property requirement is assessed. However, before any part of a belt selection procedure is carried out, details of the task being considered must obviously be collected. The next section describes how the various manufacturers do this.

3.1 Collection of Conveyor Details from Customer

Information from customers about the proposed or existing conveying system is usually collected in the form of questionnaires. If the belt required is to be a replacement belt then certain conveyor parameters such as width, idler pitch and pulley diameters are fixed and so the final belt selection must conform to these, but if the belt is to be installed on a proposed conveyor there should be scope for changing such parameters until an optimum combination of belt and conveyor is obtained. (This seems to be rarely done in practice.) In either case, the essential information to be supplied by the customer is:-

(1) Width of belt.

(2) Conveyor centres length*.

(3) Belt Speed.

(4) Peak tonnage rate of material carried.

(5) Type of material carried:-
Density.
Fines, lumps or mixed.
Size of largest piece.
Condition:- hot or cold, dry, wet or oily.

* See glossary of terms at the beginning of this thesis.

(6) Inclination of conveyor.

Difference in height between conveyor ends.

(7) Type of drive:-

Single pulley, snubbed single pulley, tandem drive etc., Whether drive drum is lagged* or bare.

(8) Horse power of driving motor.

(9) Pulley diameters:-

Drive pulley, snub pulley*, tail pulley.

(10) Type of take-up:- screw* or gravity*.

(11) Method of joining belt:-

Vulcanised splice or mechanical fasteners.

Other data can also be useful, and most of the questionnaires supplied by belt manufacturers require extra information such as details of the idlers used. The most comprehensive questionnaire form is that of BTR Belting Limited (see Figure 3.1.1). It is interesting to note that this particular form asks for details of the previous belt and the reasons for its failure and thus also performs the useful purpose of collecting service information about belt performance. The BTR Enquiry Form also requests a contour sketch of the conveyor on its reverse.

*See glossary of terms at the beginning of this thesis.

DATE .... SUBMITTED BY ....

SOURCE OF ENQUIRY ....

SITE .... CONVEYOR IDENTITY ....

* INFORMATION REQUIRED

Belt width .... mm

Pulley centres .... m

Lift............m Slope in degrees...........

Fall............m Belt speed..............m/s

TONNES PER HOUR: Peak...........Ave..............

* MATERIAL TO BE CARRIED

....

Weight per m^3 ....tonnes

SIZE: Max.........Ave............Fines...........%

* CONDITION

Oily Wet Dry Sticky

TEMPERATURE OF MATERIAL: Max.... °C Ambient..... °C

Chemical Action..............................

(If in doubt send sample)........... ....

* TYPE OF DRIVE

Single Drum/Geared Tandem/Separate motors:- Give detailed sketch overleaf showing location, angle of wrap on each pulley, whether lagged and distribution of horsepower.

Angle of wrap ....°

LOCATION: Head Foot Central

DRIVING DRUM: Lagged Bare

* TYPE OF TAKE UP

Screw Gravity

Movement...........m Take-up-weight........kg

LOCATION: Behind drive Tail Central

* TYPE OF JOINT

Metal fasteners

Vulcanised

DIAMETER OF DRUMS

Driving...............mm

HEAD ....mm Tail................. mm

H. T. Snub and Bend.............mm

L. T. Snub and Bend.............mm

* ANSWER ESSENTIAL

MOTOR DETAILS Power ....

Type ....

Method of starting ....

Break equipment ....

FLUID COUPLING: Traction type Scoop type Coupling not fitted

METHOD OF FEEDING AND DISCHARGING

Fall of material .... m

Leads from ....

Discharges to.........................

IMPACT IDLERS FITTED: Yes No

IDLERS

Troughing angle...........Distance between centres..........m

Roll diameter........................ mm

TYPE: Flat Troughed 3 roll 5 roll

Suspended 3 roll with offset centre roll

Plain bearing Ball Bearings High grade anti-friction

WEIGHT OF MOVING PARTS

of idlers, Pulleys, etc., turned by belt per metre conveyor.......................................

PROTECTION

CONVEYOR UNIT IS: Covered Semi covered Uncovered

DETAILS OF EXISTING BELT

Manufacturer............. Grade................

No. of Plies.............. Fabric ref............

Cover thickness........... Top....... Back.......

Length of service......... Total tonnage.........

CAUSE OF FAILURE: Abrasion Fabric stretch Impact

CALCULATIONS

Estimated belt power requirement ....

Effective tension (T_e) ....

Slack side (T2) ....

Slope tension.....................................

Max: Tight side tension (T_m) ....

Estimated gravity take-up weight.................

FIGURE 3.1.1

Conveyor enquiry form by BTR Belting Limited

Having collected this basic conveyor information it is possible for the belt design engineer to carry out the belt selection procedure as described in one of the manufacturers' manual. (Some of the manuals, such as that published by Dunlop [92], are designed so that the belt user himself can follow the written procedure and select the correct belt construction.) The methods by which the various manufacturers' belt selection procedure allow for the design properties listed at the beginning of this chapter are reviewed in the following sections.

3.2 Calculation to ensure that belt has sufficient capacity

Having collected the above information from the belt user it is necessary initially to ensure that the combination of belt width, belt speed and conveyor capacity represent a practical situation.

The first consideration to ensure that a belt of the width considered will be capable of carrying the load is that of maximum lump size. All manufacturers admit to using a "rule-of-thumb" approach to find the maximum permissible lump size that should be carried on a belt. Dunlop [92] suggest that when the load material is unsized (that is, when all sizes of material are present and the largest lumps only make up approximately 10% of the load), the belt must be at least three times the width of the maximum lumps. If the load is of uniform size, then the permissible lump size becomes smaller. Liggins [9] of Apex Belting Company suggests that the minimum belt width should be taken not as three, but as four times the maximum lump size, and when the load is unsized the belt width can be reduced to twice the largest lump size. Other manufacturers such as BTR [93] tabulate this information (see Table 3.2.1) and disagree with both Dunlop and Liggins. It can be seen that the differences in permissible maximum lump sizes at a particular belt width are considerable. This is not a particularly important design property, but it would be a relatively easy matter to conduct an investigation to establish a valid empirical relation for it.

A more important consideration concerning belt width is that of ensuring that the belt is capable of carrying the maximum conveyor capacity. The belt capacity, T, in tonnes per hour is given by:-

$$T = A \times V \times \rho \times 3600 \qquad (3.2.1)$$

Where A = Load cross-section (m^2)

V = Belt speed (m/s)

ρ = Material density (t/m^3)

With a replacement belt the belt speed is normally fixed and so it is not necessary to find a suitable value for this but only to ensure that the maximum recommended belt speed for the material carried is not exceeded. Most belt manufacturers tabulate this information according to belt width and material type and there is very little difference between the figures from the various manufacturers with no major revisions of them occurring in recent years. The tabulated information is based mainly on experience although sometimes the basic equations of motion are used to ensure that the chosen belt speed will provide a suitable trajectory of material discharge [30] .

Some of the methods available for determining another of the variables in equation 3.2.1, the load cross-sectional area, have been discussed in Section 2.2. This parameter depends greatly on the belt width and, as shown previously, some researchers have derived fairly complicated formulae to show this dependence. Most manufacturers tabulate cross-sectional area data without quoting formulae from which the figures were calculated. One exception, BTR [93] provides an equation in which the "edge-margin"* appears. Precise values for this are quoted according to the idler angle (see Table 3.2.2), but this would appear to be rather unnecessary because the load cross-sectional area calculation is rarely critical. It has been explained in Section 2.2 that simple empirically based formulae provide the most suitable methods of calculating load area and one manufacturer, Fenner [94], quotes such a formula in a belt selection manual.

Once it has been checked that the combination of belt width, belt speed and conveyor capacity supplied by the customer conforms to a practical situation, it is possible to continue with selection of the correct belt type to be used, as described below.

3.3 Calculation of Belt Tensile Strength Requirement

The different available methods of calculating belt tensile strength and conveyor horsepower requirements have been the subject of a doctorate thesis by Thormann [50] who demonstrated that none of them could be considered to be completely accurate and divided them into three categories:-

*See glossary of terms at the beginning of this thesis.

(1) Those using the basic law of friction; Frictional force = normal force x co-efficient of friction.

(2) Those using formulae, diagrams and tables derived empirically from measured results.

(3) Those involving the determination and summation of all the separate resistances to motion occurring on a conveyor. (Behrens [28] carried out experiments to demonstrate the magnitude of these resistances.)

The methods used by belting manufacturers can be regarded as falling mainly into the second category. As described in Chapter 2, the experiments on which the empirical formulae are based were carried out approximately twenty years ago and, although significant changes in belt conveying practice have occurred since, there have been no recent attempts to establish new ones. It is quite likely that the tension formulae used only prove adequate because of the large safety factors built into the procedures and the overspecification of belt tensile strength which occurs due to the provision of other design requirements. (See below.)

The formulae used by all the manufacturers are basically the same, although various minor features are built into each one to distinguish it from the others. As this is the case, the most logical criterion to use in the choice of tension calculation technique is simplicity of handling. Those of Uniroyal [95] and the Conveyor Equipment Manufacturers' Association [31] have a large number of calculation factors appearing in them and are probably intended more for the

Belt Width (mm)	Maximum Lump Size (mm)	
	Uniform Lump Size	If mixed with 90% of uniform sized material
300	50	100
450	100	150
600	125	200
750	175	275
900	200	350
1050	250	425

TABLE 3.2.1

Maximum lump size data presented by B T R

Troughing Angle	Edge Margin (mm)
20°	$0.047\ b + 32$
$27\frac{1}{2}^{\circ}$	$0.045\ b + 27$
35°	$0.042\ b + 25$
45°	$0.037\ b + 22$
53°	$0.033\ b + 20$

TABLE 3.2.2

Edge margin data presented by B T R

NOTE:- b = belt width (mm)

conveyor engineer than for the belt user with little experience of belt selection. The procedures of all the other major belt manufacturers can be regarded as equally straightforward to use. The Dunlop procedure is described here as a basis for discussion, but the imperfections of this are typical of those of other companies such as Boston Woven Hose [96], BTR [97], Fenner [94], B. F. Goodrich [98] and Goodyear [99].

The first step in the Dunlop belt tension calculation is to find the effective tension, T_e, transmitted by the motor when the belt passes around the drive pulley.

$$T_e = T_a + T_b + T_c \tag{3.3.1}$$

Where

T_a = tension component required to move the empty belt.

T_b = tension component required to move the load.

T_c = tension component to raise or lower the load. (If the load is lowered, then T_c will be negative.)

The various tension components are obtained by the following formulae:-

$$T_a = F_B \times L_1 \times P \times 9.8 \text{ Newtons} \tag{3.3.2}$$

$$T_b = F_L \times L_1 \times M \times 9.8 \text{ Newtons} \tag{3.3.3}$$

$$T_c = \pm M \times H \times 9.8 \text{ Newtons} \tag{3.3.4}$$

Where F_B = Friction factor for empty belt.

F_L = Friction factor for loaded belt.

L_1 = Conveyor centres length (L) plus an additional length (L_A) to allow for the frictional effects of the end pulleys (m).

P = Weight of moving parts per metre of conveyor (kg/m).

M = Load weight per metre of conveyor (kg/m). M can be calculated from the conveyor tonnage rate and the belt speed.

H = Conveyor lift or fall (m).

F_B, F_L, L_A and P are obtained from tables in the Dunlop manual and are originally based on experience with working conveyors. The tables for the calculation factors are reproduced in Appendix 5. From these tables it can be seen that in the Dunlop procedure F_B is usually equal to F_L and, for "average conditions", is quoted as 0.03. BTR claim that different values should be used for the two friction factors because "tests on actual conveyor systems indicate that greatest accuracy can be obtained in tension calculations by using two co-efficients". It would appear, however, that no details of the tests used have been published.

Some manufacturers (for example, Fenner and Goodyear) in addition to quoting formulae, simplify the effective tension calculation by tabulating values of its three components for typical conveyor dimensions.

Once the effective tension has been found it is a simple matter to find the power required to drive the conveyor from equation 2.2.6. If this equation is expanded the following is obtained:-

$$\text{Power consumption (kW)} = \frac{V}{1000}(T_a + T_b + T_c) \qquad (3.3.5)$$

and so, for a horizontal conveyor where T_c is zero, by substituting for T_a and T_b,

$$\text{Power consumption} = \frac{9.8V}{1000} \times (L + L_A)\ (F_B \times P + F_L \times M) \qquad (3.3.6)$$

For a given conveyor running at constant loading, V, L_A, P, F_B, F_L and M are fixed and so the power consumption is linearly dependent on the conveyor centres length, L. In addition to this length dependent component there is an additional constant component due to L_A. This is called the "terminal friction" component and is claimed to allow for the power consumed by frictional forces at the pulleys at each end of the conveyor, the take-up pulley and any other machine parts common to every conveyor. The power consumption formula therefore follows the relationship found by Grierson (see Section 2.2), but this does not, of course, mean that the various numerical factors used in it necessarily are correct.

Having calculated the effective tension, the next step in the Dunlop tension calculation is to find the minimum tension, T_2, which must be induced into the belt as it leaves the drive pulley in order to ensure that the effective tension can be transmitted without belt slip.

$$T_2 = T_e \ (K - 1) \tag{3.3.7}$$

Where K = Drive factor.

Comparing equation 3.3.7 with the basic drive formulae given in equations 2.2.7 and 2.2.8 shows that:-

$$K = \frac{e^{\mu\theta}}{e^{\mu\theta} - 1} \tag{3.3.8}$$

Thus K, the drive factor, depends on μ, the co-efficient of friction between belt and pulley, and θ, the arc of contact between belt and pulley (in radians).

It is at this point that confusion can arise between the different manufacturers' tension calculation methods. Some define the drive factor slightly differently and so great care must be taken if, for any reason, a combination of two different techniques is being used.

All the manufacturers present a table of values based on measured co-efficients of friction for the drive factor for various angles of wrap, θ, and for lagged or unlagged pulley surfaces. (See Table A5.8 in Appendix 5 for an example.) This implies that the co-efficient of friction has already been taken into account. However, this property can vary greatly with working conditions [83] and so, in the opinion of the present author, it would be better to quote a set of drive factors to be used according to whether the conveyor is to be running in wet or dry situations. Dunlop and several other manufacturers quote different drive factors for use with screw take-ups and gravity take-ups; those for screw take-ups are slightly higher to allow for the extra tension that might be induced in a belt because of the inherent difficulty of measuring the amount of tension actually applied with such a device.

The drive factor is a fairly important feature of belt tension calculations but none of the manufacturers appear to treat it as such. Some experimental work on the service values of the drive factor under various conditions would be extremely useful.

The loaded belt strand tension, T_1, is easily found by summing T_e and T_2. Often this value of T_1 will be the eventual maximum tension that will theoretically occur in the belt but, in some cases, it is found that the tension T_2 has to be increased to limit the inter-idler sag when the belt is loaded. This causes a corresponding increase in the loaded strand tension and so T_1 may be exceeded. The method by which the final maximum tension is calculated depends on the conveyor configuration. As more than ninety percent of all conveyors are of simple horizontal or elevating form, it is this configuration that is used as an example for the rest of the work in this section.

The tension, T_o, required to limit sag to an acceptable level is given by:-

$$T_o = G \times (B + M) \times I \times 9.8 \text{ Newtons} \qquad (3.3.9)$$

Where G = Sag factor.

B = Approximate belt weight (kg/m) - available from tables.

This is based on equation 2.2.10. G is simply a dimensionless factor which depends on the required amount of sag; if the actual sag is to be 2% of the idler pitch **G** will be, by comparison of equations 2.2.10 and 3.3.9, equal to 100/16, that is 6.25. The permissible amount of belt sag is considered to be 2% by most manufacturers, but one allows up to 3% [96] and others only accept 1.5%. Limiting belt sag between the idlers does not affect the effective tension according to the calculation methods of all the manufacturers and so neither does it affect the power consumption. However, research work by Behrens, as described in Section 2.2, has shown that power requirements are very much dependent on this parameter. As the acceptable sag levels appear to be arbitrary, some research work would be useful to find optimum sag levels which do not allow material spillage and minimise power consumption.

It is necessary to check that this tension, T_o, is present at the tail end of the conveyor by taking T_o as a starting point and allowing for the effects of "belt incline tension" and "return strand friction tension". The belt incline tension is that tension component required to move the belt vertically between the tail and head of the conveyor and is given by:-

$$\text{Belt incline tension} = B \times H \times 9.8 \text{ Newtons} \qquad (3.3.10)$$

The return side friction tension, RSF, is that tension component required to move the empty belt along the return strand of the conveyor. As the frequency and weight of rollers on this strand are far less than on the loaded side, the return side friction is estimated as forty percent of T_a and so:-

$$RSF = 0.4 \times T_a \text{ Newtons} \qquad (3.3.11)$$

It is now possible to find a value for the tension that must be present in the belt as it comes off the drive drum in order to limit sag to an acceptable level. This alternative slack side tension to T_2 is given by $T_o + B \times H \times 9.8 - RSF$. It is then possible to find two alternative values of maximum tension.

These are:-

$$T_m (a) = T_e + T_2 = T_e + (K-1) = KT_e \text{ (Newtons)} \qquad (3.3.12)$$

and

$$T_m (b) = T_e + T_o + B \times H \times 9.8 - RSF \text{ (Newtons)} \qquad (3.3.13)$$

T_m (a) represents the maximum tension which will be present if the slack side tension is the minimum required to prevent belt slip. T_m(b) represents that which occurs if the slack side tension is the minimum required to limit inter-idler sag. An analysis described in Section 6.2 shows the frequency with which each of these criteria decide the final maximum working tension.

It is the common practice of belt manufacturers to express the maximum operating tension in tension per unit belt width and so the units are kN/m. Having found this maximum belt working tension it is necessary to choose a carcase construction that is capable of withstanding it.

Manufacturers quote rated maximum permissible operating tensions for the belt constructions that they make. An example of this is shown for the Dunlop Starflex belts in Table A5.10 of Appendix 5. A safety factor of belt tensile strength to maximum operating tension is required. This is intended to allow for any additional belt stresses that occur as discussed in Section 1.2. Commonly this safety factor, based on previous experience mainly with all cotton belts, is 10:1 but can be reduced to 9:1 for fabric belting if vulcanised splices are used for high tension belts. (Vulcanised splices form a more efficient joint than mechanical fasteners.) BTR [93] state that the safety factor must never fall below 6:1 even when allowance is made for additional stresses in the tension calculation itself. The Dunlop range of belts incorporates the tensile strength into the construction name. For example, a carcase designated 630/4 means that four plies of fabric are used to give a belt having a tensile strength of 630 kN/m. Some work has been carried out on safety factors as described in Section 2.1, but no recent studies have checked the necessity of this 10:1 value for modern belting.

A diagrammatic representation of the tension calculation technique described above is shown in Figure 3.3.1 and a worked example is given below:-

Consider a belt conveyor with the following dimensions:-

Belt width, b = 1000 mm.

Centres length, L = 65.2 m.

Conveyor lift, H = 5.6 m.

Belt speed, V = 1.2 m/s.

Peak load, T = 200 tonnes/hr.

Idler spacing, I = 1.0 m.

Drive drum is lagged with 210° angle of wrap and a gravity take-up is used.

Operating conditions are "average".

From tables (Appendix 5) the following parameters are found:-

Empty belt friction factor,	F_B	=	0.03.
Loaded belt friction factor,	F_L	=	0.03.
Length adjustment,	L_A	=	45 m.
Weight of moving parts,	P	=	60 kg/m.
Approximate belt weight,	B	=	12.8 kg/m.
Drive factor,	K	=	1.38.

Then,

Adjusted length, $L_1 = 65.2 + 45 = 110.2$ m.

Load weight, $M = \frac{T}{V} \times 0.278 = \frac{200}{1.2} \times 0.278 =$ 46.33 kg/m.

Sag factor, G, is taken for 2% sag = 6.25.

Substituting these values into equations 3.3.2, 3.3.3 and 3.3.4 gives:-

$$T_a = 0.03 \times 110.2 \times 60 \times 9.8 = 1944 \text{ N.}$$

$$T_b = 0.03 \times 110.2 \times 46.33 \times 9.8 = 1501 \text{ N.}$$

$$T_c = 46.33 \times 5.6 \times 9.8 = 2543 \text{ N.}$$

Thus $T_e = T_a + T_b + T_c = 5988$ N.

and so, from equation 3.3.7, the slack side tension, T_2, is given by:-

$$T_2 = T_e \; (K - 1) = 5988 \times 0.38 = 2275 \text{ N.}$$

The alternative maximum tensions are found from equations 3.3.11 and 3.3.12.

$$T_m \; (a) = T_e + T_2 = 5988 + 2275 = \underline{8263 \text{ N}}.$$

$$T_m \; (b) = T_e + T_o + B \times H \times 9.8 - \text{RSF.}$$

$$= 5988 + 3622 + 702 - 778 = \underline{9534 \text{ N}}.$$

Thus, in this case, it is the tension required to limit sag that decides the maximum tension that will be present in the working belt. The minimum required working tension rating of a belt construction for this application is given by:-

$$\frac{T_m}{b} = \frac{9534}{1000} \simeq 9.5 \text{ kN/m.}$$

A table of belt construction properties is then consulted to find carcases with a rated working tension of at least this value. Table A5.10 of Appendix 5, shows that all belts in the Dunlop Starflex range satisfy this requirement. This does not mean that any of the of the belt constructions could be used because one or more of the design considerations discussed below might rule out certain of them.

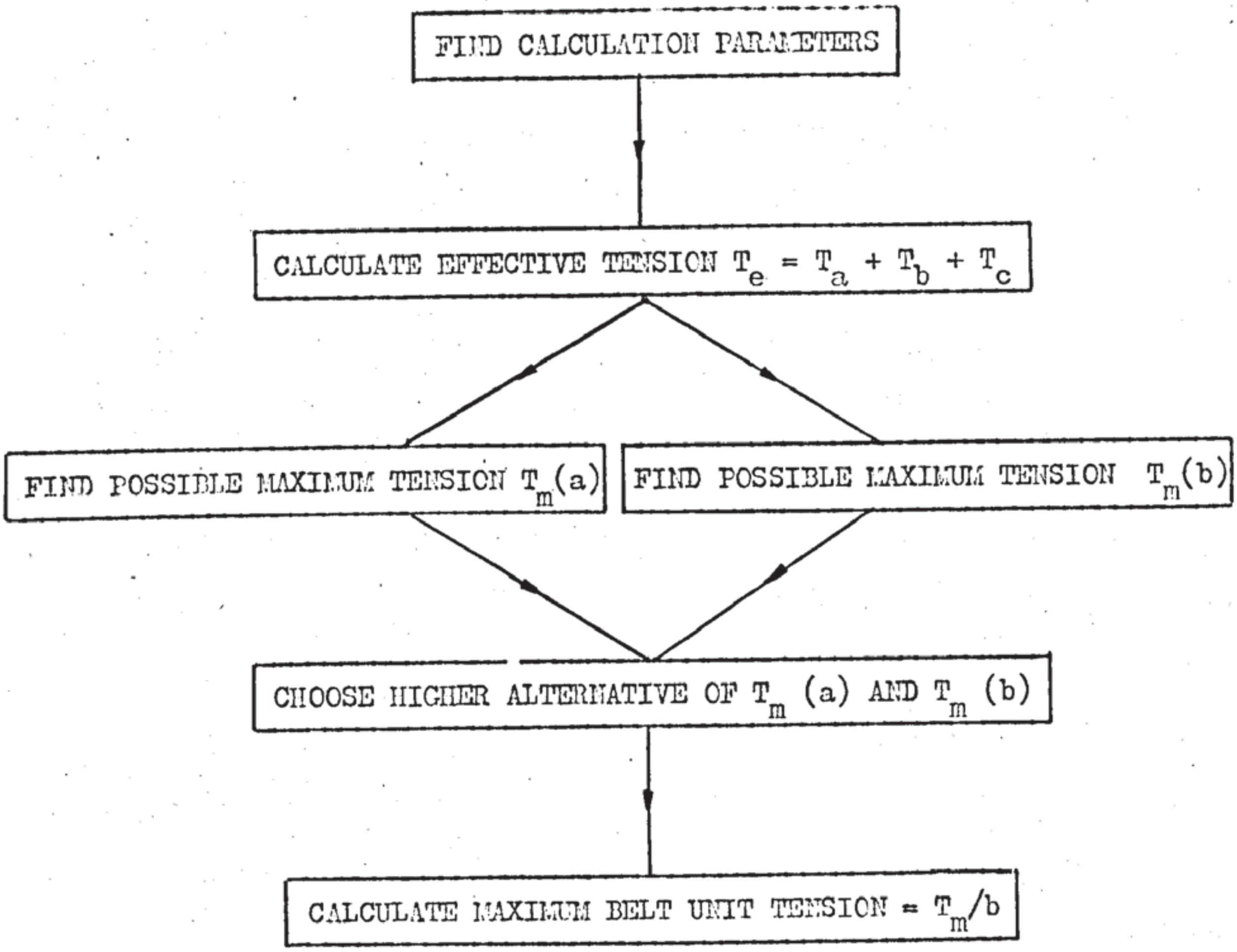


FIGURE 3.3.1

Diagrammatic representation of stages in belt working tension calculation

The significance of the various symbols is explained in the text.

The worked example given above was, in fact, for one of three installation enquiries to Dunlop Belting Division for which the customer insisted that tension calculations were also carried out using the method published by the Conveyor Equipment Manufacturers Association [31] . This method is fairly complicated because of the many different friction factors and other parameters that are used. A comparison of the results obtained by the Dunlop method and the CEMA method is presented in Table 3.3.1, and shows the fairly large discrepancies that can occur in the tension calculations using two different methods due to the fact that there is no rational basis for a common code of practice. It should be noted that differences in the figures are not consistent; in one case the CEMA method gives the higher value whereas that of Dunlop does in the other two. This suggests that Thormann [50] was correct when he concluded that the different calculation methods available each have an area of operating conditions in which they are most accurate. Unfortunately it is not possible to tell which method should be used for a particular situation because not enough tension measurements have been taken on working conveyors.

3.4 Ensuring that the selected belt has adequate transverse load support

This is the one part of the belt selection procedure in which there are major differences between the methods used by the various manufacturers. None of the selection manuals present a formula by which the load support requirement of the installation can be calculated but instead, the conditions under which the carcase constructions available in the manufacturers' standard product range will provide

Conveyor Reference	B 9	MS 1	L 5
Belt width (mm)	1000	600	1000
Centres length (m)	65.2	110.9	44.0
Conveyor lift (m)	5.6	0.65	6.2
Peak load (tonnes/hr)	200	50	600
Belt speed (m/s)	1.2	0.8	1.2
Dunlop maximum tension (N)	9534	3656	23101
CEMA maximum tension (N)	9032	7068	19462

TABLE 3.3.1

Comparison of belt maximum operating tensions calculated by two different methods the necessary property are tabulated. The methods used to do this by the major manufacturers are as follows:-

Dunlop	present a maximum permissible belt width according to carcase construction and material density (see Table A5.11 of Appendix 5).
B T R	quote a minimum number of plies required dependent on belt width, fabric type and material density.
Goodrich	quote permissible maximum material loading per unit length of conveyor according to belt width, number of plies and fabric type.
Goodyear	quote minimum number of plies required according to belt width, fabric type, idler angle, load cross-section and material density.
Fenner	who deal with solid woven* rather than plied belting make no conditions except that the idler gap* must not exceed 10 mm.
Uniroyal	specify maximum belt widths according to carcase type only.
Boston	specify permissible idler angles.

* See glossary of terms at the beginning of this thesis.

It is obvious that manufacturers consider that several belt and conveyor parameters contribute to load support, but there is no generally accepted method of quantifying the effect that each one has. It can be seen from above that Goodyear present the most comprehensive details on this aspect of belt selection. General rules that apply to the data tabulated are:-

> The required number of plies for load support increases with increasing load density, belt width and idler angle but decreases with increasing fabric weight and decreasing load cross-section. Increasing the number of plies will, of course, increase the lateral belt stiffness and all the manufacturers would appear to agree that this improves load support capability.

In Section 6.1 it will be shown that the need for meeting load support requirements decides the outcome of most belt selections and often causes gross overspecification of carcase tensile strength. An example of this can be seen from further examination of belt selection for the conveyor analysed in the section above. It was found that the maximum belt operating tension of the conveyor was in the order of 9.5 kN/m. Reference to Table A5.10 of Appendix 5 shows that any of the Dunlop Starflex belts would be capable of withstanding this tension. The belt in question was to be 1000 mm wide and carrying coke of density 0.6 tonnes/m^3 so Table A5.11 shows that the carcase type Starflex Heavy Duty 200/2 (abbreviated to 200/2HD) would not be suitable for this because of inadequate load support. Instead, a construction of the type 315/2HD must be used thus overspecifying the tensile strength of the belt and causing a large final effective safety factor in the order of 30:1.

At this point it is convenient to explain the difference between the "Starflex Heavy Duty" (HD) and the "Starflex Extra" (EX) belts. Both use the same fabrics but the "Heavy Duty" range have a larger amount of rubber between the individual plies. This additional "interply" makes the carcase stiffer than the comparable "Extra" construction and so it is claimed that improved load support is thus built into the belt [100] . It is also claimed that additional impact resistance is built into the carcase because of the cushioning effect of the extra rubber.

In view of the fact that transverse load support requirements decide the majority of belt selections and often lead to gross overspecification of other properties, the degree of importance attached to this factor in manufacturers' manuals seems inadequate. Usually only one table is used to present all details of this design criterion whereas several pages are used to explain the tension calculations. This is because, at present, load support data can only be based on the experience gained in the relatively short time since all-synthetic carcase belts came into common use, and so it can be seen that there is a need for a careful examination of this property with a view to ensuring service satisfaction with a more economical design of belt construction.

3.5 Ensuring that the selected belt has adequate troughability

This design criterion is virtually always met automatically by modern belt constructions and rarely, if ever, affects the outcome of the belt selection procedure. Nevertheless, all the major belt manufacturers still include it in their belt selection manuals. Unlike the situation with load support, there is complete agreement between manufacturers on the factors affecting troughability. In every case the permissible combinations of belt construction, belt width and idler angle are specified. The presentation methods are slightly different but nearly all manufacturers also agree on the actual numerical values given. One example of this data can be seen in Table A5.12.

3.6 Ensuring that the belt has sufficient longitudinal flexibility to pass around the end pulleys

This is another design factor that is very easy to meet with modern belt constructions and one manufacturer (Fenner) no longer includes it as part of the belt selection procedure. All the other manufacturers define this criterion in terms of minimum permissible pulley diameters for various belt constructions. In every case this data is tabulated and no manufacturer publishes a formula or empirical rule from which the quoted values are obtained; although most give footnotes saying how the minimum pulley diameters can be reduced if the belt is operating below its maximum rated tension.

There are no significant differences between minimum pulley diameter values quoted by different manufacturers for similar carcase constructions. This data for the Dunlop "Starflex" range of conveyor belts is shown in Table A5.13.

3.7 Consideration of impact resistance in belt selection procedures

Some manufacturers, for example Dunlop and BTR, claim that correct selection of belt cover quality and thickness will provide adequate resistance to any impact damage. (Cover selection is discussed in Section 3.10.) Others consider that the belt carcase also has some bearing on this parameter; those manufacturers that do this (B. F. Goodrich, Boston and Uniroyal) assume that impact damage is dependent on the height from which the load falls onto the belt and the weight of individual load lumps. Tables are presented for permissible combinations of load fall distance and lump weight for the different carcase constructions and footnotes are added showing how special impact idlers and the inclusion of a breaker* fabric can affect the quoted figures.

As the primary objective of belt covers is to protect the carcase from damage by impact and abrasion and only minor modifications can be made to the energy absorbing capacity of the carcase, it would seem logical that the belt cover alone should be considered for this design criterion. However, as manufacturers disagree about this it suggests that some valuable research work could be carried out on this subject. This is also shown by the fact that belting engineers admit to including an extra ply of fabric in belt recommendations

*See glossary of terms at the beginning of this thesis.

sometimes so that it will effectively act as a breaker (see Section 4.1).

3.8 Ensuring that belt modulus is consistent with the requirements of the conveyor structure

Most selection manuals issued by belt manufacturers to their customers do not quote much information about belt modulus. Belting is not truely elastic and so modulus, unlike the situation with metals, cannot be regarded as constant over a large range of elongations. Thus, values quoted in manuals usually represent average modulus for the range of conditions met in service. In the Dunlop [92] and BTR [93] manuals empirical formulae (which are not based on any published testwork) are presented showing how belt modulus must be compatible with any vertical curves occuring in the conveyor structure.

A concave vertical curve occurs when a conveyor changes from horizontal to incline and a convex vertical curve occurs when the change is from incline to horizontal. For concave curves, belt modulus appears in two design criteria; (a) overstress at the belt centre and (b) lack of tension at the belt edge must be avoided. These can occur because of the strain distribution that will be present in the belt as it passes through the curve and the belt centre has to travel a slightly greater distance than the belt edges. The empirical formulae quoted to ensure that these situations are avoided are:-

(a) To avoid overstress at the belt centre the minimum permissible radius of the curve is given by:-

$$R = \frac{b\ E\ \sin\beta}{9000\ (T_r - T_x)}\ (m) \qquad (3.8.1)$$

Where T_x = Belt tension at curve (kN/m).

E = Belt modulus (kN/m).

β = Troughing angle (^{o}).

T_r = Rated belt maximum tension (kN/m).

(b) To avoid lack of tension at belt edge the minimum permissible radius is given by:-

$$R = \frac{b\ E\ \sin\beta}{4500\ (T_x - 4.4)}\ (m) \qquad (3.8.2)$$

It should be noted that another condition, not dependent on belt modulus, applies for concave curves; to prevent the belt lifting off the idlers the minimum permissible radius is given by:-

$$R = \frac{T_x\ b}{9.8\ (B + M)}\ (m) \qquad (3.8.3)$$

Equations 3.8.1, 3.8.2 and 3.8.3 are used to find three possible values of the minimum radius and the smallest of these is, of course, the deciding criterion.

For convex curves the situation is reversed and the situations that must be avoided are (a) overstress of the belt edges and (b) lack of tension at the belt centre. The minimum permissible radii to prevent these occurring are, respectively:-

$$R = \frac{b\ E\ \mathrm{Sin}\ \beta}{4500\ (T_r - T_x)}\ (m) \qquad (3.8.4)$$

$$R = \frac{b\ E\ \mathrm{Sin}\ \beta}{9000\ (T_x - 4.4)}\ (m) \qquad (3.8.5)$$

As belt modulus is a property that depends very much on the fabric used for the carcase, it is very difficult to vary it for a given belt construction and so the procedure to find the minimum permissible radius of a vertical curve perhaps can be regarded as a design consideration for the conveyor rather than for the belt. It is therefore very important that the conveyor designer makes any vertical curve in an installation compatible with the properties of modern belting.

Another parameter that can be regarded as a conveyor structure one is transition distance: this is the distance between the tail pulley of the conveyor and the first set of troughing idlers. With deep trough conveying it is important to ensure that this distance is great enough to prevent overstress at the belt edges or lack of tension at the edges of the belt have to travel slightly further than the centre. Several manufacturers present minimum permissible transition distances. An example is given in Table A5.14 of Appendix 5. The figures quoted are found from formulae derived from purely geometrical considerations [93].

Vertical curves and transition distances are two factors that must be considered for any conveyor design and they should be checked when any belt construction is recommended. However, if they are found to be unsuitable, little can be achieved by changing to a different belt construction and so, in such cases, it is usually necessary to alter the conveyor structure.

3.9 Ensuring belt permanent elongation characteristics are compatible with the conveyor structure

When the selected belt has been in operation for a short time some permanent stretch will have occured in it and it is important that there be a take-up mechanism which has enough travel to accommodate this stretch. With one exception (Fenner) all the major manufacturers specify minimum amounts of take-up travel. These are usually quoted in terms of percentage of conveyor centres length and typical values are in the range of 2 - 4%. Each manufacturer requests greater take-up travel if a vulcanised splice is to be used on a long conveyor ra ther than mechanical fasteners. This is because a vulcanised splice is relatively expensive and a safety margin is allowed to make absolutely sure that after stretch has occurred a piece of belting will not have to be removed and a new splice completed. For extremely long conveyors where it is impractical for take-up travel to be even only 2% of centres length it is common practice initially to fit mechanical fasteners and once the belt has stretched a length is removed and a permanent vulcanised splice is made.

It should be noted that in addition to permanent belt stretch,an elongation dependent on the belt modulus will take place every time the conveyor starts operation because of the higher tensions present. This is almost entirely recoverable when the applied stress is removed but a gravity take-up must have enough travel to allow for this effect.

It can be seen that, like vertical curves and transition distances, this aspect of the belt selection procedure should really be regarded as a conveyor design factor and very little can be done by changing belt construction if the conveyor designer has not made adequate provision for take-up movement.

3.10 Selection of carcase covers

All belt manufacturers agree that the service factors affecting the choice of cover compound and thickness are load abrasiveness, load lump size and load weight. Every manufacturer notes that special cover compounds must be used for carrying hot or oily materials. The various compounds available and the conditions for which they are most suitable have been discussed by Buettner [37] and a summary of his work has been given in Table 2.1.1.

A typical example of how cover thickness and quality information is presented in belt selection manuals is given in Table A5.16 of Appendix 5. No significant differences occur between the sets of data given by the various manufacturers and, in fact, in most cases they are identical. The British Standard for conveyor belting [26] specifies tensile strength and elongation characteristics for two grades of cover compound; M24 for sharp and abrasive loads and N17 for less abrasive material and minimum thickness values are specified (see Table 3.10.1). It is interesting to note that many manufacturers make no mention of the British Standard compounds and those that do make only passing reference.

Service	Minimum Grade	Minimum Cover Thickness	
		Carrying Side	Pulley Side
Sharp and abrasive materials causing severe wear to conveyor, such as metallic ores (run of mine), limestone (as quarried), granite (as quarried), quartz, blast furnace clinker and slag, crushed metallic ores, sandstone (as quarried), stone chippings, slate coke (Cold) broken glass and gravel.	M24	3.2	1.6
Moderately abrasive materials such as rubble, sand (sharp), superphosphate (lump and powder), bone, coal (surface), ashes, unslaked lime, cement (run of oven).	N17	2.4	0.8
Slightly abrasive materials, non abrasive and dry materials such as soda, earth, sand (smooth), cement (ground), clay, slaked lime, charcoal, grain, vegetables, fruit, flour, dry powder (inert), wood chips, pulp (dry).	N17	1.2	0.8

TABLE 3.10.1

British Standard minimum grade and thickness

of rubber belt covers

The selection of correct grade and thickness of cover represents the final section of the belt selection procedure and once the amount of cover is decided the total weight of the selected belt can be found. Most of the manufacturers present tables showing the exact weight of the chosen carcase and the additional weight per millimetre of cover. When the weight of the belt is known the tension calculations described in Section 3.3 should be repeated using this value in place of the approximate belt weight used previously. This is to ensure that the calculated maximum operating tension is not altered so much that it exceeds the rated tension of the selected carcase. This rarely occurs and is really only a possibility if the operating tension found originally is very near the carcase rated tension and the selected covers are exceptionally thick.

A diagrammatic summary of the Dunlop Starflex belt selection procedure excluding those features that can be regarded as conveyor structure design factors is given in Figure 3.10.1.

Several manufacturers provide calculation sheets to help make the belt selection procedure easier. These are used to collect together the essential design information and carry out the tension calculation by filling in details as required. An example, from Dunlop, is shown in Figure 3.10.2, and this particular calculation sheet shows the different tension components that must be considered when finding the maximum operating tension for various conveyor configurations.

Most of these calculation sheets are very simple to complete and, on them, as with the selection procedure itself, the tension calculation

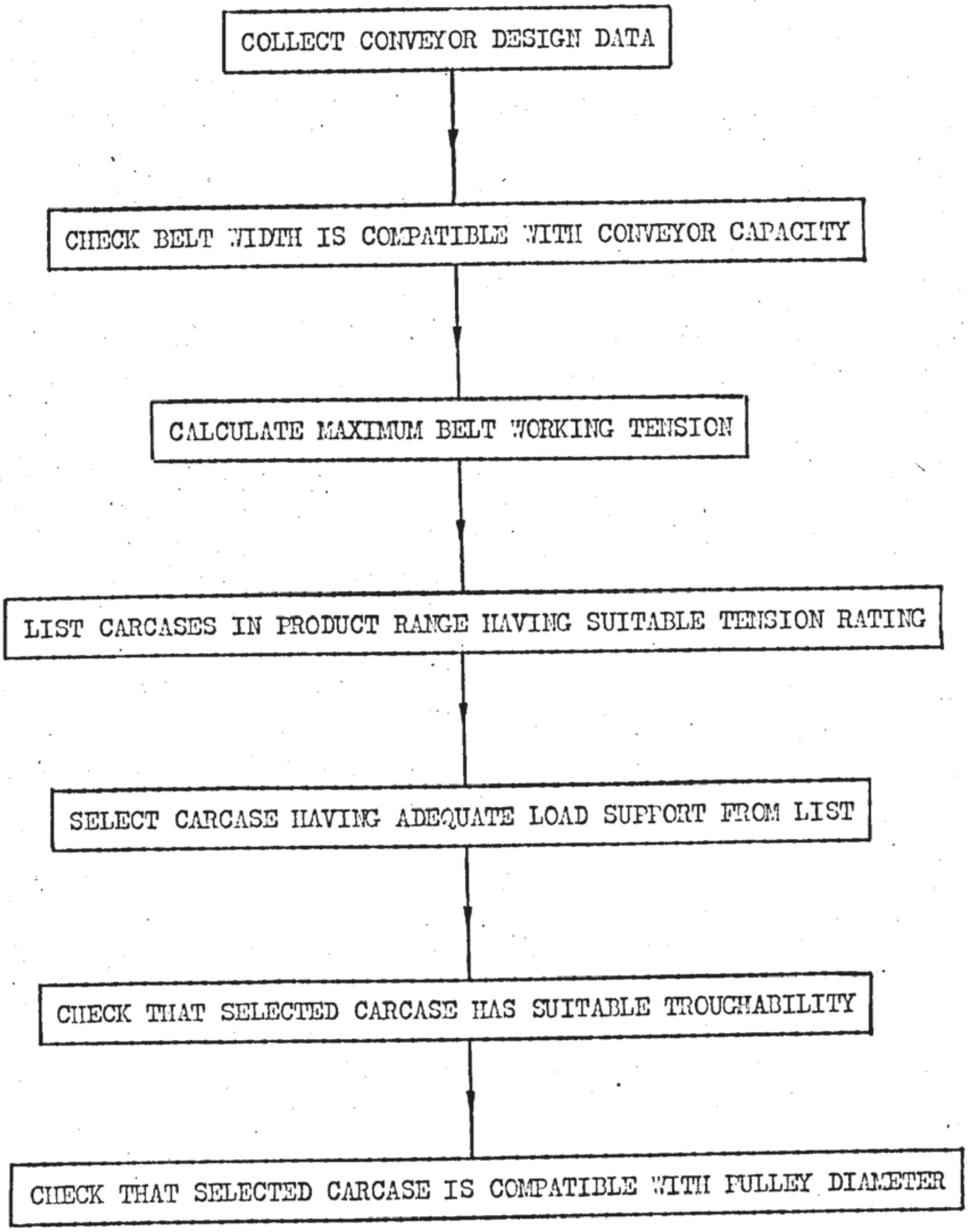


FIGURE 3.10.1

Summary of Dunlop belt selection procedure

ENQUIRY .... CUSTOMER ....

CONVEYOR .... REF ....

TYPE ....

b .... WIDTH (mm) | P .... WT OF MOVING PARTS (kg/m)

L .... CONVEYOR CENTRES (m) | LA .... LENGTH ADJUSTMENT (m)

H .... VERTICAL LIFT OR DROP (m) | L1 .... ADJUSTED LENGTH (L & LA) (m)

θ .... ANGLE OF INCLINE (°) | H .... FROM ANGLE = L x SIN θ

V .... BELT SPEED (m/s) | SIN θ =

T .... PEAK LOAD (tonnes/hr) | F_B .... FRICTION FACTOR EMPTY BELT

B .... ASSUMED BELT WEIGHT (kg/m) | F_L .... FRICTION FACTOR LOAD ON LEVEL

.... DRIVE/TAKE UP | M .... WT OF LOAD (kg/m)

K .... DRIVE FACTOR | $M = \frac{0.278 \times T}{V}$

I .... IDLER SPACING (m) | G .... SAG FACTOR = 6.25 FOR 2% SAG

MATERIAL, SIZE .... DENSITY............

TENSION COMPONENTS

(1) EMPTY BELT, FE x L1 x P x 9.8	=	x	x	x 9.8	=	...............N
(2) LOAD ON LEVEL, FL x L1 x M x 9.8	=	x	x	x 9.8	=	...............N
(3) INCLINED LOAD, H x M x 9.8	=	x	x	x 9.8	=	...............N
(4) INCLINED BELT, H x B x 9.8	=	x	x	x 9.8	=	...............N
(5) RETURN SIDE FRICTION, FE x L x P/2 x 9.8	=	x	x	x 9.8	=	...............N
(6) SAG CONTROL TENSION, TO, G x (B + M) x I x 9.8	=	x	x	x 9.8	=	...............N
(7) EFFECTIVE TENSION, T_e = (1) + (2) $\pm$ (3)	=	x	x	x 9.8	=	...............N
(8) SLACK TENSION, T2 = T_e (K − 1)	=	x	x	x 9.8	=	...............N

MAXIMUM TENSION (T_m) ACCORDING TO FOLLOWING CONDITIONS

CONVEYOR	DRIVE POSITION	PREFERENCE	NEED FOR RESTRAINT	MAXIMUM TENSION (TAKE HIGHER ALTERNATIVE)
Level or elevating	Head	1	–	1st T_e + T2 2nd T_e + BM − RSF + TO
Level or elevating	Tail	2	–	1st T_e + T2 + BM − RSF 2nd T_e + T2
Lowering	Tail	1	Yes (after starting)	1st T_e + T2 2nd T_e + BM + RSF + TO
Lowering	Head	2	Yes "	T_e + T2 + BM + RSF
Lowering	Head	1	No	1st T_e + T2 (Calculate loaded and empty) 2nd T2 + BM + RSF
Lowering	Tail	2	No	1st T_e + T2 (Calculate loaded and empty) 2nd BM + RSF + TO

T_m = ... = ... N

T_m/mm WIDTH, TM/b ________________ = ...N/mm,kN/m

KILOWATTS = $\frac{T_e \times V}{1000}$ = ________ x ________ = ...

OR T_e = $\frac{\text{KILOWATTS} \times 1000}{V}$ = ________ x ________ = ... N

FIGURE 3.10.2

Calculation sheet for Dunlop belt selection procedure is given most importance. Most manufacturers make a note that having carried out the tension calculation, consideration of other factors such as load support must be made.

An additional benefit that could be gained from the calculation sheets is that of data collection. If the manufacturer was to complete a full sheet for every belt enquiry and include details of the final construction selection and the reasons for recommending it, a file of information on the belting market would be built-up gradually. In addition, faults in the belt selection procedure might be shown up by subsequent study of belt performance. The Dunlop practice of using calculation sheets is discussed in Section 4.1.

3.11 Conclusions from study of belt selection procedures

The discussion above has shown that work on several parameters involved in belt selection procedures would be useful. These include edge margins, maximum lump sizes, co-efficients of friction and safety factors, but, in practice, they represent fairly minor problems compared to the major disadvantage of modern belt selection practice. This is that the formulae and data used have not been revised to keep pace with developments in conveyor structures and carcase constructions. If the selection manuals used at present are compared with those of twenty years ago there is virtually no difference between them. No manufacturer has yet been able to develop a new technique of selection designed especially for new belt materials. The only significant difference that exists between the different belt selection methods are the techniques for consideration of impact resistance and load support.

It is now time that the conveyor belt industry revised their manuals so that design parameters not critical to present technology are no longer included and those that have become more important are given due consideration. This would mean that ensuring the belt has adequate troughability might be relegated to a footnote explaining that it must be considered under certain extreme conditions - for example, narrow belts, high idler angle and thick belt covers. (The fact that troughability is not critical for modern belting is shown by the manufacturers taking no account of cover thickness in the data presented even though an investigation mentioned in Section 2.2 has shown that cover thickness can affect troughability considerably.) The tension calculation should also be given less importance and other factors such as load support and impact resistance should be featured more strongly. This is because often a carcase strength much greater than that required by tension considerations is needed to provide these two quantities. B. F. Goodrich Limited [98] suggest a quick method of belt selection because of this:- if the load support or impact resistance requirements demand a belt having a greater rated tension than can possibly be transmitted by the conveyor motor, then it is possible to ignore the tension calculation and select a belt by immediate reference to impact resistance or load support data. This might rarely happen in practice because of the large power requirement safety factors that conveyor designers use for motors. A new rational approach of demonstrating when the tension calculation can be ignored is described in Section 5.2.

It has already been mentioned that some manufacturers, such as Dunlop, do not consider impact resistance as part of the carcase selection procedure. This is not necessarily a valid assumption, but there has, at least, been a great deal of work on which to form an opinion carried out on this quantity as described in Chapter 2. There has, however, been no published work whatsoever on the mechanism of load support and it is really only with this parameter that manufacturers show any disagreement. Detailed studies of this parameter have therefore been carried out as part of the present research work and the methods and results of these are discussed fully in Chapters 7 and 8.

A full investigation of how a belt performs could possibly result in a radical change in belt selection. If it were possible to categorise conveyor installations according to various factors such as load characteristics and structure dimensions it might then become feasible to select the correct belt by a simple "end-use" method. For example, it might be possible to say that for a conveyor of certain dimensions carrying gravel of 1.0 tonnes/m^3 density a carcase of the Dunlop type 630/4HD must be used and top covers of 5 mm are required. An attempt to introduce such a system for very light duty belts has already been made by Stephens Limited [101]. This firm supplies conveyor belts for in-plant materials handling, typical applications of which are food processing, paper processing and baggage handling. Tensile strength is not a problem for these because, using synthetic material, only two or three plies of light fabric is more than adequate for any of the applications. It is therefore possible to select a belt without carrying out a tension calculation; instead immediate reference is made to an applications chart. For example, in airport terminals a conveyor carrying luggage and light goods on a slider bed could be fitted with a belt designated E22/2 UO V10. This coding has the following significance:-

E - 100% polyester fabric is used.

22/2 - Two plies of fabric form the carcase which requires a force of 22 kgf/cm to elongate it to 1%.

UO - Bottom cover is impregnated with polyurethane to slide easily.

V5 - Top cover is of PVC 0.5 mm thick.

The designation of belt type E18/3 V5 V10 in the sugar industry for a belt running on standard idlers is another example. In this case the belt would comprise three plies of polyester fabric and would need a force of 18 kgf to elongate it to 1%. The covers would be of PVC, the bottom being 5 mm thick and the top being 10 mm thick.

In the Stephens catalogue a whole list of possible applications is given and suitable belt types are coded. If this could be done for bulk conveyors it would significantly simplify the selection procedure. A great deal of information from service would be required to do this and properly completed belt selection forms would provide a useful basis for recording data. A suitable form would request the following information:-

Relevant conveyor structure details.
Load details.
Reason for failure of previous belt.
Results of belt selection procedure.
Belt type installed (with reasons).

If details of subsequent belt performance and failure were collected, the suitability of the various belt types for certain applications would be assessed and thus lead towards the introduction of end-use selection.

This should be the long term aim of the belting industry, but in the meantime it is important to revise existing procedures in order to make them as efficient as possible. This, in turn, means that the property of transverse load support should be investigated thoroughly but does not mean that revision of all other design factors should not be forgotten. Future developments in belt materials might make these or completely different parameters the most important to consider.

As the whole process of belt selection is too complex for a completely theoretical approach to be made, a mixture of empirical data and ad hoc theory is required if a practicable situation is to be obtained. Therefore, as much information as possible should be collected so every design property having some bearing on belt performance can be considered in belt selection.

CHAPTER 4

EXAMINATION OF THE CURRENT DUNLOP BELT DESIGN AND SELECTION PRACTICE

In Chapter 3 the current published Dunlop belt selection procedure has been reviewed and compared with those of other belt manufacturers and users. An examination is made in this chapter of the Dunlop practice of actually applying the published process. In addition, a study is made of the methods used to assess belt construction properties which influence selection and the efforts made to collect service information to ensure the relevance of the selection system. A comparison with the requirements of an "ideal system" is made and, although this chapter relates specifically to Dunlop practice, it is felt that a similar situation exists with all the major belt manufacturers.

The attributes required of an ideal belt design and selection procedure have been discussed in Chapter 1; for convenience, they are summarised here as:-

(i) Each design criterion should be assessed in relation to service information and field trials to ensure that the correct belt and conveyor properties affect the outcome of each stage of the procedure.

(ii) The belt construction data relevant to each design criterion must be obtained from testing procedures which correlate with service conditions.

(iii) The procedure must be rigorously applied in all cases and final belt construction selections should be filed along with the original conveyor specification data. The belt manufacturing details and any service information relating to its field performance must also be recorded.

(iv) The procedure and the data used in it must be under constant revision to allow for information from belt field performance and any developments in belt technology or conveyor practice.

The following sections describe situations where Dunlop practice does not yet meet these requirements fully and the reasons why this is so are discussed.

4.1 Belt Selection Practice

The belt selections in the Dunlop organisation are carried out by two departments: one deals with replacement belting for working conveyors, the other handles business with conveyor original equipment manufacturers which tends to be for proposed large materials-handling schemes.

When a customer makes an enquiry regarding the correct belt to use for an existing conveyor, a belt design engineer should complete one of the belt calculation sheets (see Chapter 3) with details of the installation and belt selection should then be carried out following exactly the procedure described in the handbook "The Starflex Conveyor".

In practice, what usually happens is that the conveyor details are not transferred to one of the calculation sheets and, in many cases, the published procedure is not fully carried out because the belt design engineer selects the belt construction, by experience,for "load support reasons".

It will be seen from Section 6.1 that this process is, basically, rational as, in most cases, the final belt construction is not affected by tensile strength considerations, but is determined by transverse load support requirements. However, as the conveyor conditions where the tension calculation is not necessary have not been quantified, it is possible that the use of experience based intuitive selection could lead to cases where tension is significant being missed. If the conditions where tension is not significant could be quantified it would be possible to select the correct belt construction by immediate use of "maximum belt width data" which, in effect forms the transverse load support data. (An attempt to introduce such a system is described in Section 5.2.) However, in practice, personalised end-use selection rules for load support such as "Belts with less than three plies cannot be used in quarries" are introduced. Thus, having dispensed with the tension calculation, the rest of the published procedure is also often not used. There is an extremely good case for the introduction of "end-use" rules into the belt selection procedure because they are very quick and simple to apply, but until a vast amount of service data regarding belt performance is available, it is impossible to establish and validate such rules. Other reasons, as well as load support, are given for the end-use rule example given. One is that a belt with only two or three plies is very thin and, to a quarry manager, looks weak and inferior. This shows the need for a customer education service to demonstrate the physical capabilities of modern belt reinforcement materials to the user. Another, more technical, reason for the rule is that belts in quarries need high impact resistance, and so one or two extra plies are included, not to act as load bearing members, but intended to form "breakers".

The fact that belting design engineers often ignore published design data is, perhaps, justified - particularly for the transverse load support data - because certain sections of it have no proven relevance to service requirements. The result of this practice is that usually the final belt selection bears no relation to that which would result from following, step by step, through the published procedure. Usually over-specification of belt properties is caused as end-use rules tend to err on the side of safety,but,in some cases, under-specification can occur. (It should be noted that the terms under-and over-specification are, in this case, relative to the belt requirements shown by the manual. As the data in the manual may not be totally accurate, the terms are not necessarily relative to the actual needs of the installation.)

Over-specification means increased cost to the customer and hence lower competitiveness for the belt manufacturer compared with other companies. As the Dunlop belt selection manual is available to customers and the procedures described in it are relatively simple to follow, it is possible that the belt user will check the construction recommendation. Customer relations will not be improved if it is found that the quoted Dunlop belt selection is for a much more expensive construction than that from the manual. There are, of course, sometimes valid economic reasons for not following the manual exactly. The most common are stock availability and standardisation of belt types on a site. A proviso to cover these eventualities is included in the "Starflex Conveyor":-

> "The following notes on belt selection will help the potential user to select the optimum belt for a given application. If the quantity of belting involved is economical this construction can be supplied. For shorter lengths, however, the user will obtain a more economical construction and better delivery by utilising the ex-stock minimum cost construction available which meets the requirements of the application, even though this may result in a belt of higher strength or heavier gauge covers etc.,"

Under-specification from the requirements in the manual can cause problems if the belt does not perform satisfactorily - even if it is not the belt itself that is at fault. An example of this occurred when a complaint was made about a belt supplied for a conveyor at a steelworks. The complaint concerned "generally insufficient load support". The conveyor in question did not operate at full capacity and so published transverse load support data was considered to be inapplicable by design engineers. (Some manufacturers, as shown in Chapter 3, reduce their transverse load support requirements if the conveyor is not operating at full capacity.) A lighter belt than that which would have been selected from the manual was therefore installed and when the conveyor's performance gave cause for concern the customer had reasonable grounds for complaint, even though it was almost certainly the conveyor structure that was causing the problem. A large and expensive test had to be carried out to demonstrate that the belt was not at fault. The fact that the belt construction was, in practice, suitable for the alleged application demonstrates that its published transverse load support capability is probably under-estimated. (The test work concerned with this complaint is described in Section 7.6.)

The practice of not following the published selection procedure can lead to inconsistency in the recommendations for the belt constructions to be used on similar installations. It also makes difficult the study of performance of belts under the field conditions for which they were intended. This problem is further exacerbated by the fact that tension calculation sheets (which detail original equipment specifications) are not always completed. If calculation sheets were completed and filed, they would, as mentioned previously, form a useful data base for studies concerning the make-up of the conveyor belt market. Such information would help to allocate research resources to the areas of greatest need and marketing resources to those of greatest possible returns. It would take a considerable amount of time to build up a complete picture of the market, but the need for such information can clearly be seen from the analyses to be described in Chapter 6, which, although using only a relatively small sample of installations, demonstrated some interesting points and gave extremely useful results.

The situation when considering proposed large materials-handling schemes is slightly different from the practice of dealing with enquiries involving only one or two belts, because the complete belt selection procedure cannot be carried out for every conveyor due to lack of available time, so instead, the original specification data is studied and the conveyors which would appear to be the "worst cases" (longest length, greatest load weight etc.,) are found. The tension calculations are then carried out for these and the correct belt types to be used are decided. The remainder of the belt selections then tend to be centred around these "worst cases". Standardisation of belt types on site seems to be the main criterion considered, and a final scheme is proposed with only two or three belt construction types being used. No studies are made of the effects on the price to the customer of increasing the number of belt types offered. Such an exercise should be carried out if the recommended product mix is to be the optimum for the customer.

This practice with large schemes could possibly lead to mistakes as the "worst cases" selected by scanning the specification data might not, in practice, prove to be the situations requiring the greatest belt tension rating or transverse load support characteristics. The system also means that, for the majority of conveyors, the final belt selection tends to have over-specified design properties and does not necessarily represent the construction that would be chosen from the manual.

Thus, for both individual conveyors and large schemes the published belt selection procedure is not always followed exactly. This, perhaps, shows that design engineers using the system prefer to use their experience rather than the data in the manual. This, of course, might be justified, but if it is, then the published procedure should be changed. Revisions to the system are very difficult to make because of the lack of systematically collected and filed specification data and field performance details of belts, particularly for the relatively modern all synthetic belts. The work required to overcome this situation is discussed below in Section 4.4.

4.2 Belt Design-Property Assessment Practice

This section describes the current practice of assessing the design information that is published for the manufactured range of belt constructions. The parameters involved have been discussed in Chapter 3 and actual values of them for the Dunlop "Starflex" range of conveyor belts are given in Appendix 5. Each parameter is dealt with below in the basic order of consideration during the selection procedure.

Recommended maximum belt working tension is easily found from the actual tensile strength of the whole belt divided by a safety factor of ten, although a slightly lower safety factor is used for belt constructions at the stronger end of the range if a vulcanised splice is to be used. This is done because mechanical fasteners tend to give a less efficient joint than a vulcanised splice. The relevance of the 10:1 safety factor has already been questioned above.

The tests used to find the whole belt tensile strength and the efficiency of joints are carried out under static and dynamic conditions as described in British Standard 4890 [79]. Dynamic tests represent service conditions more closely than static tests, and an example of an applicable technique which is more versatile than that of BS.4890 has been described by Watts [64]. A dynamic machine is available in Dunlop Angus Belting Group which can be used for the testing of splice efficiency, but this tends actually to be used for development work with new belt constructions and jointing techniques. A test series purely to establish the maximum permissible working tensions of the common "Starflex" range of belts with various methods of joining has not been considered necessary because, with present-day belting constructions, the tensile strength is not always the most important property to consider.

The transverse load support capability of the Dunlop belt constructions is expressed in terms of a maximum permissible belt width under given load conditions and Figures 4.2.1 and 4.2.2 show this information for some of the belt constructions in the Starflex range. There is no test, service-related or otherwise, that is used to assess this belt design property, and there is no theoretical method to establish the data published. Thus, experience with belting is the only basis used to assess transverse load support capability. This experience has been gained mainly with older types of conveyor belt construction, and so the data for the newer all-synthetic belt types has tended to be extrapolated with the assumption that increasing the bulk (and hence the stiffness) of the belt will improve the transverse load support capability. The differences in the load support curves for the various belt types shown in Figures 4.2.1 and 4.2.2 demonstrate the inconsistencies that can occur in the published information due to this assessment method. For example, there is no set increment of maximum permissible width for a change from "Extra Duty" to "Heavy Duty" belts or for a particular increase in number of plies. (Other parameters, such as fabric type, remaining the same.) Such discrepancies appear in the manuals of all manufacturers publishing load support information. As only experience is relied upon to assess

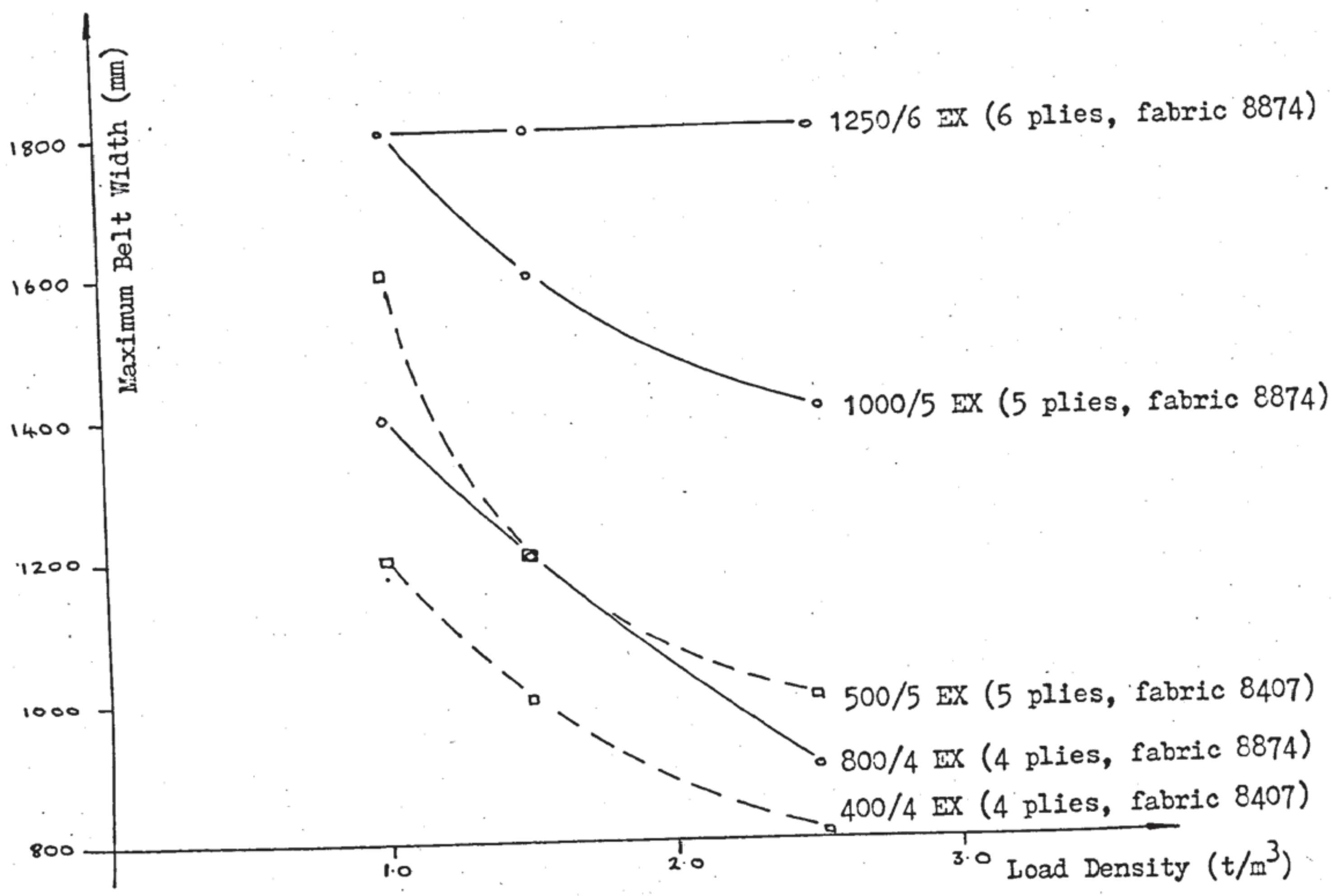


FIGURE 4.2.1

Load support curves for some belt constructions of the type "Starflex Extra"

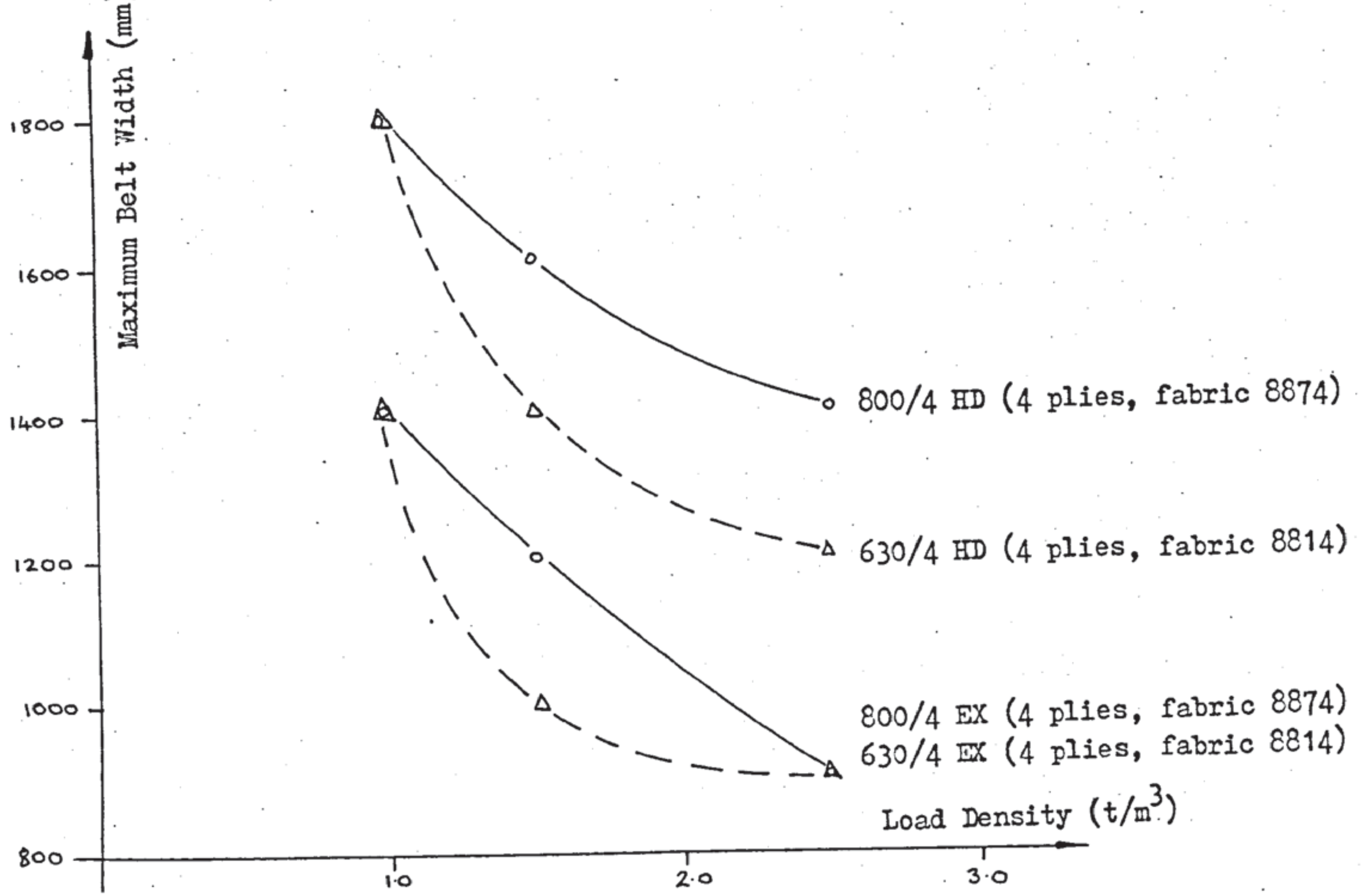


FIGURE 4.2.2

Load support curves for some belt constructions of the types "Starflex Extra" and "Starflex Heavy Duty this design property, it is probable that it is being wrongly specified, and as few problems occur with it nowadays, it is most likely that over-specification occurs. As mentioned above in Section 4.1, some belt design engineers, from personal experience, do not use the published data which, as stated here, is itself based on experience. (It should be noted that as well as there being no rational method for assessing the transverse load support capability of a given belt construction, there is no method of establishing the transverse load support requirement of a given load/conveyor combination.)

The troughability of all-synthetic belts rarely, if ever, poses problems and it is questionable whether its consideration need still be included in any selection procedure. Usually the belts are so flexible transversely that, when running empty, they touch the centre roller of the three-roller idler sets that are in common use: this condition is required to ensure ease of belt tracking. The amount of contact area actually required is not defined, but if a belt construction passes the Troughability Test of British Standard 490, it will perform satisfactorily in service for this aspect of the design procedure. The test is very simple (see Figure 2.2.6) and is service-related to the extent that the degree of deformation which occurs in service is reproduced. The test, however, is not carried out with the sample under the longitudinal tension that will occur when the belt is working, and there is no published practical work to establish a correlation between the test and the behaviour in service. It is therefore possible that the test proves to be satisfactory only because modern belting easily meets service troughability requirements and an extremely sensitive test method is not required. This does not, of course, mean that troughability will not be critical for any future belt developments and so any work to correlate a troughability test with service would be useful.

Impact and abrasion resistance do not appear, as such, in the Dunlop belt selection procedure (see Chapter 3) because it is considered that correct choice of belt covers and carcase construction will ensure that these criteria are met. (As demonstrated in Section 4.1, some belt design engineers do not think that this is the case.) There is no standard test for either of these properties, although attempts have been made to include an abrasion test in International Standards but poor reproducibility of results with the proposed test methods has, up to now, prohibited their inclusion. Test machines are available in Dunlop Angus Belting Group for these two properties. The impact testing machine is that described in Section 2.2 whilst the abrasion test is not a small scale one along the lines of those considered for inclusion in belting standards, but consists of a whole-belt sample running under a constrained load of rocks which abrade the surface of the belt. These tests would both appear to make a reasonable attempt to accelerate the effects of service conditions, but due to pressure of development work, neither has yet been used to study the resistance of the complete Starflex range of belts to these forms of damage under various conditions.

The final aspect of belt design to be considered in the Dunlop belt selection procedure is that of checking that the belt construction has sufficient longitudinal flexibility to conform to the pulley diameters on the conveyor. Again, with modern belt constructions, this is no longer a problem and, as with troughability, it is questionable whether it need be considered as part of the procedure for the normal range of belting applications. The recommended minimum pulley diameters are not found from any standard test procedures, but, instead, are based loosely on the rule-of-thumb that they should be at least one hundred times the thickness of the belt. (This should ensure that very little additional strain occurs in the outer plies of the belt due to bending. It is difficult to accurately predict the extra strain theoretically because of the very different behaviour of belting in extension and compression and the position of the neutral axis is not known.) A more accurate formula has been derived empirically from a series of laboratory tests by Spain [23] to relate the belt flex life to pulley diameter and other conveyor parameters. The formula which is given in equation 2.6.1, is not yet in general use within the belting industry.

The design property assessment discussed above has only included the criteria involved in the published Dunlop belt selection procedure. Other design properties such as cover adhesion and ply-to-ply adhesion which are demanded by British Standard 490 have not been dealt with. The small scale tests for these properties are carried out regularly, but few have any proven correlation with service requirements (see Section 2.6) and so can therefore only really be regarded as quality control checks.

The discussion above has shown that accepted service related tests for some of the more important parameters (that is, those determining belt selections) are not available. Therefore, if the published design data for a standard product range is to have a more substantial foundation, a programme should be carried out to:-

(a) Establish standard values for the design parameters of the whole manufactured construction range on test equipment available. (That is, in particular, impact and abrasion resistance and minimum pulley diameters should be assessed.) Such standard values are required, anyway, as a basis for any new belt construction development.

(b) Develop new tests for the properties which, at present, cannot be assessed by logical means. The most important to consider is that of transverse load support, as no test whatsoever is available and this parameter decides the majority of belt selections - (see Section 6.1).

It is recognised that no test can be exactly correlated with service unless a great-deal of field data is at hand. As this data is not yet available, any laboratory test can really only be comparative. It is therefore important to collect all results from all the tests in order to facilitate the required correlation in the future, and the information needed is discussed in Section 4.4

4.3 Service Data Collection Practice

The only service data collected by Dunlop Belting Division as a matter of routine is the original conveyor specification data recorded when a site survey is carried out. A survey results in a list of recommendations for Dunlop Belt constructions to be used on each conveyor and so the information collected is therefore that which is used in the belt selection procedure. One of the forms used to collect it is shown in Figure 4.3.1.

Once the construction selections have been made, a list of all the conveyors on site is made with their belt length, width and construction recommendation. One copy of the survey results is sent to the customer and another is filed for future reference - usually with the original conveyor specification data, although this is sometimes destroyed. This data could be analysed to help give a better idea of the major design requirements of the belting market, although it should be remembered that these surveys represent a fairly specialised section of the total market because they are usually carried out for larger sites such as steel plants and roadstone quarries. The comments made in Section 4.1 concerning belt selection practice are also applicable to the construction recommendations resulting from the surveys, and so, again, constructions that would be achieved from following exactly the published procedure are not always selected. The reasons for the final construction recommendations are not recorded, thus making it very difficult to assess the relevance of the various personalised selection rules from by a study of belt service performance if the recommendations are taken up.

Few observations concerning the performance characteristics of belts are fed back to the development department from routine site visits even though it would be a very easy matter to make simple observations such as the presence of spillage, cover tears, cover cracking and damage to the belt by catching in the idler gap under various conditions of use. This information would eventually build up a useful picture of the capabilities of the different belt construction types under various conditions. At present, the only circumstances which tend to be

CONVEYOR

1. Belt Ref	
2. Width	
3. Distance between centres	Belt Length
4. Speed	
5. Troughed or Flat	
6. Weight carried	
7. Peak Load	
8. Type of material	
9. Wt. t/m^3	
10. Size of material	
11. State of material	
12. Fumes or moisture	
13. Idlers	
14. Level or inclined	
15. Trippers	
16. H.P. of motor	

17. Drive	Type	Wrap	Lagging	Position

18. Tensioning	
19. Joining	

20/21.	Pulleys	Head	Snub	T. D.	Tail
	Dia.				
	Width				

22. Idler Spacing	
23. Delivery to belt	
24. Idlers at loading point	
25. Discharge from belt	
26. Use of scrapers	

FIGURE 4.3.1

Example of data collection sheet used by

Dunlop Belting Division for site surveys reported are those that have led to complaints, and although these can provide useful information, they cannot give a complete picture of service performance.

An ideal situation would be to make actual quantitative assessments of such items as inter-idler sag, power requirements and belt wear. It would therefore be useful for the development department of Dunlop Angus Belting Group to have a team of engineers to do this work with the co-operation of selected customers. This work would be of great value in the general development of the standard product range and service-related testing techniques.

Controlled field trials should represent the most useful method of obtaining full performance data. However, because of the expense of setting up such an operation, they are rarely used except for observations on completely new developments rather than on constructions in the standard product range. When field trials are carried out it is important to make as many detailed measurements as possible and to keep full and accurate records of the belt's running conditions. A field trial carried out by Dunlop and finishing in November, 1975 demonstrated that such practice has not yet been developed fully. A belt had been prepared with six different cover compounds to evaluate and compare the heat resistant qualities of each. Similar tests had been carried out by the Malaysian Rubber Producers' Research Association [77] and Hannoyer of Kleber-Colombes [75]. These researchers had experienced difficulty with "before and after use" measurements of various parameters such as cover thickness but the problems had been overcome by the development of special testing techniques. However, this literature was not reviewed prior to carrying out the field trial mentioned above and some of the problems met by the previous researchers were encountered. For example, a special template as used by Hannoyer would have been useful to ensure that measurements of belt thickness before and after use were taken at exactly the same positions. (It has already been mentioned that National Coal Board work has established a rational field trial practice which should be used as the basis for any future work.) It was argued that the purpose of the test was only to consider the heat resisting properties of the different cover compounds, and other parameters were of no concern. However, it was unfortunate that, having gone to the trouble and expense of setting up the trial, full advantage of the situation was not taken to study the changes occurring in "whole-belt" properties such as tensile strength, longitudinal and transverse belt stiffness and ply adhesion during a complete belt life cycle.

This exercise not only demonstrated the need to set up a data collection service which would include control of field trials, but also showed the need for constant reviewing of published belting literature to ensure that the greatest possible benefit was being obtained from other researchers' work. Some proposals for the scope of a data collection exercise are discussed below in Section 4.4.

4.4 Conclusions from The Study of Dunlop Belt Design Practice

Section 4.1, 4.2 and 4.3 respectively have shown that (a) the published Dunlop belt selection procedure is not necessarily adhered to, (b) the design data used in the selection procedure is not always derived from testing techniques relevant to service conditions and

(c) there is no working system for the continuous collection of field data. Thus the current Dunlop practice does not meet all the requirements of an ideal system as summarised at the beginning of this Chapter.

Most of the problems associated with the belt selection procedure basically stem from the fact that there is a lack of available information concerning the belt design property requirements of modern conveyor installations. If this were available, then it would be possible to design, by correlation with service data, service-related assessment methods which could then be used to give accurate design data for publication and use in the belt selection procedure. It is not only the design properties actually used at present that should be studied in service, but every possible feature of belt behaviour should be included. This is necessary because the properties that decide the outcome of belt selection procedures for modern conveyors might, because of future belting developments, become irrelevant. This pattern of events was shown when synthetic materials replaced cotton as the major carcase reinforcement: with a cotton carcase, tensile strength and troughability were the important parameters to consider, whereas now, transverse load support requirements determine most belt selections. Unfortunately the belting industry is still faced with the situation of not knowing how much "load support" should be built into belt constructions.

A similar situation might occur if developments of using films of synthetic fibre instead of fabric plies are introduced into service. For example, it might then prove to be the case that troughability and longitudinal flexibility, which are not critical design parameters for the existing product range, become the most important belt properties to consider. When any modification such as this is proposed for an existing product it is essential to know how the original performed its task and how the modification would perform in comparison. One possible way of carrying out this comparison would be to obtain a complete collection of service data from the original product and then carry out exactly the same procedure for the new product. An exercise along these lines for belting would represent an extremely expensive task because of the many different sorts of service conditions that would have to be considered. A much better method is to have service-related tests which could be carried out relatively quickly on small samples of the product. If the proposed modification had favourable results in these tests then it should be safe to assume that the service performance would also be satisfactory. Thus, service-related tests are not only essential for assessment of the design properties of the existing product range, but are also required for development purposes.

Section 4.2 showed that few standard belt tests having proven correlation with service performance are available. The property of most concern from this point of view is that of transverse load support because this determines the majority of current belt selections and, as shown above, the published data for this property has no logical basis apart from "experience". If service data is not to be immediately available for this property, the next best arrangement is to reproduce, as nearly as possible, field conditions in a controlled test project. This is essential, anyway, to find the performance criterion (or criteria) that will be used to assess this property in service. (An experimental programme carried out to do this is described in Chapters 7 and 8.)

However, the only long-term solution to the major problem of the belting industry, that of relying on rules extrapolated from experience with older types of belt constructions, is the use of a data collection service to establish the factors affecting belt performance on modern conveyors. Section 4.3 has shown that the Dunlop organisation has not yet fully devised such a system.

A belt data collection system would require centralised collection of information from six sources. (The advantages of centralisation have been stressed by the National Coal Board Approvals Section [102] .) These six sources are:-

(1) Production.
(2) Sales.
(3) Control Testing.
(4) Development and "Service-Related" Testing. ("Service-Related" is in inverted commas because no correlation with service has yet been proven.
(5) Site Visits.
(6) Field Trials.

Suggestions for the information that should be collected are given in Appendix 6. A belt numbering system common to all departments should be devised so that no confusion could arise between the different sources of information. It is realised that it is impracticable to collect all the required information from every belt sold, but it is vital to ensure that as much data as possible is recorded, and that <u>complete</u> data is made available from field trials. A thorough Field

Trial Practice must therefore be devised and the work of the National Coal Board described in Section 2.5 would provide a useful foundation for such practice.

Setting up such a system would be an expensive and long-term project. (A computer facility would probably be required for storage.) However, it is the only method of obtaining a data base for analysis which would, eventually, lead to:-

(a) A rational belt selection procedure for present-day conveyor installations.

(b) Valid construction data to be used in a belt selection procedure.

(c) A complete picture of the belting market and its requirements.

(d) Correlation of testing methods with service performance.

(e) Belt data to be used as a standard for future developments.

It is therefore essential that such a centralised system be set-up as soon as possible.

CHAPTER 5

SIMPLIFYING THE CURRENT DUNLOP PRACTICE OF BELT SELECTION

The Dunlop belt selection procedure, as described in the Starflex manual [92], and the basis for it have been critically discussed in Chapters 3 and 4. The existing basis for the procedure will be unchanged until significant amounts of new data are obtained, but the procedure itself may be improved. In this chapter proposals, originally put forward in a previous report [103] by the present author, for a more "streamlined" approach to belt selection are described; their relative merits and the reaction to them from Dunlop Belting Division Sales Department are discussed and a procedure for adoption is recommended.

If the Dunlop belt selection, summarised in diagrammatic form in Figure 3.10.1, were to be carried out thoroughly, the time taken could easily be thirty minutes, and up to fifty percent of the time would be spent on the belt working tension calculation stage. This stage requires no subjective judgement whatsoever and involves only simple, but tedious, arithmetic which could lead to errors occurring. It was realised, therefore, that there was most scope for speeding-up the procedure at this stage and so, although every part of the process was examined thoroughly, most proposals for simplifying the procedure related to the tension calculation. It should also be noted that whilst experience of the selection procedure was being gained with specification details of actual conveyors, it was realised that the consideration of troughability rarely, if ever affected the final outcome of the belt selection procedure. This demonstrated that, as expected, troughability is not a critical parameter for modern belt constructions.

5.1 Proposals for simplifying the use of Dunlop's present belt selection procedure

One possibility for achieving simplification in the selection procedure is the use of graphical methods, but the large number and range of variables involved in the complete maximum belt tension calculation make it very difficult to utilise such techniques without having to set up an unmanageable number of graphs. However, if only the effective tension, T_e, is found graphically, then the number of variables is reduced to a workable size. (The effective tension has always to be found, whatever the conveyor configuration, in order to find the power requirements of the belt drive.) The first two proposals given below describe ways in which this can be done. As at least ninety percent of all conveyors have the form of a head-drive level or elevating conveyor, this is the situation assumed in all further work in this chapter. This allows the use of formulae given in Chapter 3.

Proposal 1 : Effective Tension calculation by simple linear graph

Equations 3.3.2, 3.3.3 and 3.3.4 show that in the effective tension calculation the tension components T_a and T_b are linearly dependent on L_1, the adjusted conveyor length, and that T_c is linearly dependent on H, the conveyor lift. Thus the effective tension equal to the sum of these three components can be reduced to the sum of two components: one dependent on L_1, the other on H. It is a simple matter to find these two components from a graph and then add them together to find the final total effective tension.

The two components are given by:-

$$T_i = (F_B \times P \times F_L \times M) \times 9.8 \times L_1 \qquad (5.1.1)$$

$$T_{ii} = M \times 9.8 \times H \qquad (5.1.2)$$

The symbols F_B, P, F_L and M are as defined in Chapter 3. In most circumstances F_B is equal to F_L and so T_i can be reduced to:-

$$T_i = (P + M) \times F \times 9.8 \times L_1 \qquad (5.1.3)$$

where $F = F_B$ and F_L

If T_i is plotted against L_1, a linear graph with a gradient of $(P + M) \times F \times 9.8$ will result. Conversely, if a line of gradient $(P + M) \times F \times 9.8$ is drawn relative to an L_1 axis, it will be possible to read off the tension T_i corresponding to various L_1 values. Similarly a line drawn at a gradient of $M \times 9.8$ to an 'H' axis allows T_{ii} to be found. An example of how this can be done is shown in Figure 5.1.1.

Referring to Figure 5.1.1, suppose that, for a certain conveyor, the adjusted length, L_1, is 200 m and the vertical lift, H, is 45 m. The load weight, M, is 12 kg/m and the weight of moving parts, P, is 24 kg/m. The friction factor, F, is taken as 0.03, that for "average conditions". These are all typical values of the parameters mentioned and can be obtained from the tables given in Appendix 5.

$$\text{Thus,} \quad P + M = 36 \text{ kg/m}$$
$$M = 12 \text{ kg/m}$$

In order to find T_i a ruler is placed between the origin and the value of $P + M$ (in this case 36) on the right hand side of the graph. This is represented by the continuous line in Figure 5.1.1. As the $(P + M)$ scale has been adjusted to allow for the factor $F \times 9.8$ which appears in the T_i calculation, it is possible to read off the value of T_i directly in Newtons at the intersection of the required value of L_1 and the ruler.

$$\text{In this example,} \quad T_i \simeq 2100 \text{ N}$$

A similar procedure is carried out for T_{ii}, a ruler being placed between the origin and the quoted value of M. (This is represented by the broken line.) T_{ii} is then read off the tension axis at the appropriate value of H. (The M scale at the right hand side of the graph has been adjusted to allow for the factor 9.8 in the calculation.)

$$\text{Here,} \quad T_{ii} \simeq 5300 \text{ N}$$
$$\text{and so} \quad T_e = T_i + T_{ii} \simeq \underline{\underline{7400 \text{ N}}}$$

The exact effective tension value calculated arithmetically is 7408.8 N, and so, the errors introduced by this graphical technique are easily

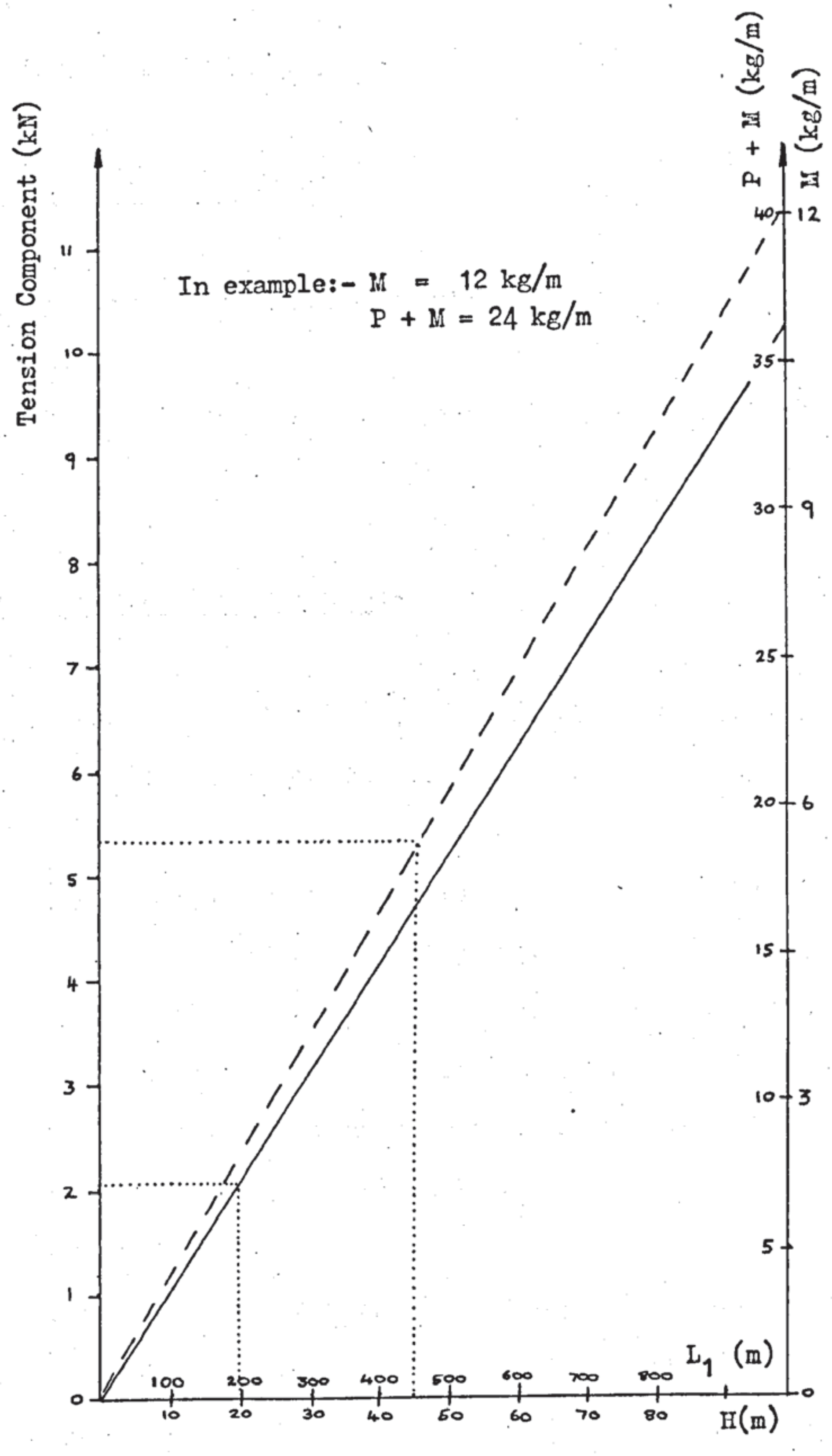


FIGURE 5.1.1

Example of effective tension calculation

by simple linear graph

within acceptable limits.

Advantages of this method:-

(a) Very basic arithmetic only is required.

(b) Many different values of the parameters L_1, H, P and M can be included on one graph.

Disadvantages of this method:-

(a) Only one value of F, the friction factor, can be included on one graph.

(b) Although the procedure is simple, it can take as long as an experienced belting engineer takes to carry out the present technique. However, once similar experience is gained with the graphs, it is likely that they will prove to be quicker and the user will be less prone to make mistakes.

(c) As mentioned above, only the effective tension is found and the maximum tension must still be calculated. Under some conditions it is only necessary to multiply T_e by K, the drive factor, to arrive at the maximum tension. If this was always the case it would be an easy matter to adjust the graph axes' scales to allow for this additional factor. However, an alternative maximum belt working tension, that required to limit sag between adjacent idlers to an acceptable level, must always be considered as described in Section 3.3.

(d) The two scales on some of the axes could cause confusion. This, of course, could easily be overcome by having two separate graphs, but this would then increase the time required to carry out the procedure.

Proposal 2 : Effective tension calculation by carpet curve

The next proposal for speeding up the effective tension calculation involved the use of carpet curves. Carpet curves allow the graphical determination of a variable which is dependent on two other mutually independent variables. In this case T_e is the variable to be found and the conveyor length and height form the mutually independent variables. The method of using this technique for effective tension calculation is shown in Figure 5.1.2.

A network is drawn on the paper to represent different values of conveyor length and lift. The basis for the network is purely practicability; the grid size is chosen to give convenient increments of the variables represented. The effective tension corresponding to the required combination of length and lift can be read off directly from the axis alongside the network as described in the example below.

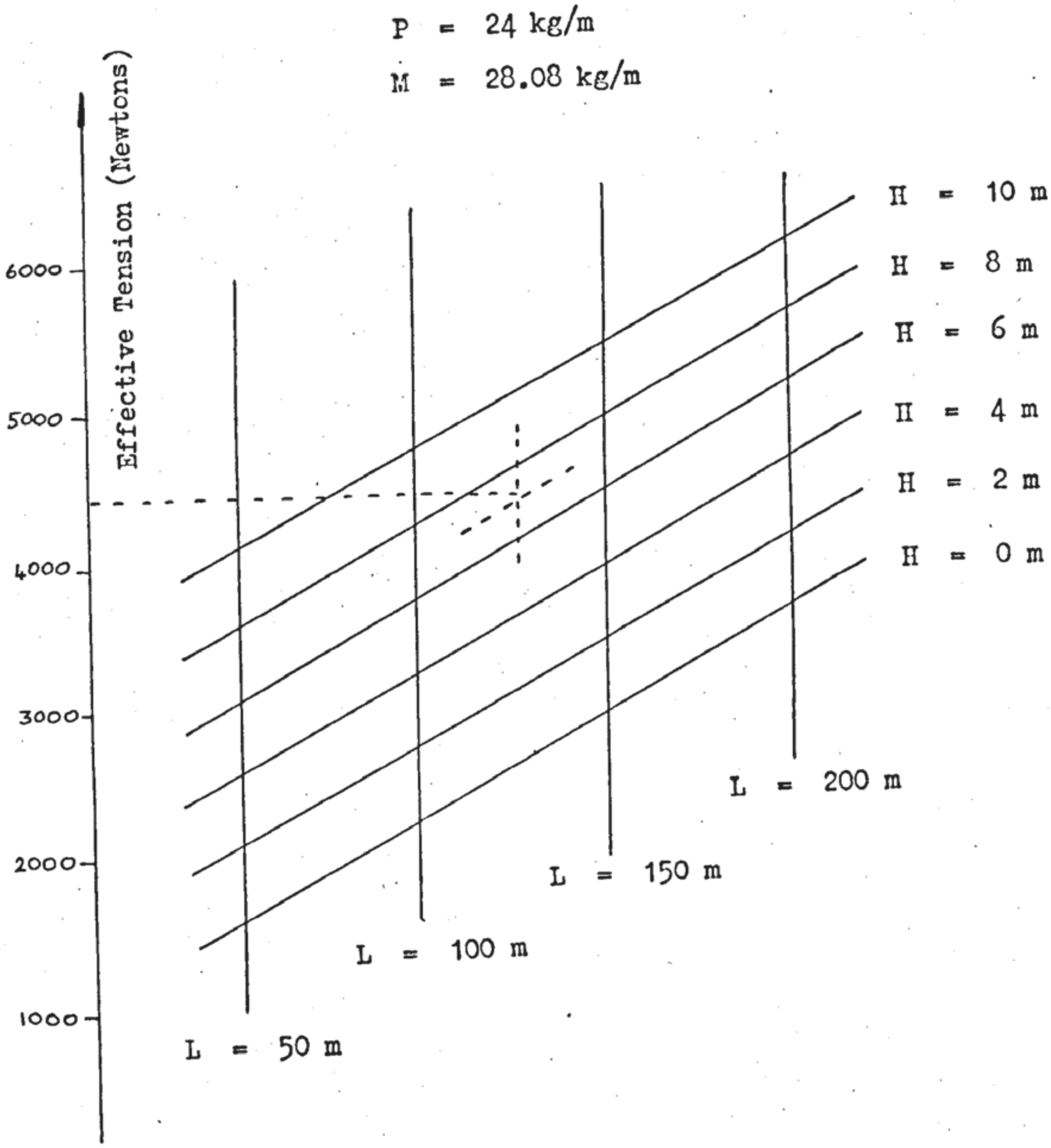


FIGURE 5.1.2

Example of effective tension calculation

by carpet curve

Referring to Figure 5.1.2, assume a conveyor with an actual length, L, of 125 m between the end pulleys and a lift, H, of 7 m. (Note that in this case the actual conveyor length and not the adjusted length, L_1, is used because with carpet curves the length adjustment, L_a, which is always 45 metres for the conveyor configuration being considered, can easily be allowed for in the scale of the axes.) The weight of moving parts, P, is 24 kg/m and the load weight is 28.08 kg/m. A friction factor, F, of 0.03 is used once again. This, in fact, corresponds to a 500 mm wide belt fully loaded with material of density 1.0 t/m^3 working under "average conditions".

The intersection of the lines corresponding to $L = 125$ m and $H = 7$ m is found. This is shown by the intersection of two broken lines in the diagram. The effective tension is then read off from the axis at the side.

In this case, $T_e \simeq \underline{4500\ \text{N}}$

The actual calculated value is 4529.24 N, and so, again, the errors introduced by using this technique are not significant.

Advantages of this method:-

(a) No arithmetic whatsoever is required.

(b) The procedure is very simple and leads immediately to the effective tension value.

(c) One carpet curve can cover many values of conveyor length and lift.

(d) As the network on the carpet curve consists of straight lines there are only a few calculations to be carried out to make up one graph.

Disadvantages of this method:-

(a) Again only the effective tension is found.

(b) It is only possible to cover one value of F, P and M on each carpet curve. It would therefore be necessary to have a selection of graphs available, each corresponding to one particular combination of these variables, and to select the appropriate one prior to application. This is not an insurmountable difficulty as there is only a finite number of practical possibilities for these parameters.

The two graphical methods presented above for simplifying the belt selection procedure both have the fairly major disadvantage of only calculating the effective tension. The two suggestions below, are, perhaps, more useful in that they are attempts to simplify a much greater proportion of the process.

<u>Proposal 3</u> : Selection of belts by load support characteristics only

When carrying out the belt selection procedure for actual installations it soon becomes obvious that, for the majority of cases the tension calculation is unnecessary because load support turns out to be the controlling factor with tensile strength requirements satisfied automatically. (This demonstrates the need for valid load support data and justifies an experimental exercise to investigate this property as will be described in Chapters 7 and 8.) An investigation was carried out to try and identify the application areas where load support determined the final belt construction and then present the information so that it would be possible to detect whether or not the correct belt carcase for a particular task could be selected by immediate reference to load support data only. If this were possible it would mean that a tension calculation which would later prove to be redundant would not have to be carried out.

The investigation to show the feasibility of such a system is described below, and, for convenience, the relevant specification details of the belt constructions included in the investigation are given in Table 5.1.1. (Note that the maximum belt width figures act as load support data.) Only the most common carcase constructions in the Starflex range were considered for this analysis; some of the constructions mentioned in the Starflex manual are not normally held as stock items and would only be considered for selection if a customer enquiry was for such a large quantity of belting that it would be economical to manufacture it specially.

Belt Construction	Maximum Working Tension (kN/m)	Maximum recommended belt width (mm)		
		Load Density (t/m^3)		
		Up to 1.0	1.1 - 1.5	1.6 - 2.5
200/2HD	20	900	650	500
315/2HD	31	1000	800	800
315/3HD	31	1200	1000	800
500/3HD	50	1400	1000	800
630/4EX	63	1400	1000	900
630/4HD	63	1800	1400	1200
800/5EX	80	1800	1400	1200
800/4HD	80	1800	1600	1400
1000/4EX	110	1800	1400	1200
1000/5EX	110	1800	1600	1400
1000/6EX	110	1800	1800	1600
1250/4EX	140	1800	1600	1400
1250/5EX	140	1800	1800	1600
1250/6EX	140	1800	1800	1800
1400/5EX	155	1800	1800	1800
1400/6EX	155	1800	1800	1800
1600/6EX	180	1800	1800	1800

TABLE 5.1.1

Technical Data for the common constructions in

the Dunlop Starflex range of belts

NOTE:- The belt constructions are arranged in order of increasing cost.

Table 5.1.1 gives the maximum working tension ratings for these common constructions. It is usual, especially for the lighter constructions in the range, to quote two values for this belt parameter: one applies when mechanical belt fasteners are used and the other is for vulcanised splices. The latter value is the higher of the two because of the better joint efficiency obtained by vulcanisation; the value quoted here is that for mechanical fasteners unless these are not recommended for use with the particular belt construction in mind. This is done because during the work this belt parameter was used as a limiting tension to decide whether or not a tension calculation was necessary. It will become clear below that choosing the lower possible value provided a safety factor to ensure that if a statement was made that "for this particular application a tension calculation is unnecessary and the belt selection can be made by immediate reference to load support data" then this was an unquestionable fact. (In between the belt applications which will definitely have the final construction decided by tensile strength and those which will definitely have it decided by load support there will always be a band where either, or both, of these properties could determine the result. The work described in this section always tended towards "safety" even though it meant that there would still be certain cases where the tension calculation would be carried out and would later be found to have been unnecessary.)

Before carrying out the investigation it was necessary to show that its objectives were theoretically possible. Two examples are given to show that they were:-

(a) If a load of bulk density 2.5 t/m^3 was carried on a belt 500 mm wide, the maximum width figures show that a belt of construction type 200/2HD has adequate load support. Thus, if it was known that the tension calculation for such an application would result in a figure of less than this belt's rated working tension (20 kN/m in this case), it would be possible to go straight to the load support data and select the correct belt. If the result was going to be greater than 20 kN/m then the 200/2HD construction would not be strong enough and another belt having a higher rated working tension would have to be used.

The example above could be thought to be a special case because of extreme conditions of very dense load and narrow conveyor. The next example shows that this is not so:-

(b) If a belt 1400 mm wide was carrying a load with bulk density of 1.5 t/m^3, a belt construction of at least the type 630/4HD would have to be used in order to provide adequate load support. Even if the working tension was as low as only 3 or 5 kN/m this construction would still be required. If, however, the maximum working tension induced was greater than the belt's rated tension, a stronger construction would have to be used.

These two examples show that, for a given conveyor application, there is a limiting working tension below which a belt can be selected from load support data alone. This limiting tension is 20 kN/m and 63 kN/m for cases (a) and (b) respectively. Thus, a simplified design procedure was shown to be possible in principle and the task reduced to finding a method by which it would be known that a belt working tension would be below the limiting value without having to carry out the actual calculation in every case. In order to tabulate such information, a great many tension calculations would have to be carried out covering all possible belt installations and it would be desirable to present this information in a form convenient for the belt design engineer.

So that a tabular presentation would not lead to an unmanageable number of tables, it was decided always to use the variable values that would give the maximum possible tension for a given situation. That is, it was always assumed that the belt was fully loaded, that the heaviest possible belt was used and that the weight of moving parts was at a maximum. This, again, acted as a "safety measure" and ensured that the maximum working tension that could possibly be induced in a belt for a particular application would be found. Then, if this maximum possible tension was below the limiting value, all belts working in the category being considered, whether fully or partly loaded, could be selected from load support data only.

This meant that the highest relevant values of the variables described by the symbols, F, P, B and M had always to be used. F, the friction factor, was taken as 0.03 (the highest value quoted in the Starflex manual and described as being for "average conditions".) The weight of moving parts, P, and the approximate belt weight, B, are presented in the manual under the headings, "light duty", "medium duty" and "heavy duty"; these terms were quantified as load density ranges 0 - 1.0, 1.1 - 1.5 and 1.6 - 2.5 t/m^3 respectively. (These density ranges being the ones used in the Starflex book.) The maximum possible value of M, the load weight per metre of belt length, was found by multiplying the maximum possible load cross-sectional area found from the manual for a given belt width by the maximum load density in the range considered. The values of the idler pitch to be used when finding the tension needed to limit sag were also to be found from the manual. It was decided that, when presenting the information, one table would represent different lengths, L, and lifts, H, of conveyors for one particular value of the drive factor, K. It should, again, be noted that a summary of all the relevant data published in the Starflex manual is given in Appendix 5.

This work resulted in a set of twenty-seven combinations of parameter values to be used in the calculations. These are presented in Table 5.1.2 and, in this, the variable 'Z' is the limiting tension below which only load support need be considered for belt selection.

Approximately 60,000 combinations of belt parameters had to be analysed for this investigation and the only feasible method of doing this was to write a program for a computer or programmable calculator. Such a program was written to find the maximum tension for a large number of different combinations of conveyor length, lift and drive factor for each set of variables shown in Table 5.1.2. Details of the program and the machine used to carry it out are given in Appendix 7.

Load Density (t/m^3)	Belt Width (mm)	Weight of moving parts (kg/m)	Belt weight (kg/m)	Load weight (kg/m)	Idler Spacing (m)	Z (kN/m)
1.0	500	24	4.1	28.1	1.65	20
1.5	500	36	6.2	42.3	1.50	20
2.5?	500	36	10.3	70.3	1.43	20
1.0	650	30	5.3	50.9	1.65	20
1.5	650	47	8.0	76.5	1.50	20
2.5	650	62	13.3	127.3	1.43	31
1.0	800	39	6.6	79.8	1.50	20
1.5	800	60	9.9	119.8	1.35	31
2.5	800	77	16.4	199.6	1.28	31
1.0	900	45	7.4	128.5	1.50	20
1.5	900	70	11.0	192.4	1.35	31
2.5	800	91	18.5	320.8	1.28	63
1.0	1000	49	8.2	189.3	1.35	31
1.5	1000	77	12.3	283.8	1.20	31
2.5	1000	102	20.5	473.2	1.13	63
1.0	1200	63	9.8	261.9	1.20	31
1.5	1200	95	14.8	393.1	1.00	63
2.5	1200	128	24.6	655.0	0.93	63
1.0	1400	71	11.5	345.0	1.20	50
1.5	1400	110	17.2	517.6	1.00	63
2.5	1400	148	28.7	862.6	0.93	80
1.0?	1600	96	13.1	441.5	1.20	63
1.5	1600	127	19.7	661.9	1.00	80
2.5	1600	173	32.8	1103.4	0.93	110
1.0?	1800	142	14.7	549.3	1.20	63
1.5?	1800	170	22.2	824.0	1.00	110
2.5	1800	198	37.0	1373.3	0.93	140

FIGURE 5.1.2

Belt design parameter combinations used in the investigation of belt selection by load support data only

NOTE:- The sets of data with a question mark next to the load density figure are, in fact, unusual combinations. (For example, it is unlikely that a 500 mm belt would be carrying a high density load of 2.5 t/m^3.)

The results were presented in the form of a set of tables, each table referred to one particular belt width, b; different conveyor lengths and lifts were represented and results for the three density ranges referred to above were shown by three different lines. If, before the tension calculation was carried out, the design engineer referred to the table and found that details of the installation being considered fell above the line relevant to the load density being carried, the correct belt construction could be selected by immediate reference to the load support data and thus a considerable time saving would be made. An example of one of the tables is given in Figure 5.1.3 and the method by which they are used is as follows:-

(1) Find the table corresponding to the belt width and drive factor being considered; if the exact drive factor is not available, it is necessary to go to the nearest higher value.

(2) Find the intersection of the conveyor length and lift. If this intersection lies above the line corresponding to the density range of the load being carried, there is no need to carry out the tension calculation, and the correct belt selection can be made by considering only the load support data. If, however, the intersection is below the relevant line, the tension calculation should be carried out.

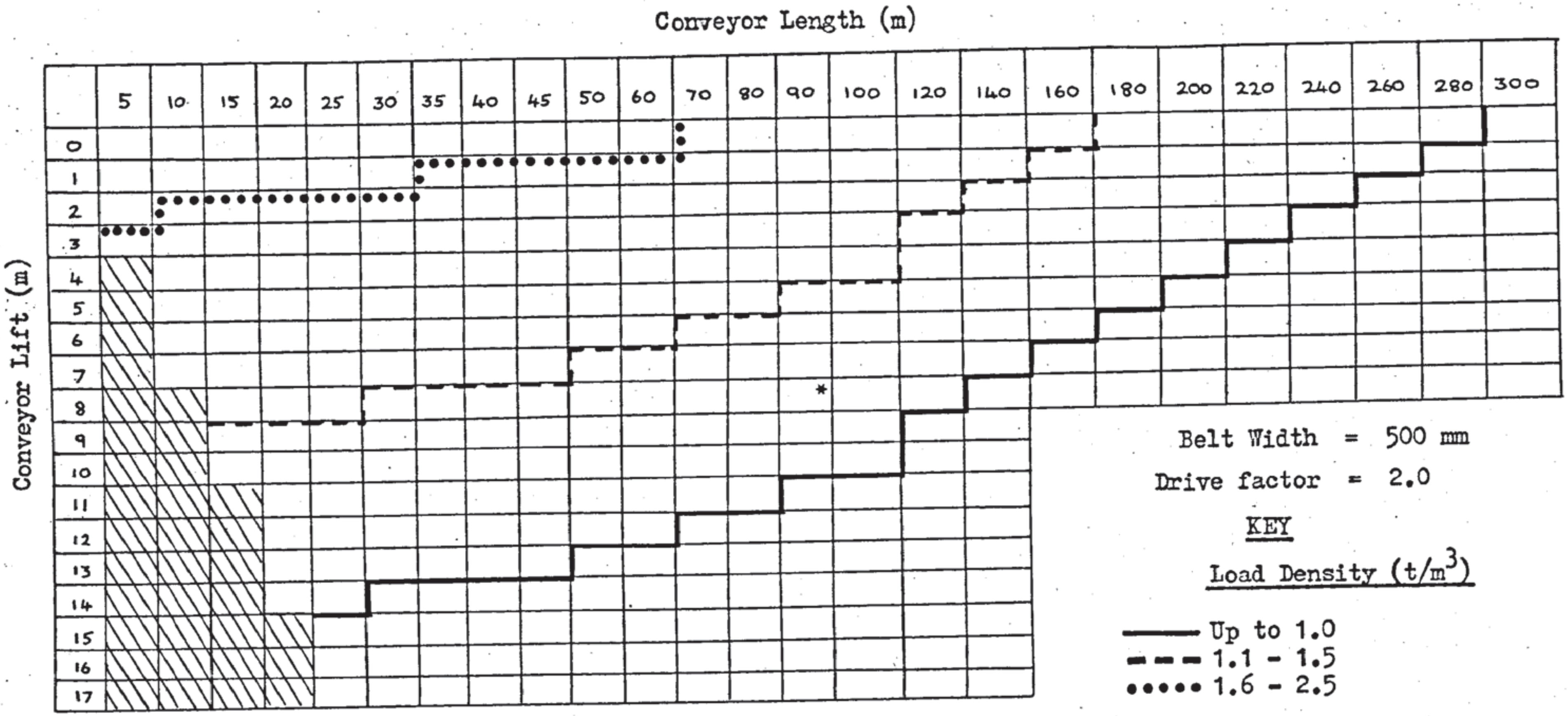


FIGURE 5.1.3

Example of table presenting results of investigation of the feasibility of belt selection by load support data alone

NOTE:- Belt selections for conveyor installations above the applicable load density line are determined purely by load support.

As an example, consider a belt 500 mm wide carrying a load of density 0.8 t/m^3. The conveyor is 90 m long, has a lift of 8 m and has, from the manual, a drive factor, K, of 1.92. Figure 5.1.3 is the table corresponding to this situation and the continuous line identifies the density range for the load being carried. The intersection of the conveyor length and lift, shown by the asterisk, lies above the continuous line so belt selection can be made immediately from the load support data: reference to this data in the Starflex manual shows that a construction of the type 200/2HD would be adequate. If, however, the load density had been 1.3 t/m^3 the zone above the broken line would be appropriate; in this case the asterisk lies below the line and so the tension calculation and the full belt selection procedure must be carried out.

This example has shown how the tables can be used to dispense with the tension calculation of the belt carcase selection procedure. This was the original objective of the study, but there is still scope for improvement. Some of the refinements possible are discussed below in the summary of the main advantages and disadvantages of the system.

Advantages of this method:-

(a) This system simplifies the whole belt selection procedure for certain conveyor installations.

(b) No calculations whatsoever are required by the system.

(c) Using a similar technique of investigation it might be possible to refine the system so the tables not only show that a belt selection can be made without carrying out the tension calculation but also actually show the construction to be used. This would be a form of end-use selection rationally based on the procedure and data in the Starflex manual. Such an investigation would have to be of considerable depth and the belt parameter values used would have to be carefully chosen.

Disadvantages of this method:-

(a) Not every belt selection is simplified. Even if the system was 100% efficient at showing where the tension calculation was not needed this would still be the case.

(b) As mentioned previously there is a region under each density range line where the rated tension requirement could be calculated and still be found to be less than the limiting tension. This could occur, for instance, with a belt which is not carrying its maximum possible load. As great care has been taken always to choose the calculation parameters that give the highest tension, this situation could arise quite frequently. Using the most common variable values rather than those giving the highest possible tension would partially overcome this problem: much greater care in choosing the correct table to use would then have to be taken and the number of tables required to present the data would be greatly increased.

(c) Any changes in the Starflex construction range or the data applicable to it would mean that the whole system would have to be reviewed and brought up to date. This is a fairly major fault.

It has already been stated that approximately 60,000 full belt tension calculations were carried out for the above analysis: if these were done at the rate of one every ten minutes (a difficult average to keep up), it would take one person approximately five working years to complete them. This fact shows the benefit of carrying out the tension calculations by computer, and as discussed in the next proposal, one arrives at the possibility of computerising the whole selection process.

Proposal 4 : Belt Selection by Computer

A programmable calculator was used for the investigation described above; in the course of this it became obvious that the complete belt selection procedure, including cover selection, would be an ideal logical sequence for programming if it was modified slightly. This can be seen if the diagrammatic representation of the belt selection given in Figure 3.10.1 is changed into flow-diagram form as shown in Figure 5.1.4.

All relevant information about belt selection, at present "stored" in the Starflex manual, could be retained in a data-store which would be searched during the computer program in order to find the correct caloulation factors corresponding to the characteristics of the conveyor installation being considered.

Advantages of this method:-

(a) Using a computer for the belt selection procedure is very versatile, because either the whole procedure or any individual part of it could be programmed depending on the capacity and sophistication of the machine being used.

(b) No calculations whatsoever are required from the operator of the process. All that is required is that the conveyor data has to be fed into the machine.

(c) Intermediate stages of the procedure could be printed out; for example, the effective tension and the required motor horsepower could be worked out. Once the initial system is set up, very little extra effort is required to introduce refinements like these.

(d) This method would be extremely fast; this advantage would be really apparent with proposed large conveyor schemes such as would be required in a steel processing plant. In these situations an "original equipment manufacturer" usually submits all the proposed conveyor configurations and asks for suitable belt specifications. The site surveys (mentioned in Chapter 4) carried out by Dunlop Service Engineers could also be quickly analysed by machine.

(e) A complete printed record of the selection procedure would be prepared including a copy of the original input data as a check for the system operator.

(f) A computer or programmable calculator could only use the system written into it, thus ensuring consistency of results as the operator could not introduce personalised end-use selection rules. If, however, any end-use selection rules were shown to be valid they could easily be written into the program.

(g) It would be very easy to make any necessary adjustment to the data store if any alterations occurred in the data relevant to the belt constructions available.

Disadvantages of this method:-

(a) For a single belt calculation to be carried out it might be quicker (but certainly not easier) to work out the result "by hand" unless the machine to be used was readily accessible to the belt designer. In this circumstance it would be no use if the belt designer had to take his place in a queue of people waiting to use the machine for other work.

(b) The initial outlay for a suitable machine is fairly high. This disadvantage would not apply if the program could be used on an existing computer system.

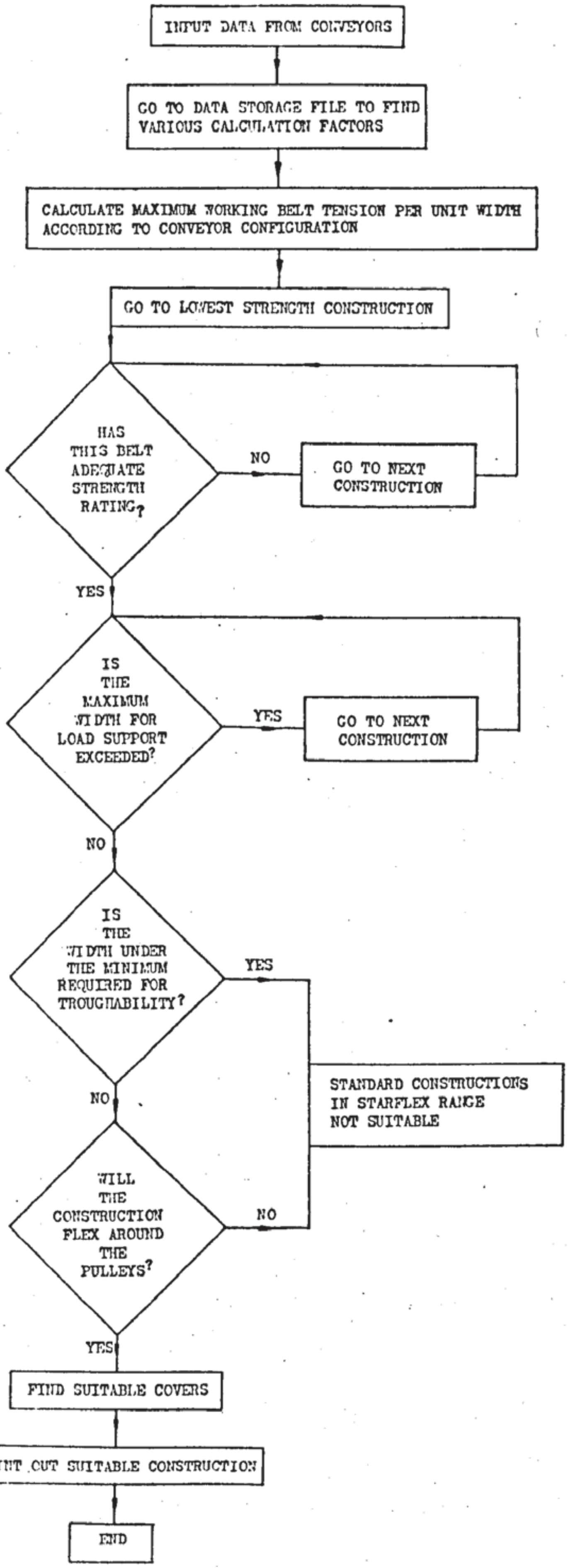


FIGURE 5.1.4

Flow diagram of Dunlop belt selection procedure suitable for programming

The four proposals described above for streamlining the belt selection procedure were circulated in a report [103] to all people concerned with belt specification: reactions to the proposals are described in the next section.

5.2 Reaction to the proposals for streamlining the belt selection procedure

The work involved with belt selection within the Dunlop Belting Division organisation falls into three categories as described in the following paragraphs:-

(i) Work in the first category is that resulting from enquiries by conveyor original equipment manufacturers, and is usually concerned with large materials handling schemes proposed for new plant. On average only four such schemes are dealt with in a year, but two or three conveyor structure manufacturers tendering for the work request belt details and prices. There are always slight variations in the specifications from the different manufacturers, and so, effectively, there are approximately ten schemes to be dealt with per year. The number of belts involved in each may range from twenty to two hundred.

(ii) The second category of work is concerned with incoming enquiries about replacement belts for existing conveyors. On average this is approximately eighty enquiries per month, but only about fifty percent of these are analysed by the full published belt selection procedure; the remainder are dealt with "by experience" as described in Section 4.1.

(iii) The third category is the work resulting from the site surveys described in Section 4.3. On average, one survey per month is carried out and again the number of conveyors involved may range from tens to hundreds. At present the results of approximately three hundred and fifty surveys are filed.

Basically, the three different categories of work are dealt with by three different small departments and so, obviously, belt selection is only part of the workload of these. For example, those people analysing survey data also carry out the actual site visit.

In total, there is enough work to keep at least one person occupied full-time on belt selection. If such a person were employed, this would be the situation that would most benefit from simplification of belt selection, as it would help to ensure that the procedure was actually used every time and so give consistency of results. However, there is a certain amount of reluctance from the various departments to such a re-allocation of workload as they wish to carry out their own belt selections. The main concern is therefore the speeding-up of the process rather than its simplification because they are, of course, already fully conversant with the published procedure for belt selections.

The first attempt at streamlining the belt selection procedure involved the use of simple linear graphs for effective tension determination. This rather trivial development did not really speed-up the calculation time or provide any other significant advantages and so did not generate any real interest. In any case, it was superceded by the proposal for using carpet curves which represented a more refined graphical method of effective tension calculation. In dealing with the suppliers of original equipment, the effective tension is required for power specification of the drive unit (see Equation 2.2.6). Carpet curves were accordingly found useful for this category of work even though the whole belt selection procedure was not significantly speeded up.

Both of the graphical methods for procedure streamlining were rather overshadowed by the computer work, and it was this that drew most reaction from those concerned with belt selection. It was realised that, from experience, some belting engineers would have been able to rule out almost as many unnecessary tension calculations as the computer did. The point of the exercise was, of course, to establish, on a rational basis, the situations where belt constructions could be selected by immediate reference to load support data. This was achieved, and so it would be a relatively easy matter to refine the technique so that tension calculations could be ruled out for many more conveyor situations. This would require a full survey of the parameter values actually used in the selection procedure and this would lead to a more efficient system where the tables represented the most common installations rather than the "worst cases" in the investigation above. The need for further work on this topic became less, however, when the concept of carrying out the whole belt selection procedure by computer was accepted in principle.

A demonstration of the efficiency of a programming technique was requested and so the tension calculations of approximately one hundred and seventy conveyors were carried out on the University of Aston's computer. These represented one particular customer's requirements on several sites. (Full details of the program used are given in Appendix 8.) In this instance, the whole belt selection procedure was not carried out, but the exercise did demonstrate the feasibility of doing so. Under normal circumstances, as described in Section 4.1, only those conveyors which, from the specification data, appeared to be critical situations would have been analysed and so it is quite probable that several important cases could have been omitted. With the computer, the tension calculation was carried out for every conveyor, and it is interesting to note that the print-out showed one conveyor to have an unusually high maximum working tension: it was later shown that an impossible situation of extremely high capacity on a narrow and slow moving belt had been specified by the original equipment manufacturer and so it was possible to provide a useful customer service by pointing out this error.

In many cases the original equipment manufacturer calculates the belt tension requirements as part of the specification data. Usually, because of lack of time, these are accepted as correct, but with a computer program they could be quickly checked and any mistakes or anomalies would be discovered immediately. An example of when this would have been useful occurred in March, 1976: the conveyor manufacturer's tension estimates were found to be only one third of those according to Dunlop. The tension calculations at this stage of conveyor design are extremely important because, as mentioned above, they decide the choice of several of the equipment components such as pulley bearings and drive units. A computer would therefore make possible a useful customer relations exercise as it could provide a quick and easy tension calculation check.

The usefulness of the computer for dealing with large schemes was now proven and it was accepted that it could also be used for the replacement belt business provided that access to the machine was readily available. It is interesting to note that one aspect of the work thought to be useful was the format of the calculation results. The print-out provided a full and neat record of the tension calculations suitable for filing for future information and for sending to the customer. It also gave the user ready access to all the results thus making it easier to consider standardisation of belt types on site. (Having found the correct belt constructions for each conveyor it is always necessary to consider standardisation on site in order to save the customer from having to keep too many spare constructions of belt in stock.)

In general, the proposal to computerise the belt selection procedure was welcomed by all those concerned with its use. The original intention was that a "mini-computer" such as that used in the present work should be made available. Further developments described in the next section meant that there would be no need for this and that a more sophisticated system would be available in the foreseeable future.

5.3 Conclusion to the work on streamlining the belt selection procedure

Acceptance, in principle, of computerisation of the belt selection procedure meant that the work described in the earlier sections of this chapter had become outdated. All that was now needed was the setting up of a suitable programmed system. As mentioned above, it was envisaged that one or more desk-top mini-computers would be best for the task, but in March, 1976, a review of the computing needs of the division as a whole was carried out. It was therefore possible that the belt selection procedure could be included on a new main-line computer when this became available. In order to provide adequate accessibility for those carrying out the procedure, consoles and visual display units or printers could be made available in their offices. It would then only be necessary to type in the relevant data and an answer would be available within seconds. (It should be remembered that the extra cost of providing these facilities would be marginal compared to that of the main computer.)

There would, of course, be additional advantages with this more sophisticated system; as much greater storage facility would be available, it would be possible to include the costing procedure in the programme; a process which, at present, takes a day or more to complete but, again, only involves simple but tedious calculations to find the total cost of a belt from the present costs of the raw materials. (The raw materials costs stored in the computer would, of course, have to be updated regularly.) All installation data could also be stored in the machine according to certain characteristics, thus giving, after a suitable period of time, a complete market picture which could be used to study whether or not the Starflex range provides the optimum belt construction for the present market requirements. This and many other studies of both commercial and technical nature are, at present, impossible because of the lack of service data available in a convenient form.

The immediate benefits of adopting a computerised belt selection procedure cannot be quantified in terms of money actually saved, because it is likely that such action would result, not in belt design engineers becoming redundant, but in a much improved system making it possible to carry out a full analysis of every incoming enquiry. A more rational approach to belt selection will, however, eventually result in financial benefits because it could lead to rationalisation of the standard product range to fulfil present market needs more efficiently. An exercise which will be described in Section 6.3 demonstrates that the present product range probably does not represent the optimum and large financial savings could be possible if information from the whole range of conveyor installations was analysed with a view to improving the situation. At present this information is not available, but the use of a computer to carry out the belt selections and the subsequent storage of the results would eventually provide it. A proposal was put forward, on this basis, that any new computer for Dunlop Belting Division should include a facility for carrying out the belt selection procedure. The proposal was accepted in principle and so, as a result of the present work, such a facility should become available in the foreseeable future.

CHAPTER 6

ANALYSES TO HIGHLIGHT AND QUANTIFY THE PROBLEM AREAS OF CONVEYOR BELT DESIGN

The general development history of troughed conveyor belting, the published scientific literature, and the comparison of different manufacturers' belt selection procedures dealt with in previous chapters have all suggested that the main problem area of present-day belting design is the provision of adequate transverse load support. Some analyses were carried out to try and quantify this problem and to demonstrate that it justified considerable research effort. The lack of systematically filed information on conveyor installations and the belt constructions used on them meant that the samples of applications available for analysis were fairly restricted. However, it was found that, even after considering this limitation of the basic data, the commercial implications of the results proved, beyond doubt, that an experimental programme should be devised to investigate the mechanism of transverse load support. This chapter describes the analyses and the results which led to this conclusion.

6.1 Analysis to demonstrate the importance of transverse load support

The published Dunlop belt selection procedure has been discussed fully in Chapter 3, and it was shown then that there are four basic criteria that must be met by the belt carcase construction. These are, in order of consideration, that the belt must have:-

(i) Adequate working tension rating.

(ii) Adequate load support capability (This is found from the maximum belt width data.)

(iii) Adequate transverse flexibility to trough correctly.

(iv) Adequate longitudinal flexibility to go round the pulleys.

Criterion (ii) is more correctly defined as transverse load support and it is generally accepted by manufacturers that belt construction plays a major role in determining this property (see Section 3.4) whereas longitudinal load support (or the limitation of belt sag between adjacent idler sets) is governed essentially by the tension induced in the belt. Thus, longitudinal load support becomes one facet of criterion (i), and a full investigation of this is described later in Section 6.2.

Method of Analysis

An analysis was carried out on a sample of installations to determine the frequency with which each of the design criteria decided the final belt construction. The sample used was collected by the author whilst gaining first-hand experience of the Dunlop belt selection practice and represented the enquiries passing through the Technical Sales Representative's office in October, 1974. During this month, a large steel plant scheme comprising 172 conveyors was handled and so two sets of data were analysed, one set included details of the steel plant scheme whilst the other did not. It was thought that the latter formed a more representative application sample as it was not biassed towards one particular market segment. Each of the enquiries was dealt with exactly as described in the Starflex manual so that the selection procedure used was as summarised in diagrammatic form in Figure 3.10.1.

It was found that, for the cases considered, neither criterion (iii) nor criterion (iv) determined the outcome of the belt selection procedure, and, in fact the only effect that either of these could have had on belt selection would have been to show that the normal constructions in the Starflex range were not suitable for the particular application being considered. This did not happen for any of the belt enquiries involved in the analysis.

The analysis thus became concerned only with differentiating between belts that were selected because of their tension rating and belts selected because of their transverse load support capability. Some definitions were introduced to denote these two situations: if the tension calculation was carried out and the cheapest construction having adequate tension rating was also found to have adequate transverse load support, the belt was said to be "determined by tension", but if a stronger and more expensive belt construction had to be used purely to provide the required transverse load support, the belt was said to be "determined by transverse load support".

This method of analysis meant that some belts defined as "determined by tension" would, in fact, have required the selected carcase for load support reasons as well as for the tensile strength considerations that decided it. This situation was brought about purely by the order in which these two design criteria are considered and an example is as follows: if the working tension of a particular belt installation was found to be 48 kN/m, a construction of the type 500/3HD would be required (see Table 5.1.1). If the density of the load being carried on this belt fell within the range 1.1 to 1.5 tonnes/m^3 and the belt width was 1000 mm, the 500/3HD construction would also be required to provide transverse load support. Such a belt would therefore be "of maximum width for the applicable load density range". If the load density had actually been the maximum in the range (1.5 tonnes/m^3), the belt would be "of maximum width and carrying the maximum permissible load density". The number of times that either of these situations arose was shown separately in the results, as such belts, strictly speaking, were selected because of both their tensile strength and their load support capability. The Starflex range of constructions contains some sets of two or more belt carcases which have the same tension rating but different transverse load support properties. For example, the constructions of the types 630/4EX and 630/4HD both have a tension rating of 63 kN/m. If these belts were to be carrying a load with a density in the range 1.1 to 1.5 tonnes/m^3, then the maximum permissible width of the 630/4EX construction would be 1000 mm whereas that of the 630/4HD would be 1400 mm because the construction 630/4HD has better transverse load support capability. This means that a belt installation requiring the tension rating of one of these "sets" of carcase constructions might require the transverse load support of one of the more expensive types in the set. The number of times that this occurred was also recorded in the full set of results [104].

Two important characteristics of the final belt selection were found for every installation; these were the "percentage over-tension" and the "percentage over-cost".

If the calculated belt working tension was found to be T kN/m and the tension rating of the selected construction was T_s kN/m, the percentage over-tension was found from:-

$$\text{Over-tension (\%)} = \frac{T_s - T}{T} \times 100\% \qquad (6.1.1)$$

The percentage over-cost was introduced to give some idea of the cost of providing transverse load-support; it was obtained by comparing the "factory variable cost", C_s, of the final selection with the cost, C, of the belt that would have been recommended if the tension rating was the only design criterion considered.

$$\text{Over-cost (\%)} = \frac{C_s - C}{C} \times 100\% \qquad (6.1.2)$$

Note:- The factory variable cost includes the direct raw materials, labour and power costs of the belt.

The range of Starflex belt constructions considered in this analysis is given in Table 5.1.1, along with the design data relevant to the investigation. When the analysis was completed a full set of results was issued [104]. A summary of these is given in the next section.

Results of the Transverse Load Support Analysis

As mentioned previously, two full sets of results were calculated; the first set included the effect of belt selections for a large steel plant scheme whereas the second set excluded this. Both of these sets were issued in histogram form as well as in tables, in order to provide a greater visual impact. The number and total length of belts of each carcase type were shown and average values of the items such as "percentage over-cost" were quoted. A summary of the most important results over the whole Starflex range of belts is given in Table 6.1.1. This table represents the results for the installations thought to provide the more realistic sample of the total market; that is, they exclude the effect of the steel plant scheme.

The results show that a significant proportion of belt selections were decided by transverse load support considerations; 55% of the belt installations fell into this category, and these represented 29% of the total length of belting. (These two figures were 63% and 45% respectively for the analysis including the steel plant scheme.) The significance of these results is discussed below in the conclusions from this analysis.

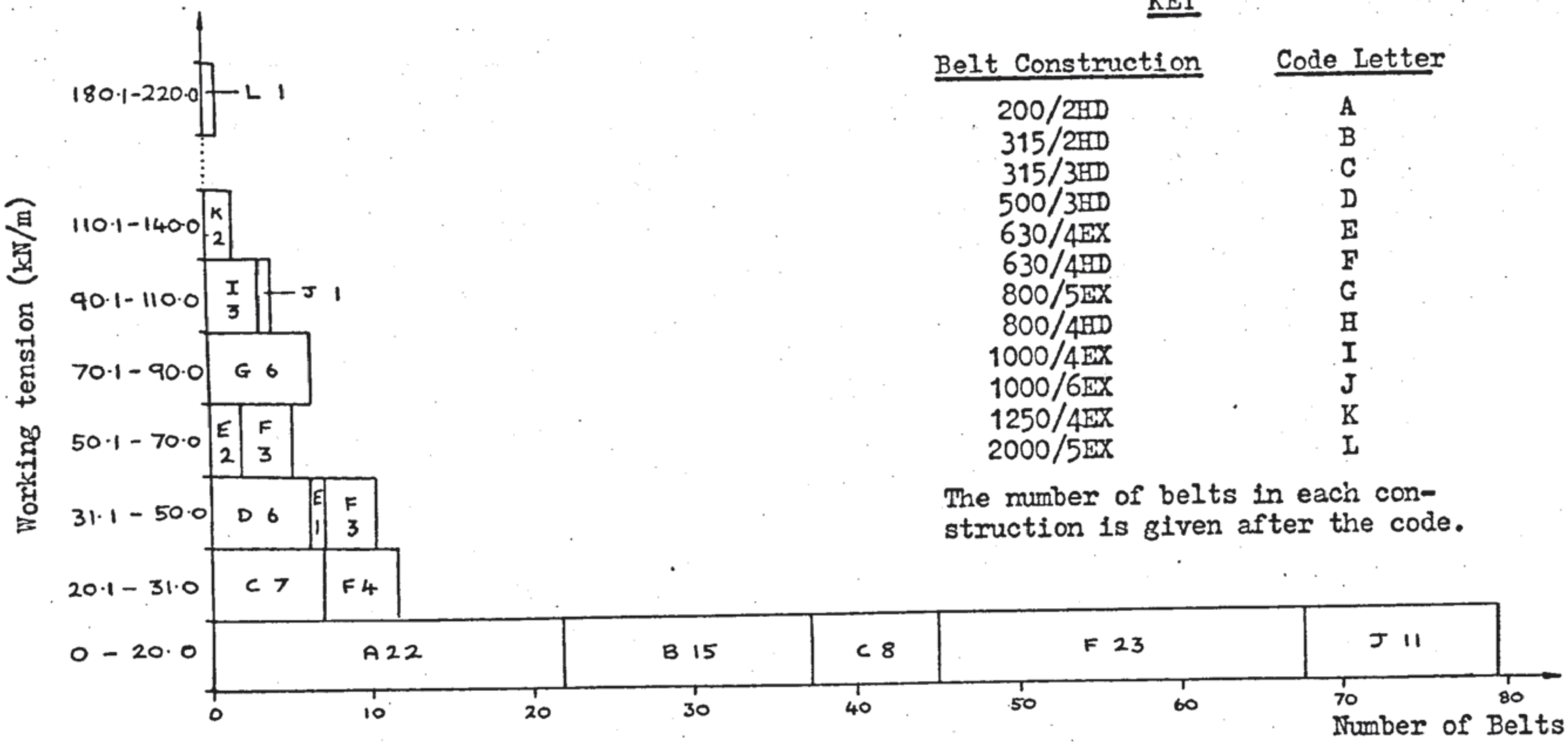
One other important aspect of the work was an attempt to find the combinations of tension rating and transverse load support capability that would "optimise" a belt construction range. This was done by finding the number and total length of belts of each of the Starflex constructions that fell within certain working tension ranges. (See Figures6.1.1 and 6.1.2.) These tension ranges coincided with the

	ALL BELTS	"BELTS DETERMINED BY TENSION"	"BELTS DETERMINED BY TRANSVERSE LOAD SUPPORT"
Number of Belts	118	53 (45%)	65 (55%)
Length of Belting (m)	52196	37010 (71%)	15186 (29%)
Average over-tension (%)	696	158	1136
Average over-tension weighted to length (%)	221	25	699
Average over-cost (%)	24	2	42
Average over-cost weighted to length (%)	15	4	44

TABLE 6.1.1

Summary of results of the analysis

to show the importance of transverse load support



KEY

Belt Construction	Code Letter
200/2HD	A
315/2HD	B
315/3HD	C
500/3HD	D
630/4EX	E
630/4HD	F
800/5EX	G
800/4HD	H
1000/4EX	I
1000/6EX	J
1250/4EX	K
2000/5EX	L

The number of belts in each construction is given after the code.

FIGURE 6.1.1

Breakdown, by construction, of the calculated working tension values of the belts involved in the analysis described in Section 6.1

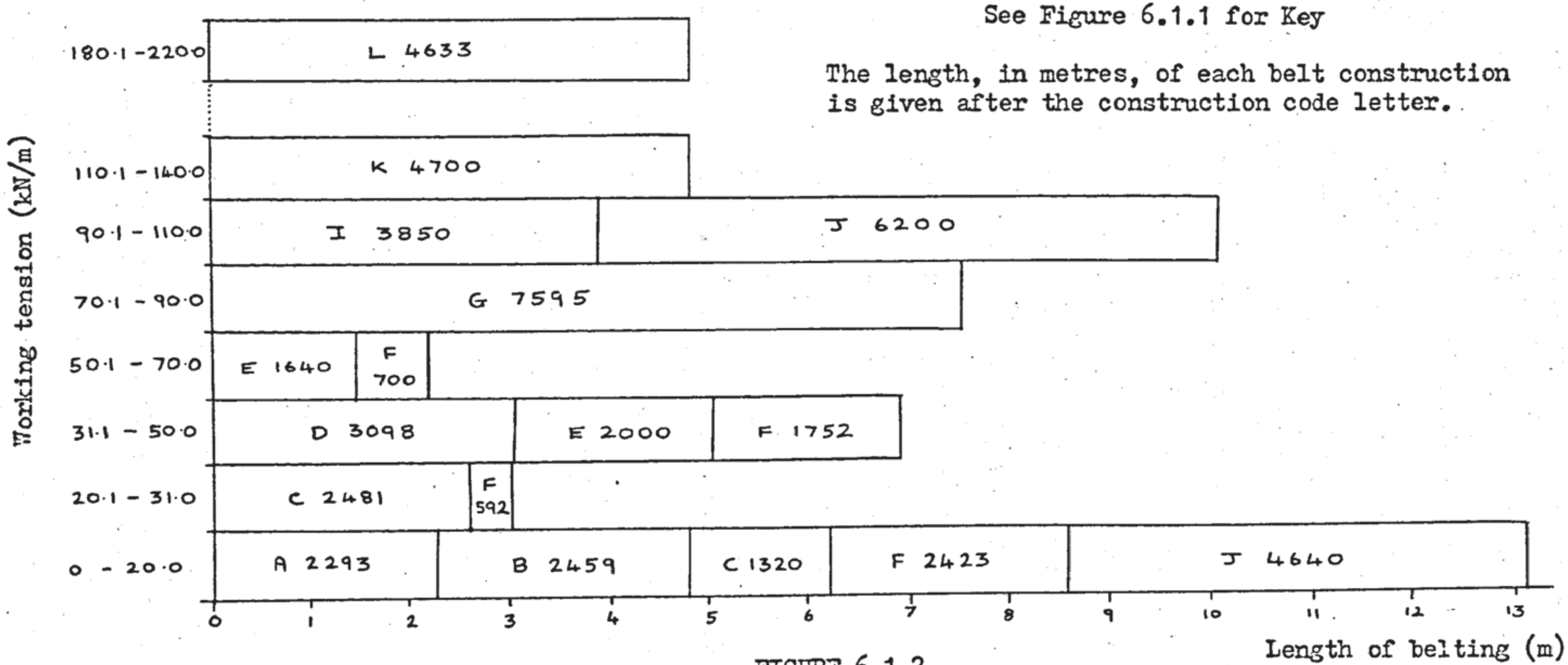


FIGURE 6.1.2

Breakdown, by construction and length, of the calculated working tension values of the belts involved in the analysis described in Section 6.1

tension ratings of the existing Starflex range. It was possible, from this work, to find the number of installations which would be covered by various combinations of tension rating and transverse load support capability. The final results of this work are shown in Tables 6.1.2 and 6.1.3. As an example of their use, suppose a new belt construction had a tension rating of 20 kN/m, that of the present 200/2HD construction, and a transverse load support capability equal to that of the 315/3HD type of construction. Table 6.1.2 shows that this hypothetical belt could be used for 38% of the total number of installations. If the transverse load support of this new carcase construction was improved so that it matched that of the present construction 630/4HD, 58% of the installations would be covered. The table also shows the extra number of installations that would be accommodated by increasing the tension rating. For example, returning to the "new construction" having a tension rating of 20 kN/m and the transverse load support of the 315/3HD type of construction, increasing the tension rating to 31 kN/m would only cover an additional 6% of the belt applications and so 44% of all cases would be covered. These tables can therefore be used to indicate the properties required from an optimum range of constructions, and the significance of these results is discussed in the next section.

<u>Conclusions from analysis to show the importance of Transverse Load Support</u>

The analysis showed that a considerable amount of conveyor belting is selected because of its transverse load support capability. For the sample of installations thought to be more representative of the whole market, 55% of the number of belts were selected in this way, and these

Transverse Load Support equivalent to the construction:-	Tension-rating (kN/m)					
	20	31	50	70	90	110
200/2HD	19					
315/2HD	31	31				
315/3HD	38	44				
500/3HD	38	44	49			
630/4EX	38	44	50	51		
630/4HD	58	67	75	80		
800/5EX	58	67	75	80	85	
1000/4EX	58	67	75	80	85	87
1000/6EX	67	76	85	89	94	97

TABLE 6.1.2

Percentage of total number of belt applications that would be covered by various combinations of transverse load support capability and tension rating

Transverse Load Support equivalent to the construction:-	Tension-rating (kN/m)					
	20	31	50	70	90	110
200/2HD	4					
315/2HD	9	9				
315/3HD	12	16				
500/3HD	12	16	22			
630/4EX	12	16	26	29		
630/4HD	16	22	35	39		
800/5EX	16	22	35	39	54	
1000/5EX	16	22	35	39	54	81
1000/6EX	25	31	44	48	63	82

TABLE 6.1.3

Percentage of total length of belting that would be covered by various combinations of transverse load support capability and tension rating represented 29% of the total belt length supplied. (These figures show that belts selected by tension considerations tend to be longer than those selected by transverse load support: this is a direct result of the working tension calculation which shows that for given load conditions, the longer the conveyor, the higher the working tension required in the belt.) The reliability of the transverse load support data in the Dunlop Starflex manual is questionable, and the Starflex procedure for belt selection, in common with those of several other belt manufacturers, does not really make prominent the importance of having to consider transverse load support. In fact, it tends to suggest that tension rating is the most important belt characteristic whereas the present analysis has shown that less than half of the number of installations have the final belt construction determined by tension.

Since, from the analysis, "satisfactory" load support implies adequate tension rating in many cases, time is wasted when the working tension calculation is carried out during the selection procedure in these situations. This result suggested that it would be useful to identify the situations where this occurred so that belt selection could be made by immediate reference to transverse load support data without having to carry out the tension calculations, thus reversing the order in which the two main design criteria are considered in the selection procedure. (The work described in Section 5.1 above was concerned with this.)

An extremely high overspecification of tension rating occurred for the belts analysed; 696% was the mean for the whole sample. It was pointed out by members of Dunlop Belting Division that this overspecification might help to allow for other factors: for example, a conveyor structure might move slightly during service thus causing certain sections of the belt to be under greater tension than originally calculated. A fairly high safety factor of 10:1 for the ratio of actual tensile strength to quoted tension rating is taken into account for the published data. This value is accepted generally throughout the industry for fabric reinforced belting, and it is this that should allow for such adverse circumstances. If, after a survey of operating installations, it was found to be inadequate, then it should be increased. It should be noted that overspecification of tensile strength to meet transverse load support requirements can cause a final safety factor greater than 100:1.

The belts involved in the present analysis were selected exactly as described in the Starflex manual; in practice, the belts could be even more overspecified for a variety of reasons such as:-

(a) Standardisation of belt types on site. This is a commercially sound reason for overspecification. For example, if there were twenty-one belts 1500 mm wide on a particular site and nineteen of them required a carcase construction of the type 500/3HD whilst the remaining two only needed a weaker construction such as a 315/2HD belt, it would be reasonable for the customer to install 500/3HD type belts on all the conveyors. This would then reduce the amount of spare stock belting required.

(b) Availability of certain constructions of belting. A particular type of belting might not be available, and so a stronger construction would be installed.

(c) "Psychological reasons". This would occur mainly at the low tension rating end of the product range. Although a two ply belt might be adequate according to the manual it would appear that certain industries would not accept belts made of less than three plies. This demonstrates the need for customer education about the technical requirements of belting.

(d) The use of personalised end-use selection rules. These have been discussed previously in Chapters 3 and 4. At present such rules have not had their validity proven. (It should be remembered that these might also lead to under-specification.)

Due to lack of service data, it is impossible to quote precise figures for the amount of over-specification that actually occurs in practice. However, the results of the present analysis suggest that it is extremely high. The over-specification would be reduced for some installations if the transverse load support data was presented more fully in the manual. (That is, if smaller load density ranges were introduced.) It must be remembered, however, that the data presented in the maximum permissible belt width table is only based on experience and might, in practice, over or under-specify the transverse load support requirements.

Over-specification has, eventually, to be paid for by the customer and an estimate of the additional price to do this was obtained from the over-cost figures. If new belt constructions were developed having better transverse load support characteristics for the same order of tension rating, then this over-specification could be reduced considerably, thus making the new constructions extremely competitive for a given installation. The combination of belt characteristics that would optimise this situation were found during the analysis. For example, a belt with the same transverse load support as that quoted for the present construction 630/4EX and the tension rating of a 200/2HD construction would cover 38% of the installations (12% of the total length). Improving the transverse load support slightly to that of the construction 630/4HD would then cover 58% of all cases (16% of the total length). From these results (which are summarised in tables 6.1.2 and 6.1.3) it was thought that the two combinations of parameters that would be of most use are:-

(i) Transverse load support quoted for the present 630/4HD construction together with 20 kN/m or 31 kN/m tension rating.

(ii) Transverse load support quoted for the present 1000/6EX construction together with 50 kN/m tension rating.

The first combination would be most useful as it would cover the many applications utilising of short, medium width conveyors whilst the second combination would cover installations with wider belts carrying heavy loads.

Notwithstanding the limitations of the basic data used, the above analysis shows:-

(a) The mechanism of transverse load support should be investigated to determine the contribution to it from various conveyor and belt parameters such as belt stiffness, idler pitch, idler angle etc.,

(b) As transverse load support has been shown to decide more than 50% of belt selections, there is a need for an experimental programme to evaluate, on a rational basis, the transverse load support capability of the present construction range.

(c) There is a need for the development of new belt constructions, particularly aiming towards the combinations of properties described in (i) and (ii) above. Perhaps recent developments of belts using "sheets" of film instead of fabric plies might be able to provide these properties.

In addition to these major points, the analysis again showed the need for service data and also cast doubts on the rationality of the 10:1 safety factor for the ratio of actual tensile strength to tension rating.

6.2 Analysis to demonstrate the importance of Longitudinal Load Support

The previous section has shown the importance of transverse load support in belt selection. Transverse load support is affected by several conveyor parameters and, up to now, no attempts have been made to determine the contribution from each one. In contrast, longitudinal load support has been shown to be governed, for all practical purposes, by the working tension induced in the belt as was described in Section 2.2. Inter-idler sag is the generally accepted measure of longitudinal load support and a maximum sag of 2% of the idler pitch is considered to be acceptable in the Dunlop selection procedure. The tension required to limit sag to this value is found from equation 2.2.10 which is reduced, for the purposes of belt selection, to a more convenient form as described in Section 3.3:-

$$T_o = G \times W \times I \times 9.8 \text{ Newtons} \tag{6.2.1}$$

Where:-

T_o = Tension required to limit sag (N)

W = Total belt and load weight (kg/m)

G = Sag factor

I = Idler pitch (m)

If two percent is to be the accepted amount of sag, G is 6.25 (see Section 3.3).

It is sometimes found that the tension induced in a working belt has to be greater than the minimum required to provide drive in order to ensure that the tension T_o is present at the low tension end of the working conveyor. An analysis [105], described here, was carried out to find the frequency with which this situation occurs, and thus demonstrate the effect of longitudinal load support on belt selections.

Method of Analysis

A large number of belt tension calculations were to be carried out by computer (see Section 5.1). This provided a convenient data source to collect easily the information required for the present analysis. The installations "calculated" on the computer contained a selection of belts ranging from 5 m to 800 m centres length and 500 mm to 1800 mm width. They actually represented one customer's total belting requirements on several different sites. This means that the sample of belts was probably not as representative of all belting requirements as that used in the previous analysis described above. However, as the design factor being investigated here was not considered to be as important as transverse load support (an assumption endorsed by the results), it was thought that the sample of belts taken was adequate for the purpose.

The computer program was used to calculate, for each conveyor, the two possible values of maximum working belt tension as described in Section 3.3: one value considered drive requirements, the other limitation of sag, and these were called $T_m(a)$ and $T_m(b)$ respectively. (The program used is described, in full, in Appendix 8.) The final maximum tension that would be induced in the working belt was then found from the larger of the two possible values. The computer printed out this value, and if the tension required to limit sag, $T_m(a)$, was greater than that for drive requirements, $T_m(b)$, a marker 'SAG' was printed next to the result indicating that longitudinal load support had determined the final belt maximum tension.

Thus, the computer print-out gave the maximum tension in the working belt and also showed which of the two criteria had decided it. The print-out therefore served as a convenient source for the basic data required in the analysis.

The belts were then divided into two groups:-

<u>GROUP I</u> containing those belts where T_m, the maximum tension, was found from the minimum tension required to provide drive and so $T_m = T_m(a)$.

<u>GROUP II</u> containing those belts where the tension was increased in order to limit inter-idler sag and so $T_m = T_m(b)$.

The number and total length of belts in each group were found. The "over-tension" caused by the sag criterion was calculated for each conveyor in GROUP II and was expressed as a percentage of the minimum tension requirement. The definition of "over-tension" was, in this case, given by:-

$$\text{Over-tension (\%)} = \frac{T_m - T_m(a)}{T_m(a)} \times 100\% \qquad (6.2.2)$$

Thus, if $T_m(a)$ was, in fact, the final maximum tension (that is, as with GROUP I belts), then the percentage over tension would be zero. It is important to note that this is the "over-tension" caused by the sag criterion, and not that caused by using a belt stronger than actually necessary, as was the case in the analysis of Section 6.1.

One other term requiring definition was introduced into the results; this was "New Tension Category Belts":- In the Starflex range of belts there are obviously maximum permissible working tensions for each construction. For example, the 200/2HD type of construction has a tension rating of 20 kN/m, and the 315/2HD type has one of 31 kN/m. If, from the computer print-out, the maximum tension found from considering drive requirements, $T_m(a)$, was in the region of the maximum permissible working tension of a particular belt construction, it was quite possible that the sag criterion would require an increase in tension such that the original belt construction was no longer suitable. For example, if $T_m(a)$ was found to be 15 kN/m and the maximum tension after considering sag, $T_m(b)$, was 22 kN/m, a construction of the type 200/2HD would not be suitable, and a 315/2HD type belt would have to be used. Such cases were called "New Tension Category Belts". It was only in these situations that the factor being investigated might increase the cost of the belt to the customer. It is important to remember, however, in these cases it was quite likely that the final construction selection would be governed by transverse load support considerations. The amount of new tension category belting was calculated and presented as one of the results of the analysis.

Results of the Longitudinal
Load Support Analysis

The results of this investigation are displayed in Tables 6.2.1 and 6.2.2.

Table 6.2.1 summarises the overall results of the analysis. It shows the number and length of belts occurring in GROUPS I and II with the average values of belt length, maximum working tension using the drive factor, $T_m(a)$, final maximum tension and over-tension.

Table 6.2.2 applies to "New Tension Category Belts" only and shows the number of times that these occur.

The results showed that 73% of the installations investigated had to have increased induced tension in order to limit inter-idler sag to 2%. These 73% of belts represented only 31% of the total length of belting, thus showing that longer belts tend to have the maximum tension calculated directly from the drive factor and effective tension. (A direct result of the method of tension calculation.)

The "New Tension Category Belts" represented only 14% of the number of installations and these made up 5% of the total length of belting. The levels of "over-tension" caused by this situation were negligible compared to those caused by the transverse load support requirements in the analysis in the previous section.

The significance of these results is discussed in the next section.

	Number of belts	Total length of belting (m)	Average Length per belt (m)	Average tension using drive factor (kN/m)	Average final maximum tension (kN/m)	Average increase in tension due to limiting sag (kN/m)	Average percentage over-tension	Length weighted average increase in tension (kN/m)	Length weighted average percentage over-tension
Group I Belts where drive determined tension	45 (27%)	31629.0 (69%)	702.9	75.85	75.85	0	0	0	0
Group II Belts where sag determined tension	120 (73%)	14462.2 (31%)	120.5	19.59	25.43	5.84	49	3.50	22
All Belts	165 (100%)	46091.2 (100%)	279.3	34.93	39.18	4.25	36	1.10	7

TABLE 6.2.1

Summary of results of analysis to show the importance of longitudinal load support

Number of belts in new tension category	Length of belting (m)	Number as a percentage of Group II belts (%)	Number as a percentage of all belts (%)	Length as a percentage of Group II belting (%)	Length as a percentage of all belting(%)
23	2321.8	19	14	16	5

TABLE 6.2.2

Results of "new tension category" belts

Conclusions from Longitudinal Load support analysis

At first sight it appeared that provision of longitudinal load support affected a large percentage of belt selections because it determined maximum tension in 73% of the cases. However, it only actually affected the choice of belt tension rating in 14% of all cases. This figure was small compared with the 55% of belt selections shown to be determined by transverse load support considerations (in the previous analysis).

It should be remembered that from the results of the previous analysis a large proportion of this 14% of belts would, themselves, eventually have a construction selected because of transverse load support considerations. (55% according to the previous analysis - but probably more because it has been shown here that those belts affected by the sag criterion tend to be at the low end of the tension range, and these were the belts that tended to be "determined by transverse load support".)

It was therefore concluded that 4 - 6% of belt selections would, eventually, be decided by longitudinal load support considerations and these would probably only represent 1 - 2% of the total length of belting. These results showed that the problem of longitudinal load support, providing that it can be overcome by extra induced belt tension, is a minor one compared with that of transverse load support, because the number of belts affected by it is small and the potential savings of over-specification caused by it are minimal. However, the figure of 4 - 6% of belts is not insignificant and so a certain amount of work on this problem is justified if it can be investigated out whilst experiments to study the mechanism of transverse load support are being carried out. The results of the two analyses described were therefore used to allocate experimental time and resources to the most important research areas as will be shown in Chapters 7 and 8.

One other point, which has been discussed in Chapter 3, arises again from this analysis. The limit of 2% sag is apparently arbitrary; German research work [28] has shown that sag severely affects belt power requirements. It would therefore be useful to collect data from working conveyors to find the optimum sag level which provides adequate load support to avoid load spillage and minimises power requirements.

The above two analyses have demonstrated the major importance of transverse load support considerations in belt selection. They have not, however, really shown the commercial implications of needing to know the exact contribution to transverse load support that can be obtained from the various carcase constructions. The analysis described next in Section 6.3 is an exercise which does demonstrate this aspect.

6.3 Analysis showing the need for correct transverse load support data to help rationalise the Starflex product range

The work described in this section was carried out in collaboration with R. W. Sabin, Interdisciplinary Higher Degrees Scheme student, Dunlop Belting Division. [106]

It was realised, from the work described in Section 6.1, that the belt tension rating of a significant number of conveyor installations is greatly overspecified in order to provide adequate transverse load support. This results, of course, in increased expense to the customer. An investigation was therefore carried out to examine the possibilities of providing this transverse load support more cheaply. One possible way of achieving this without any radical changes in the methods of carcase construction was to use fewer plies of heavier, stronger, fabrics than those used in the existing Starflex belt types. Cost reduction would then result from (a) savings in production time due to the lower number of plies, (b) savings in raw materials, and (c) the relatively low incremental cost of heavier fabrics. (This is because part of fabric costs is attributable to production time, which is approximately standard whichever fabric is being made.)

The analysis which is described in this section was therefore carried out to test the hypothesis that belts, fulfilling the technical design requirements, could be provided more cheaply by increased use of heavier fabrics.

Method of Analysis

The constructions in the Dunlop Starflex range of belts are commonly made from one of six fabrics. Other fabrics are sometimes used, but these tend to duplicate the technical properties of one of the common six. It was assumed that each fabric available was used to make a belt of two, three, four, five and six plies. Each possible combination of fabric and number of plies could then be made into a "Heavy Duty" or "Extra" construction depending on the amount of rubber interply used.

Some of these possible combinations would, of course, represent the belt constructions already forming the common Starflex range of belts.

Each resultant belt construction would have certain transverse load support and tension rating characteristics. As mentioned earlier in the present thesis, transverse load support is described, in the Starflex manual, in terms of maximum belt width capable of carrying certain ranges of load density (see Table A5.11, Appendix 5). Belt tensile strength is described in terms of breaking force per unit belt width, so the maximum working tension rating of a proposed construction could be found simply from the actual tensile strength divided by a safety factor of ten. The figures for tensile strength, calculated from fabric strengths, are shown in Table 6.3.1. (The quoted values are the nearest International Standard values to the actual strength.)

The transverse load support data was, unfortunately, more difficult to obtain. As described in Section 4.2, the published figures are not based on proven relationships, and so the required maximum width figures had to be extrapolated from those quoted in the Starflex manual for existing constructions. The results of this are shown in Table 6.3.2.

Other technical properties such as impact resistance and ease of splicing were not considered because even less data was available on these, and they do not form part of the published Dunlop selection procedure.

Factory variable costs (FVC - the production and materials cost) were obtained from the Costing Department for each of the proposed constructions. All figures related to a standard production width of 1550 mm (see Table 6.3.3).

Fabric Coding	Number of Plies				
	2	3	4	5	6
8407	200	315	400	500	630
8814	315	500	630	800	1000
8874	400	630	800	1000	1250
8911	500	630	1000	1250	1400
8961	630	800	1250	1400	1600
8962	800	1250	1600	2000	2500

TABLE 6.3.1

Tensile strength rating in kN/m of possible fabric/ply number combinations

NOTE:- Broken lines contain ratings calculated from the fabric strength. Dotted lines contain the range of constructions normally available.

No. of Plies		2			3			4			5			6		
Load Density t/m^3		Up to 1.0	1.1 to 1.0	1.6 to 2.5	Up to 1.0	1.1 to 1.5	1.6 to 2.5	Up to 1.0	1.1 to 1.5	1.6 to 2.5	Up to 1.0	1.1 to 1.5	1.6 to 2.5	Up to 1.0	1.1 to 1.5	1.6 to 2.5
Fabric and interply coding	8407 Ex	650	500	-	900	650	500	1200*	1000*	800*	1600*	1200*	1000*	1800	1600	1400
	HD	900*	650*	500*	1200*	1000*	800*	1600	1200	1000	1800	1600	1400	1800	1800	1600
	8814 Ex	800	650	500	1000	800	650	1400*	1000*	900*	1800*	1400*	1200*	1800*	1800*	1600*
	HD	1000*	800*	650*	1400*	1000*	800*	1800*	1400*	1200*	1800	1800	1600	1800	1800	1800
	8874 Ex	800	650	500	1200	900	800	1400*	1200*	900*	1800*	1600*	1400*	1800*	1800*	1800*
	HD	1000	800	650	1400*	1200*	900*	1800*	1600*	1400*	1800	1800	1800	1800	1800	1800
	8911 Ex	900	650	500	1400	1200	900	1800*	1400*	1200*	1800*	1800*	1600*	1800*	1800*	1800*
	HD	1200	1000	800	1800	1400	1200	1800	1800	1600	1800	1800	1800	1800	1800	1800
	8961 Ex	1000	800	650	1400	1200	900	1800	1600	1400	1800*	1800*	1800*	1800*	1800*	1800*
	HD	1400	1200	1000	1800	1600	1400	1800	1800	1800	1800	1800	1800	1800	1800	1800
	8962 Ex	1200	1000	800	1800	1400	1200	1800	1800	1800	1800	1800	1800	1800	1800	1800
	HD	1600	1400	1200	1800	1600	1400	1800	1800	1800	1800	1800	1800	1800	1800	1800

TABLE 6.3.2

Maximum permissible belt widths for given combinations of fabric, interply, ply number and load density (load support data)

NOTES: 1. An asterisk following a width thus '900*' denotes availability in the manufactured range.
2. All other maximum widths have been extrapolated from data available.
3. 1800 mm is the maximum production width available, and does not necessarily represent a theoretical maximum.

Fabric and interply coding	Number of Plies				
	2	3	4	5	6
8407 Ex	.902	1.119	1.341	1.559	1.797
HD	.983	1.204	1.530	1.788	2.061
8814 Ex	.960	1.248	1.491	1.709	1.964
HD	1.062	1.108	1.704	2.004	2.320
8874 Ex	1.024	1.300	1.628	1.910	2.217
HD	1.108	1.431	1.773	2.091	2.424
8911 Ex	1.064	1.359	1.663	1.980	2.275
HD	1.357	1.444	1.796	2.136	2.477
8961 Ex	1.113	1.436	1.788	2.110	2.448
HD	1.169	1.519	1.899	2.265	2.633
8962 Ex	1.476	2.001	2.513	3.032	3.554
HD	1.532	2.085	2.640	3.190	3.739

TABLE 6.3.3

Factory variable costs (October, 1975) expressed in £/metre for various combinations of fabric, amount of interply, and number of plies. (All figures are based on 1550 mm width production.)

Knowing the three pieces of information shown in Tables 6.3.1, 6.3.2 and 6.3.3, it was possible to find the minimum cost construction fulfilling the requirements of any particular belting application. As an example, suppose that a belt of 730 m total length, 1200 mm wide is required to carry coal of 0.9 tonnes/m^3 density. The maximum working tension is found to be 62.9 kN/m. A belt with tensile strength of at least 630 kN/m (to allow for a 10:1 safety factor) having a maximum permissible width of at least 1200 mm when carrying a load density of 0.9 tonnes/m^3 (to give the required transverse load support) is required. The sequence of events to find the least cost construction is then:-

1) From Table 6.3.1 find the number of plies/fabric combinations of at least 630 kN/m tensile strength. For example, fabric type 8814 with four, five or six plies; fabric type 8874 with three, four, five or six plies; fabric type 8961 with two, three, four, five or six plies; etc.,

2) From Table 6.3.2 find those constructions in the list from Table 6.3.1 which give adequate transverse load support. For example, four plies of fabric type 8814 with Extra (EX) or Heavy Duty (HD) interply rubber; three plies of fabric type 8874 with EX or HD interply; two plies of fabric type 8961 with HD interply; etc.,

3) From Table 6.3.3 find the least cost construction in the list from Table 6.3.2. In this case, we have two plies of fabric type 8961, with HD interply at a Factory

variable cost of £1.169/m.

A sample of applications was analysed in this way to show whether or not the belt construction range quoted in the manual was consistent with the minimum cost range which would emerge from this work. It was thought that there would be a high degree of inconsistency because the Starflex range of belts tended not fully to utilise the minimum ply concept of synthetic belts when the changeover from cotton fabrics occurred. (This was deliberate policy in order to "bulk-out" the belts because conservative customers were under the illusion that fewer plies meant inferior quality.) The analysis would show if this inconsistency was restricted to specific belt types, and, if so, would suggest a basis for rationalisation.

The sample of applications used for this analysis was the same as that used for the work described in Section 6.1. It represented the enquiries passing through the Technical Sales Representative's office during October, 1974, and was used because it was the only readily available set of original conveyor specification data. (A large steel plant scheme was excluded from the results - see Section 6.1.) The results from the analysis of this sample are given below.

<u>Results of the analysis to show the possible scope for Starflex product range rationalisation</u>

The analysis showed that a useful change of belt construction using heavier fabrics could be effected from the existing Starflex belt constructions known as 630/4, 800/5 and 1000/6; four, five and six plies respectively of the fabric having the code number 8814. The main reservation to this result was that of the accuracy of the extrapolated transverse load support data. Belt constructions at the lower tension rating end of the range were not altered because, if a new belt with a lower tension rating but with the same transverse load support as the existing constructions was required, then either a new fabric or a mono-ply belt would be needed. Unchanged belt constructions at the higher tension rating end of the range resulted from a similar situation where an even heavier fabric than those used at present would be required to decrease the number of plies.

Full results of this analysis are given in a report [106], but the main results were:-

1) Total belt length changed to one of the new proposed constructions was 27,587 m, and this represented 53% of the total sample.

2) The factory variable cost saving effected by the revised constructions was approximately 12%. (It should be remembered that the cost figures used were based on the existing manufacturing and materials purchase mix. If the mix were to be altered on the lines suggested by the report, then the costs might also vary in favour of the new constructions as a greater proportion of the fabrics used for them would be required, thus bringing down the purchase price per metre.)

3) The percentage saving in factory variable cost over the whole analysis sample was 7%.

4) The percentage changes in length of belting produced in each ply number were:-

Number of Plies	Percentage Change
2	+ 82
3	+ 183
4	+ 11
5	- 62
6	- 100

More complete results are shown in Figure 6.3.1 which shows the direction of construction changes from the existing belt range to that proposed, for the length of belting "supplied" in each construction.

The coding of belt type used in this figure is:-

The fabric number is followed by the amount of interply (Heavy Duty, HD, or Extra, EX) which is then followed by the number of plies. Thus, a coding 8814 HD 4 represents a belt construction made from four plies of the fabric type numbered 8814 and the amount of interply used is that of the Heavy Duty belts.

The conclusions drawn from the results of this analysis are discussed in the next section.

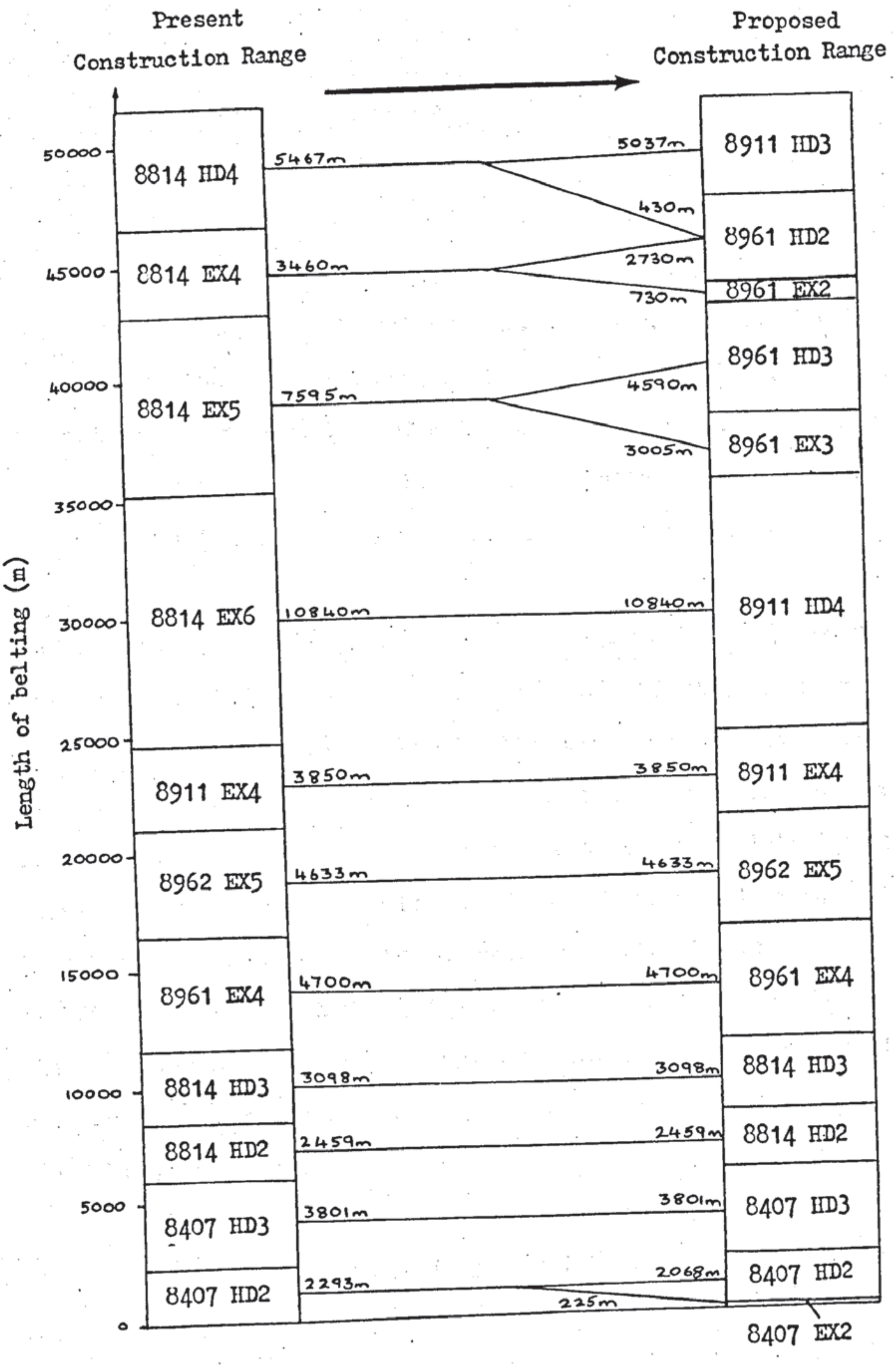


FIGURE 6.3.1

Direction of construction changes from the existing Starflex range to that proposed in Section 6.3

NOTE:- Explanation of construction codes can be found in text.

<u>Conclusions from analysis to show the possible scope for Starflex product range rationalisation</u>

The analysis showed that significant cost savings might be possible from a review of the construction range. However, as pointed out previously, there were several reservations to the validity of the analysis; the main ones were (a) lack of a data base with proven relevance to the present belt market and (b) the extrapolation of load support and other technical data from that of existing belt constructions.

The number of belt construction specifications in the new proposed range turns out to be greater than that in the existing Starflex range. Further work on the same sample of belts with the same "extrapolated" technical data could have been carried out to show the effects of rationalisation of the new range and the possibilities for introducing different fabrics could have been studied. However, this work would have been of limited value whilst there were still doubts about the technical data used. Instead, the major recommendation to come out of the analysis was that the gaps in the technical data should be filled by a long term programme satisfying the needs of both the Development and Sales Departments.

It is suggested that if such a programme is started; it should include the following elements:-

a) Collection of field service data. The common Starflex range should be the focus of activity and work on the transverse load support of these belts would be especially useful.

b) Laboratory evaluation of the design properties of the existing range and comparative data on other possible constructions. (Again, the most important property to study would be transverse load support as no rationally derived data whatsoever was available for the analysis.)

c) Development work to improve the efficiency of splicing and fastening of belts with only two or three plies. (One of the doubts about the introduction of more two-ply constructions related to the possibility of splicing problems.)

d) The correlation of belt technical properties with industry group applications. (This could provide evidence for the validity of the end-use belt selection rules mentioned in Chapter 4.)

e) Customer surveys to find the sensitivity of relevant industry groups to product and price changes. This would show the commercial implications of altering the technical properties of the available belt constructions.

f) The development of a customer education programme to reinforce the quality image in belts with fewer plies. Certain industry segments still view belts with fewer plies as being reduced quality, and such a programme would help to overcome this attitude. (See the "psychological" reasons for overspecification of belt properties in the analysis of Section 6.1.)

g) The monitoring of production cost data to allow a comparison of present and proposed product ranges.

Thus, the analysis described above which set out to find cheaper ways of providing transverse load support led to the conclusion that there was insufficient technical and commercial data to be able to justify changes in the existing belt range. It did, however, show that considerable cost savings might be possible if this data was available. As transverse load support had already been shown to be one of the most important criteria in the selection of belts by the work described in Section 6.1, the present analysis helped to demonstrate the commercial implications of knowing the exact "amount" of load support required by any belting application and the exact "amount" that could be supplied by any particular construction. Analysis data for this had to be extrapolated from that relating to the existing belt range in the Starflex manual, and so, after accepting the published data (which is, itself, based on doubtful premises), assumptions had to be made about the extra transverse load support capability that would be supplied by using heavier fabrics.

Thus the present analysis also showed the need for a service-related test for transverse load support capability when considering any rationalisation of the existing belt range. An interesting point which showed the urgency of this, was that competitors had started, in some cases, to quote three ply constructions in applications for which Dunlop recommended four or five ply belting. This suggested two alternatives:-

a) They were using heavier fabrics or had improved the transverse load support capability of their belts in some other way.

b) They had carried out work which showed that, previously, certain belt design properties were being grossly over-specified.

Either alternative shows the need for research work to optimise the use of design properties in belting.

The significance of the major conclusions from this and the two previous analyses of this chapter is presented below.

6.4 Summary of conclusions from the analyses described in Chapter 6

The three pieces of work described in this chapter have had to use existing published technical data as a basis for investigation and reservations about the validity of certain areas of this data have been expressed. Also, the sample of conveyor installations used in the analyses was fairly limited because of the lack of full data on the present market, but, nevertheless, the results of the analyses have been so conclusive that certain statements are justified:-

(1) Transverse load support is a very important factor in the belt selection procedures. It decides the final belt construction to be used in more than half of all belting applications and meeting the requirements of this parameter can cause gross overspecification of belt tensile strength.

(2) Meeting longitudinal load support requirements often decides the working belt tension for a conveyor, but seldom affects the eventual outcome of the belt selection procedure.

(3) The lack of valid service-related technical data for existing belt constructions prohibits the investigation of the possibilities for altering and rationalising the product range. This is particularly true of transverse load support data as no suitable test whatsoever is available for the assessment of this belt property.

This chapter has therefore proved beyond doubt that, as suspected, the most important research area to consider is that of transverse load support. It has shown the commercial implications of providing the correct amount of transverse load support without excessive over-specification of other design criteria, and so has justified an experimental programme to study this property. Such a programme should lead to a method of assessing and describing the capability of any belt construction to provide transverse load support.

The work in this chapter has also shown the need for a record of details of belt applications and the belt constructions actually used on them. This is required not only to correlate laboratory testing of belts with service conditions as mentioned in Chapter 4,but also to provide a data base for the type of analyses described here.

It has been pointed out in earlier sections of this thesis that the introduction of deep-trough belt conveyors and strong synthetic carcase fabrics have led to problems with transverse load support; that very little published scientific literature relating to this property is available; that belting manufacturers, whilst agreeing on most aspects of the belt selection procedure, differ considerably when describing this property; and that Dunlop's transverse load support data is based purely on experience. The analyses described in this chapter have quantified the importance of transverse load support and have thus provided further support to the view that this property is the area of belt design most urgently requiring research. An experimental programme to demonstrate the factors affecting the provision of this vital design criterion has therefore been justified on both technical and commercial grounds. The following chapters describe the work carried out to set up such a programme.

CHAPTER 7

EXPERIMENTAL INVESTIGATION OF TRANSVERSE LOAD SUPPORT

Earlier chapters have suggested that transverse load support represents the major problem area of belt design and selection. The work described in Chapter 6 proved this and justified an investigation into the parameter on both commercial and technical grounds. Ideally, such an investigation should be carried out using field trials of belts running under controlled conditions. However, because of the numerous possible combinations of belt and conveyor parameters that might affect load support, this would be an extremely expensive and time consuming project. It was therefore decided that, initially, an experimental exercise should be carried out on a test rig on which it was possible to vary those features which might affect load support. This compromise situation had been adopted previously by several workers; for example, Behrens [28] constructed a conveyor rig on which he carried out work on some basic design parameters of belt conveyors and Saucier [107] built a special rig to investigate the use of belt conveyors to transport mass concrete. Of course, any results found in such a way should be checked with service behaviour if the opportunity arises.

It was proposed that, as the major problem of load support was that of the belt catching in the idler gap, the transverse belt deformation in this area should be measured for various belt constructions, and its variation with idler angle, idler pitch, load density and load weight should be studied. These parameters are those which belt manufacturers consider to affect load support (see Section 3.4). In addition to this, the variation with belt tension, a parameter easily controlled on the available test rig, was to be studied. It was hoped that this work would lead to a criterion by which load support could be assessed.

7.1 Test Rig Details

If a test rig had been built specifically for this project,it would have been very much like that of Behrens [28] ; that is, it would have comprised of a long belt sample under tension supported by adjustable angle idler sets which could be moved easily to any required position. Static measurements of the belt deformation at the idler gap could then be obtained with displacement transducers using the technique described below in Section 7.2. Behaviour under dynamic conditions could be studied by mounting several idler sets on a moveable platform and moving this at various speeds under the loaded belt. The measurement of belt deformation under this condition would be more difficult but would not pose an insurmountable problem. Devices used by Oehmen and Alles [59] could be adapted: these consisted of a small roller pressed against the belt by a light spring; when the belt deformation changed, the roller followed it. The movement of the roller could be detected by a standard displacement transducer. If such devices were to be used for measurement of belt deformation at the idler gap,the linkage between the roller and the displacement transducer would have to be designed so as not to interfere with the rotating idler.

Building an ideal rig was not feasible for the proposed project as the construction time would be greater than one year and the cost would be unacceptably high. Instead, the best use had to be made of an available test rig. This rig, built approximately ten years prior to the present work, was far from ideal for the proposed investigation. It consisted of two inclined conveyors, approximately twelve metres long, positioned side by side. These were designed to form a "closed circuit"; that is, one conveyor unloaded material onto the tail end of the other which transported the load to its head end and then discharged onto the tail end of the first conveyor. (This was therefore a similar machine to that described in the article "Measuring conveyor belt wear" [108]. The purpose of the closed circuit arrangement was that belt wear and breakdown could be studied without the need for continually replacing the load. The time taken for the load to complete one cycle was approximately five minutes and it was found that the material had broken down so much after fifteen to twenty cycles that it no longer abraded the belt. The load had therefore to be completely replaced frequently - an operation taking a very long time. In addition to this problem, the conveyor was fitted with very large diameter end pulleys designed for use with high strength cotton belts, which did not impose damaging strain levels on all-synthetic carcases. The conveyor rig had therefore not been used for any major investigations for several years. This conveyor rig, when installed, cost approximately £60,000 to build, house and instrument. Allowing for inflation, it can be seen that to build an ideal rig for the proposed work would probably require investment in excess of £180,000 and so the best use of the redundant test rig had to be made. A small overall photograph of the rig is shown in Figure 7.1.1 and other photographs taken during the experimental work are given later.

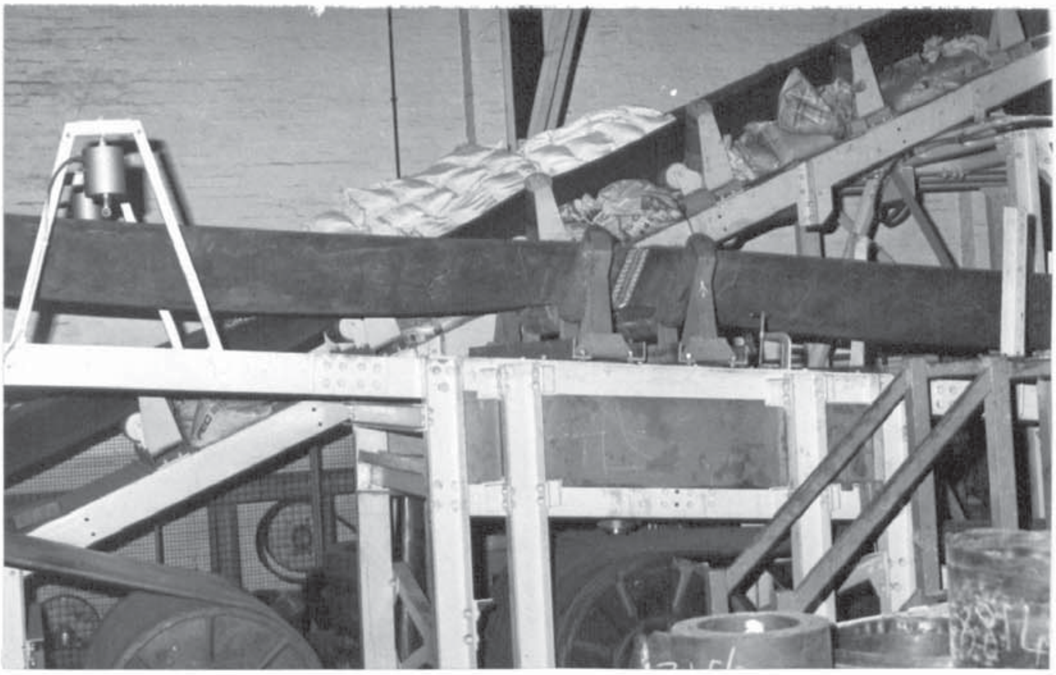


FIGURE 7.1.1

General view of test rig used in the experimental investigations described in Chapters 7 and 8

The conveyor equipment available for use on this test rig consisted of:-

(a) A belt tension measurement device. This was in the form of a gravity take up which could be forced downwards by hydraulic rams. Load cells were connected to the rams and their read-out was calibrated to show belt tension in pounds-force. Some notes on the accuracy of this device are included in Sections 7.6 and 7.7.

(b) Three idler sets which could be adjusted to either 35° or 45° troughing angle. These idler sets were of standard three rollers form; the central roller length could be varied so as to be compatible with different belt widths. Idler sets of 20° troughing angle were present at each end of the conveyor to provide a smooth transition between flat and troughed belt.

(c) A 1000 mm wide "vehicle" belt. As only static tests were to be carried out for the proposed work,there was no need to install complete belts of the types to be tested. Instead, it was only necessary to incorporate a sample into this vehicle belt. It was, of course, essential to ensure that the sample was long enough to prevent the thick and heavy vehicle belt affecting the belt shape measurements in the test area.

When the importance of studying transverse load support was demonstrated by the work described in previous chapters, a budget of £1000 was allocated to the project to allow this existing test rig to be adapted for the proposed work. The following sections in this chapter describe how this was done.

7.2 Design of a special Idler Set

It was decided that a special idler set should be designed so that the transverse shape of the belt at an idler station could be measured under static conditions. It was proposed that this could be done simply and cheaply by drilling holes in the idler through which a dial gauge or displacement transducer could be used to measure the distance of the belt from the idler surface. This is demonstrated in Figure 7.2.1. The "depth" δ would be measured and then the distance between belt and idler could be found by simply subtracting the known thickness, Δ, of the idler. Unfortunately, the diameter of the idler rollers on the test rig was fairly large (152 mm) and the height of the idler bearing supports was approximately 100 mm; thus, only very limited access of 24 mm (100 - 152/2) was available to the underside of the roller. As only static measurements were to be taken it was possible to overcome this problem by making the cross-section of the special idler set that of a circle segment. (See Figure 7.2.2.) In order to simplify the manufacture of the idler set and thus keep the cost to a minimum, the "idler segment" was made from wood and a metal base plate was attached to provide strength and a reference plane for the measurements. Adequate access was provided thus to the underneath of the idler for a dial gauge. (It was decided that a dial gauge should be used rather than electrical displacement transducers

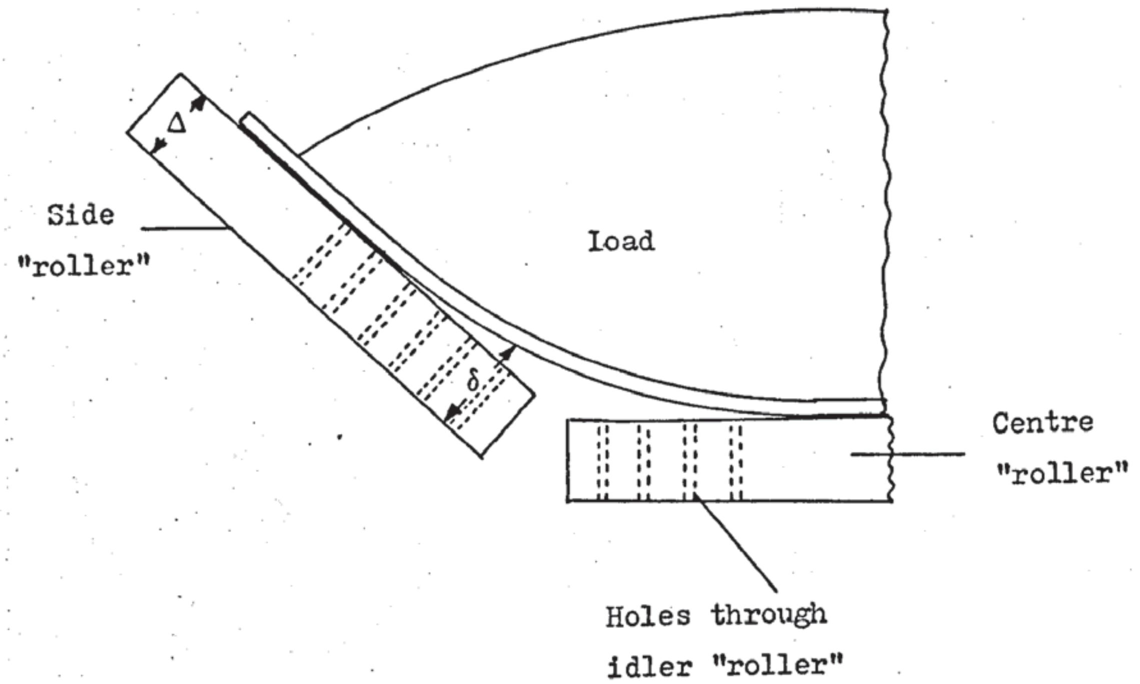


FIGURE 7.2.1

Belt shape measurement at idler gap

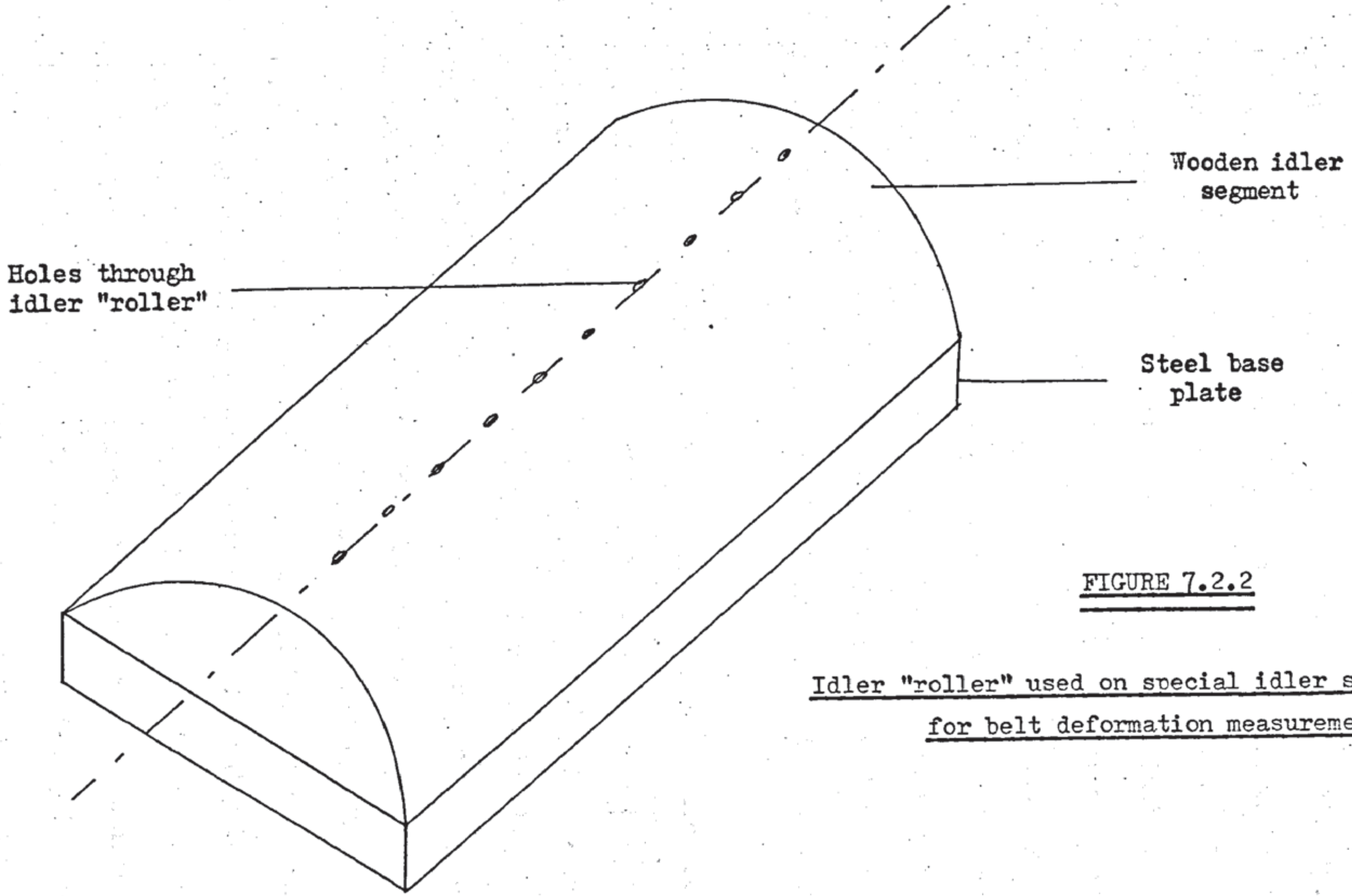


FIGURE 7.2.2

Idler "roller" used on special idler set designed for belt deformation measurement in order to keep expense to a minimum.) In all, six of these special idler "rollers" were made; two were designed as side "rollers" and were 530 mm long, the other four were centre "rollers" and were 530 mm, 465 mm, 380 mm and 315 mm long to form idler sets consistent with belt widths of 1400 mm, 1200 mm, 1000 mm and 800 mm respectively. Most of the test work was carried out using 1000 mm wide belts but the other centre idlers were produced so that different widths could be tested at a later date. In practice, on actual conveyor installations the side roller length would always be equal to that of the centre roller, but, for the purposes of test work this was not necessary, provided that the side roller was at least as long as that at the centre. An allowance for the longer than usual side roller was required if idler support loadings were measured for belts narrower than the 1400 mm width for which the 530 mm side rollers were designed. (See below.)

The next design criterion to be considered for the idler set was the provision of adequate supports. The work of Grimmer [68] was used to find the maximum loads that would occur on each support. Grimmer carried out experimental work to establish the validity of a theoretical formula for the load occurring on each support and, on the basis of his results, modified the formula by empirically derived factors. Using this formula showed that the maximum load weight that would act on any one support was in the order of 250 kg. This was under conditions of the maximum possible belt width being fully loaded with gravel of density 1.6 tonnes/m^3 and an idler pitch of the maximum acceptable for such conditions. This figure agreed fairly well with that found from a very simple technique suggested by Brockway Engineering Limited, conveyor equipment manufacturers. This company used the rule that each idler support took the percentage of total weight as shown in Figure 7.2.3 and so each centre roller support took thirty percent of the total weight of load and belt. It was found that supports made from steel channel compatible with the dimensions of the idler "rollers" and the test rig would be more than adequate to support this load level without significant bending. It had been decided that the forces at the points A, B, C and D in Figure 7.2.3 would be measured as part of the test work and so the need for a suitable combination of load cell and support design compatible with the limited space available became an important consideration. (An idler gap of 10 mm was to be used during the experiments and so very little space for support was available if access to the holes through the idlers was to be provided.) Initial studies showed that insufficient space was available for inclusion of load cells consisting simply of a small strain gauged cylinder of aluminium or steel designed to deform to 1000 microstrain at the anticipated load levels. It was therefore necessary to consider commercially available load cells, and it was decided that "load washers" having dimensions typical of normal washers should be used. A full description of these washers and the details of their calibration is given in Appendix 9. The load washers were available with a rated capacity of 225 kgf or 500 kgf and it was claimed by the manufacturers that the output signal was unaffected by shear forces occurring across the surface of the washer. It was decided that the lower rated alternative should be used even though the maximum loading might possibly exceed its capacity. This was done because most of the loads would be much less than the maximum possible and so it was likely that a poor output signal would be obtained from the 500 kgf washers under most conditions. The safe overload of these washers was stated as being 150% of the rated capacity and so it was unlikely that the maximum possible loads would damage them.

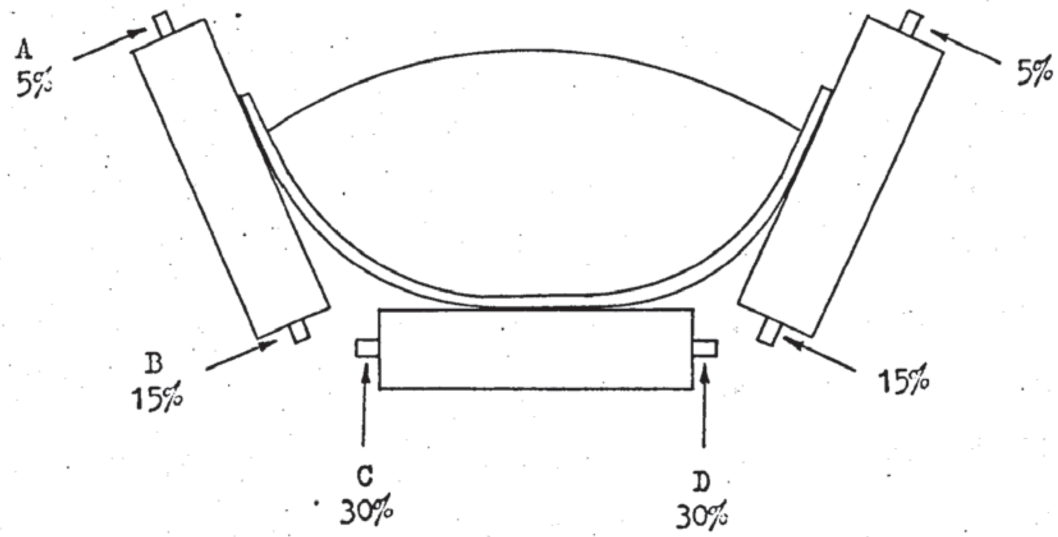


FIGURE 7.2.3

Estimated percentage of load weight taken at each idler support

However, the output of these load cells was carefully monitored during the experiments to ensure that the rated capacity was not exceeded. For economy, only four load cells were used as, by symmetry, the forces on each side "roller" should be the same. Washers having the same dimensions as the load washers were included on the supports at which force measurements were not carried out to ensure that the apparatus was symmetrical.

The use of load washers meant that the idler "rollers" could be supported simply by a metal stub as shown in Figure 7.2.4. Two sets of side "roller" supports were made - one for 35° idler angle, the other for 45°. In order to provide access to as many of the holes through the idlers near the idler gap as possible, the centre "roller" supports were moved inwards and so the final idler set was as shown in Figure 7.2.5. As mentioned above, the use of 530 mm side "rollers" for belts less than 1400 mm wide meant that an allowance to the output signals from the load cells on the side roller would have to be made in the analysis of results. The relationship between the measured forces and those that would occur if shorter side "rollers" were used is derived in Appendix 10.

All the wooden idler surfaces were treated with a teflon spray to minimise frictional forces between belt and idler. On a normal conveyor the rotation of the idlers would mean that the frictional forces would be reduced to a minimum, and as only static experiments were being carried out in the present work it was thought that a surface coating was required to reduce friction as much as possible.

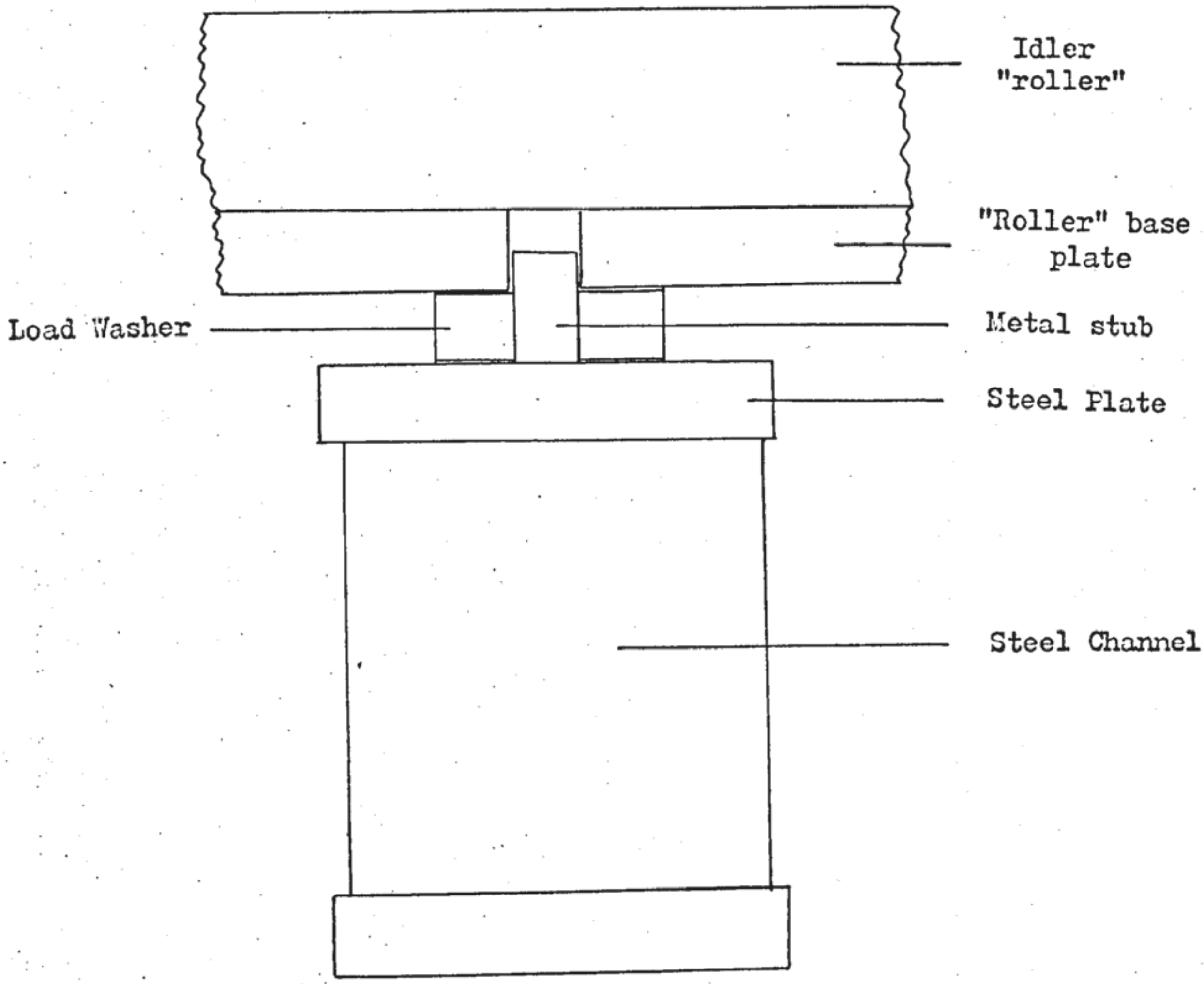


FIGURE 7.2.4

Method of support for idler "rollers" as used during experimental investigation

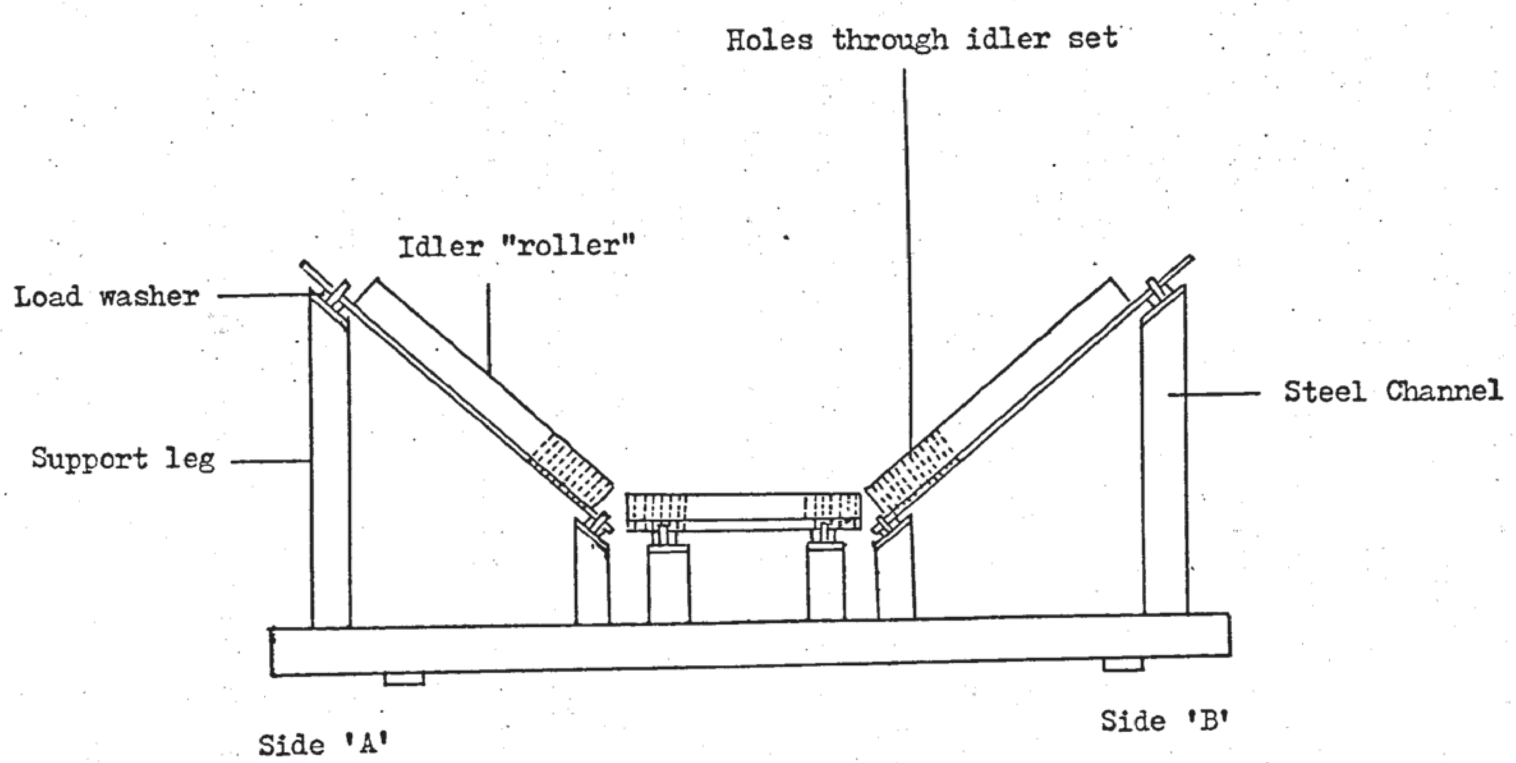


FIGURE 7.2.5

Idler set designed for study of belt transverse load support

Although most interest centred on the belt shape at the idler station, it was also measured at a section mid-way between two adjacent idler sets. This then provided a comparison between the two different positions and showed the cyclic change in deformation caused in the belt as it travels over several idler sets. A wooden structure, shown in Figure 7.2.6 by which the belt shape could be measured was therefore designed to be positioned between the idlers. The most critical dimension of this structure was its height as it had to be such that it did not touch the belt even when the belt was fully loaded. The method of measurement of belt shape used was the same as that at the idler set; that is, holes were drilled through the wooden structure at regular intervals and the distance of the belt from the structure could then be measured using a dial gauge. As it was a fairly lengthy process to set up the proposed investigation, it was obviously necessary to ensure that as much useful data as possible was collected during each experiment. The inter-idler sag was therefore also recorded from the measurements taken from the wooden structure. The work described in Chapter 6 had demonstrated that this design criterion was not as important as transverse load support, but the number of belt selections that it affected justified some work being carried out.

The equipment described above could therefore be used to:-

(a) Measure the belt shape at the idler station.

(b) Measure the forces occurring at the idler supports.

(c) Measure the belt shape mid-way between idler stations.

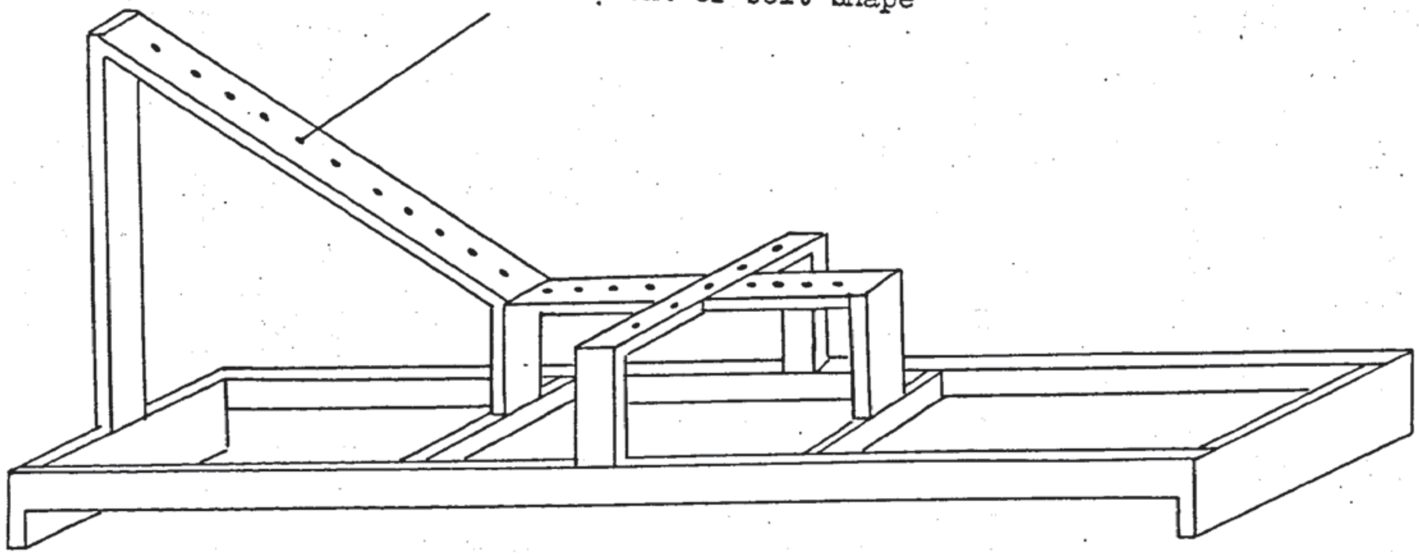


FIGURE 7.2.6

Wooden structure designed to fit on test rig for measurement of belt shape mid-way between idler sets

The manufacture of the special idler set meant that, in total, four idler sets were available for use. It was thus possible to form three "inter-idler" sections - the absolute minimum number required if experiments over one idler pitch were to be at all meaningful.

Ideally, far more idler sets should have been used, but these were not available and the short length of the conveyor prevented the use of many more anyway.

The cost of the equipment described above was as follows:-

Four load washers	-	£302.00
Idler set and accessories (manufactured by Dunlop Machine Shop)	-	£303.00
Dial gauge and extension rods	-	£ 12.00
		£617.00

This, therefore, left approximately £400 for any additional expense that was incurred during the investigation. Most of this remaining money was required for equipment to develop a suitable strain transducer for use in the experiments, the need for which is explained in the next section.

7.3 Development of a Strain Measurement Device suitable for belting

When the experimental investigation was started it was not clear which parameter would eventually be used as the criterion by which transverse load support would be assessed. The equipment described above allowed several measurements to be taken that might be used for this purpose. The most likely parameters were:-

(1) Length of non-contact between belt and idler.

(2) Distance of belt above idler gap.

(3) Radius of curvature to which belt is deformed. (Calculated from belt shape measurement.)

It was thought that belt strain levels near the idler gap might be damaging if insufficient load support was available, and so another possible criterion to add to the list given was that of belt strain. As strains in belts are relatively large compared with those that occur in metals, it could, at first, be thought that it would be possible to draw a grid on the belt and simply find strain levels by straightforward measurement of changes in length of the grid lines. It was likely, however, that this would give insufficient accuracy and work carried out described below in Section 7.6 demonstrated that this was the case, and so another method of strain measurement had to be devised. A development programme was therefore started and this was carried out in parallel with initial experiments on the main test rig. These experiments demonstrated that strain was not the criterion by which load support should be assessed for <u>modern</u> belt constructions. Nevertheless, the development of a strain transducer was continued as such a device would be useful for many other investigations into the mechanism of belt performance and failure.

The investigation is described fully in Appendix 11 and resulted in "clip gauges" being adapted into a form suitable for use on conveyor belts. The total cost of this development, met from the budget allowance of the main work, was approximately £220. This figure includes the cost of several special tools required for the bonding of electrical resistance strain gauges to the materials being studied.

7.4 Preparation of Load Materials

It was proposed to study the effect of load weight and load density on load support and so it was necessary to have at least two different types of load in a form suitable for the experiments. It was not feasible to use any load in a loose form, because continual loading and unloading of the test rig was to be carried out; if loose materials had been used it would have been very difficult to ensure a uniform distribution of load and to keep a record of the actual load weight on the rig. Therefore, it was decided that any load must be contained in bags without affecting excessively the physical behaviour of the load material. It was proposed that two loads should be used and gravel and coal were suitable materials readily available at the research site. (The coal used was approximately half the density of the gravel.)

Ten kilogrammes was considered to represent a suitable maximum load weight to be used in one bag. Suitable dimensions for the bags to be used for this weight of gravel were decided as approximately 1000 mm x 200 mm x 25 mm; the length of each bag was chosen as one metre because this simplified the calculation of load per unit length of conveyor and the other dimensions ensured that movement of load material could occur within the bag. A number of bags were therefore made up from a light, flexible fabric which had been intended for use as belt carcase reinforcement but had been found to be unsuitable and had therefore been scrapped. This fabric was used because no charge for it was made to the project allowance. The bags were filled with either ten kilogrammes of gravel or five kilogrammes of coal, and divided into four equal sections to ensure that an even load distribution was obtained when the bags were placed on the conveyor rig. The final result of this work was that approximately two hundred bags of gravel and two hundred bags of coal were produced and these were easily handled and provided a simple method of ensuring an even load distribution. They can be seen in some of the photographs taken during the experiments. (Figures 8.1.1, 8.1.2 and 8.1.3.)

7.5 Collection and Choice of Belt Samples

Belt samples of constructions in the Dunlop Starflex range were to be studied in the experiments. As a belt length of approximately eight metres was required for the work, all the samples had to be obtained from the factory shopfloor as no other facilities were available to make such a large piece of belt. If the belt samples had been specially made, they would have been extremely expensive. Accordingly, all samples were obtained from normal production batches; this implied a certain compromise in the selection of belt types used in the tests.

In Section 6.1 it was shown that a combination of the load support properties of the belt type "630/4 Heavy Duty" and the tensile strength of the construction "315/2 Heavy Duty" would probably provide an optimum belt. It was therefore decided that an attempt should be made to obtain belt samples of these construction types for the test work. These two constructions have the same reinforcing fabric; two plies for the type "315/2" and four plies for that designated "630/4". It was also considered worthwhile to demonstrate the contribution to load support from the amount of rubber interply. The additional amount of interply present in the "Heavy Duty" range of belts over that of the "Extra Duty" belts is claimed to improve load support dramatically. (See Table A5.11, Appendix 5.) An attempt was, therefore, also made to obtain a sample of the construction type "630/4 Heavy Duty" from shop-floor production.

As the vehicle belt on the test rig was 1000 mm wide, it was decided that the majority of the test work should be carried out at this belt width. Thus, three samples were required as follows:-

(1) 9 m of 1000 mm wide "315/2 Heavy Duty" construction.

(2) 9 m of 1000 mm wide "630/4 Heavy Duty" construction.

(3) 9 m of 1000 mm wide "630/4 Extra" construction.

It can be seen from Table A5.11 of Appendix 5 that these belt types have, between them, a fairly wide range of "maximum permissible belt widths for satisfactory load support", thus providing a useful range of claimed load support capabilities to be tested.

Samples of the three belt types were obtained from the production department but, unfortunately, there were some variations in the cover thicknesses. (The actual details of the belts tested are given in Section 8.1.) As the samples were obtained from normal production, no charge was made against the project budget but, of course, there was an effective "opportunity cost" of the belts equal to their selling price. Details of this also are included in Section 8.1.

It so happened that, before any of the above belt samples could be placed on the test rig, an opportunity arose to study a load support problem that occurred in service. As this work was directly relevant to the project and would provide an opportunity to gain experience of working on the test rig, it was decided that some time should be devoted to experiments concerning the problem and these are described in the next section.

7.6 Test Rig work concerned with Service problem of belt at a Steelworks

The service performance complaint that led to the test work described here resulted from the behaviour of a Dunlop Starflex belt type "800/4 Extra" installed on a conveyor carrying iron ore at a steel plant. The belt was said to be unsatisfactory on the grounds of "generally inadequate load support" at the concave curve where the belt left the main run of the conveyor and moved into a travelling tripper*. The main cause of the complaint was the occurrence of:-

(a) Ripples in the belt edges.

* See glossary of terms at the beginning of this thesis.

(b) "Tubing" of the belt. (This term was used to describe lifting of the belt edges clear of the side rollers of the idler sets near the tripper, thus causing the belt cross-section to become more tube-like.)

(c) Excessive belt sag which could cause trapping of the belt underneath the first idler of the tripper if the tripper moved backwards with the belt stationary.

These three effects did not necessarily occur simultaneously but each of them was apparent at certain combinations of loading conditions and tripper positions.

It was argued by the customer that a thicker, heavier belt such as the Starflex construction type "1000/5 Extra" would not have demonstrated these "faults". (Faults is in inverted commas because it is arguable whether or not any of the observed effects were detrimental to the performance of the conveyor.)

Dunlop Belting Division claimed that the construction 800/4 Extra was perfectly adequate for the task and that the phenomena listed above were due purely to the layout of the conveyor framework and the amount of tension induced in the belt. It was thought that under the same conditions any other rubber/fabric conveyor belt would exhibit these effects although, perhaps, to a slightly different degree. A test was therefore set up on the available test rig to try and reproduce the service conditions and then demonstrate that the belt was not at fault.

This work was not the next most logical step in the research project on transverse load support but, as the work was required anyway, as much use as possible was made of the situation to gain experience of the test rig and to collect any useful data.

In order to make the test conditions as near as possible to those occurring in service, four idler sets of the type installed on the actual working conveyor were mounted on the test rig. (These idlers were suitable for a belt width of 1200 mm and had a roller diameter of 100 mm and an idler angle of $37\frac{1}{2}^{\circ}$.) It was found that an unsupported belt length of three metres could occur on the working conveyor at the point where the belt moved onto the travelling tripper. This fairly long effective idler pitch was due to the belt lifting clear of some idlers in the vicinity of the concave curve. It was decided that this should be the idler pitch used for the test work. The behaviour of two different belt constructions was to be compared on the test rig and so it was necessary to include two idler pitches of three metres. The only possible method of obtaining this situation was to position the four idler sets as shown in Figure 7.6.1. (It was, unfortunately, not possible to install the special idler set described above because of the different roller diameter and troughing angle.) It would probably have been better to test the two belt constructions individually and so ensure that the behaviour of one was not being affected by the other; but as the test was also to serve as a demonstration to the customer, and changing a belt sample would require at least one full day, it was decided that the two different belt constructions should be installed on the rig together.

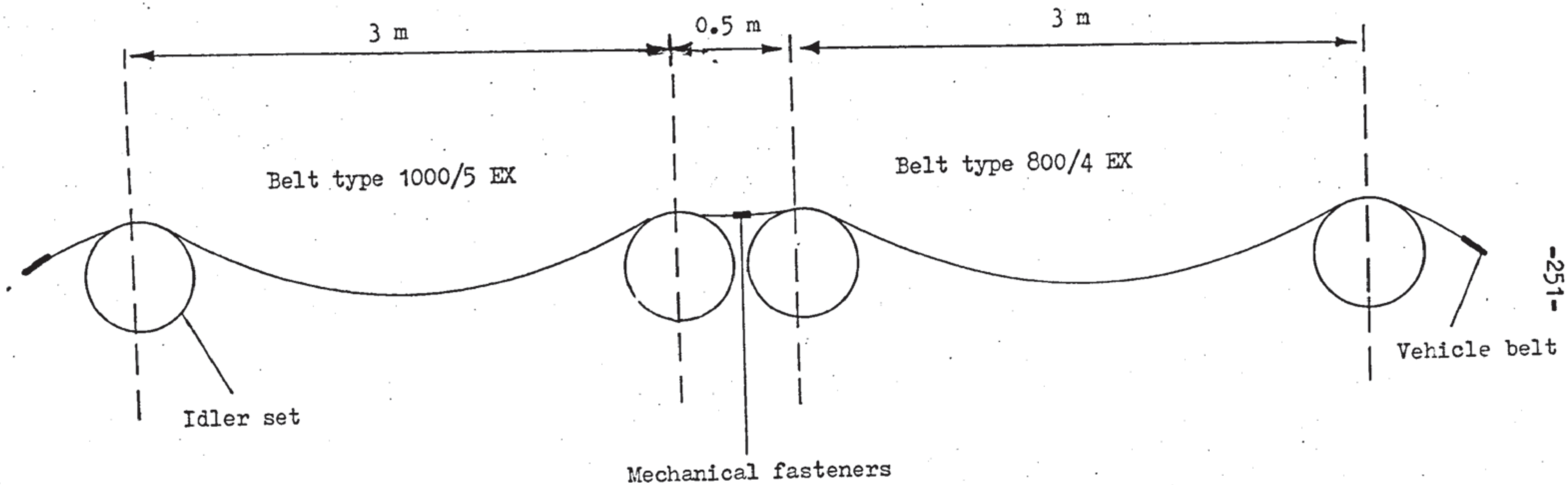


FIGURE 7.6.1

Idler arrangement for test work concerned with steelworks service complaint

(Not to scale)

Some iron ore of approximately 2.2 tonnes/m^3 density was also obtained for the test work as this was the material being carried on the conveyor in service.

It was decided that the measurements taken during the test work should be:-

(1) Inter-idler belt sag at various tensions.

(2) Distance between opposite belt edges at several points along the belt and at various tensions. (This parameter, known as "inter-edge distance" in the present thesis, would demonstrate the magnitude of any ripples that might occur in the belt edges.)

(3) An assessment of the strain distribution occurring in the belt samples.

It was anticipated that no significant differences would be exhibited in these parameters measured on two different rubber/fabric belt constructions.

The strain distribution was to be investigated by direct measurement on a grid formed by positioning staples in the belt at various points as shown in Figure 7.6.2. This grid was designed to demonstrate the longitudinal and transverse strains occurring at various points along the idler pitch and grid gauge lengths of approximately 250, 500 and 1000 mm were used. Before the belt samples were installed on the test rig, they were laid flat and accurate measurements of the lengths between the staples were taken. These were coded as shown in Figure 7.6.2. The two 1200 mm wide samples were then mounted in the vehicle belt and experiments were started. Details of the two belt samples are shown in Table 7.6.1.

Three load conditions were used during the tests:-

(a) Unloaded.

(b) Loaded to approximately 120 kg/m.

(c) Loaded to approximately 240 kg/m. (This was the load weight corresponding to that present on the actual conveyor.)

At each load condition, measurements were taken at various belt tensions. The lowest tension that could be measured was approximately equivalent to 17.8 kN and the maximum possible tension was much greater than the rated operating tension of either of the belt constructions on the rig. It has already been mentioned that the tension measuring device was calibrated to provide a read-out directly in pounds force. The scale of the instrument was such that readings only to the nearest 250 lbf (1.1 kN) could be taken. In addition to this problem, the tension was not actually measured at the position of the belt samples but at a point some distance away. In between the test area and the tension measuring equipment there were the end pulleys of the conveyor system and it was probable that, even under static conditions, frictional forces at these would introduce a difference in the tension between the loaded and unloaded strands of the conveyor. The terminal frict-

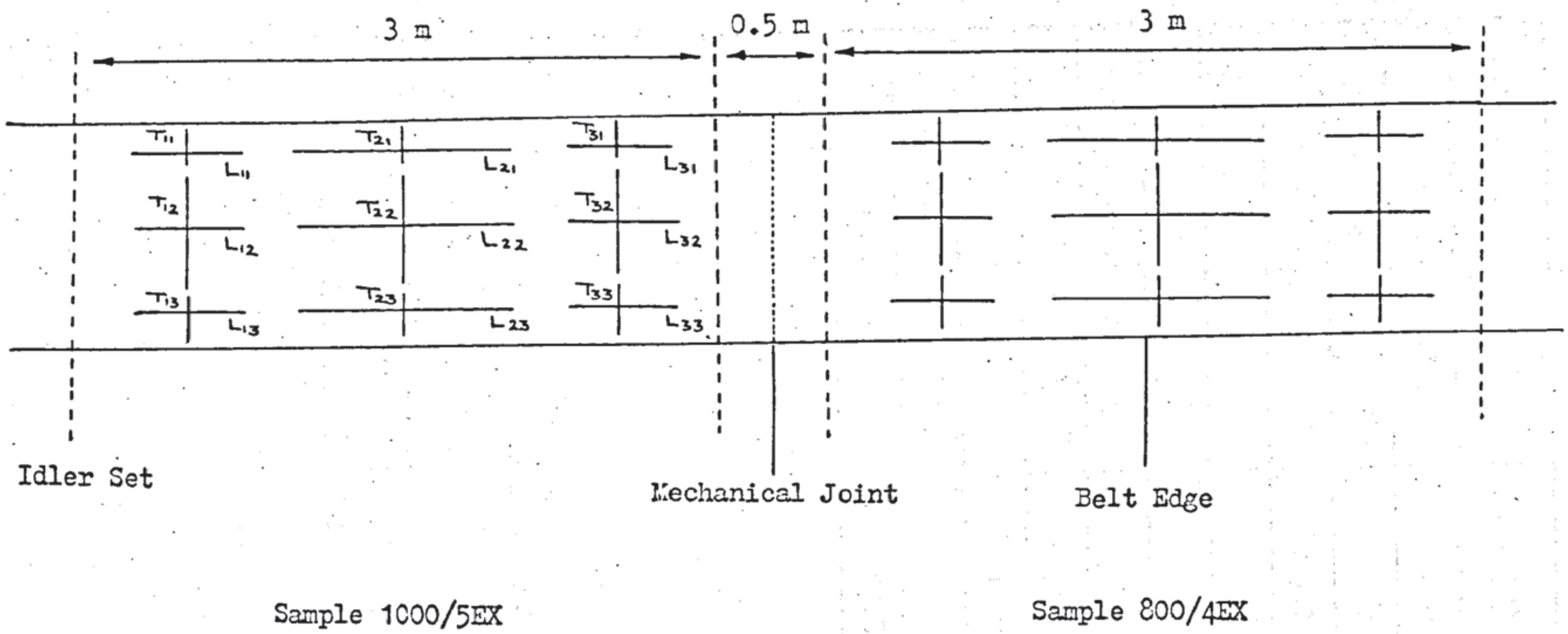


FIGURE 7.6.2

Plan of strain measurement grid layout for test work concerned with steelworks service problem

Grid Lengths

Transverse			Longitudinal		
T_{11}, T_{12}, T_{13}	=	250 mm	L_{11}, L_{12}, L_{13}	=	500 mm
T_{21}, T_{22}, T_{23}	=	500 mm	L_{21}, L_{22}, L_{23}	=	1000 mm
T_{31}, T_{32}, T_{33}	=	250 mm	L_{31}, L_{32}, L_{33}	=	500 mm

Belt Type	800/4 Extra Duty	1000/5 Extra Duty
Tensile Strength (kN/m)	800	1000
Number of Plies	4	5
Type of reinforcement	All Nylon	All Nylon
Fabric Code	8874, NN 202	8874, NN 202
Top Cover Thickness (mm)	3.0	5.0
Bottom Cover Thickness (mm)	1.5	1.5
Carcase Thickness (mm)	5.5	8.5
Overall Belt Thickness (mm)	10.0	15.0
Belt Sample Width (mm)	1200	1200
Belt Weight (kg/m)	12.8	19.2

TABLE 7.6.1

Details of belt samples used in test work associated with steelworks service problem ion factor (see Section 3.3) was calculated for the test rig and found to be in the order of one kilonewton; this factor is considered to represent the frictional force acting at the end pulleys on a working conveyor. It can be seen that for the test rig this was in the same order as the errors introduced by the accuracy with which the tension could be read from the machine scale and so it was decided that no allowance should be made to the tension readings for frictional effects. In fact, it was found that, over several tension cycles, the belt measurements were reproducible to within one or two percent for a given tension value. This suggested that the errors caused by frictional forces were less than those calculated from the "terminal friction factor". This would, of course, be expected because the test rig was kept in a virtually dust and dirt-free environment and in generally better condition than most conveyors in service.

As well as there being reproducible results over different tension cycles, it was found that no significant differences occurred in the measurements taken if the belt was allowed to remain under the same test conditions for long periods of time. This suggested that the major changes in belt deformation when a change was made in the test conditions took place within the time taken to start measurement,and any time dependent changes were minor compared to these.

The actual results recorded from the experiments on the two construction types (800/4 Extra and 1000/5 Extra) can be summarised as follows:-

(1) Inter-idler sag measurements

These measurements were taken at the mid-point of the idler pitch. It is possible that the small conveyor gradient could have displaced the point of maximum sag slightly. (See Section 7.7.) It was found that, under all load conditions examined, there was very little difference between the measured sag values of the two belt types at any given tension. An example is shown in Figure 7.6.3. The construction type 1000/5 Extra exhibited slightly greater sag and this could have been due to the slightly greater mass of this belt. The experiments demonstrated that sag was controlled by belt tension but, in this extreme loading condition, did not exaclty obey the formula used in the belt selection procedure (see equation 2.2.10). This behaviour pattern was in agreement with that observed by Behrens [28] (see Figure 2.2.5).

(2) Inter-edge measurements

These measurements, also, were taken at the mid-point of the idler pitch. It was found that the result was very dependent on belt tension; an example is shown in Figure 7.6.4. At very low tensions a "fold" occurred due to lack of tension at the belt edge. (See Figure 7.6.5.) This caused the measured parameter to be fairly high and gave a situation that could lead to spillage of the load from the sides of the belt. As the tension was increased this fold disappeared, as shown in Figure 7.6.6,

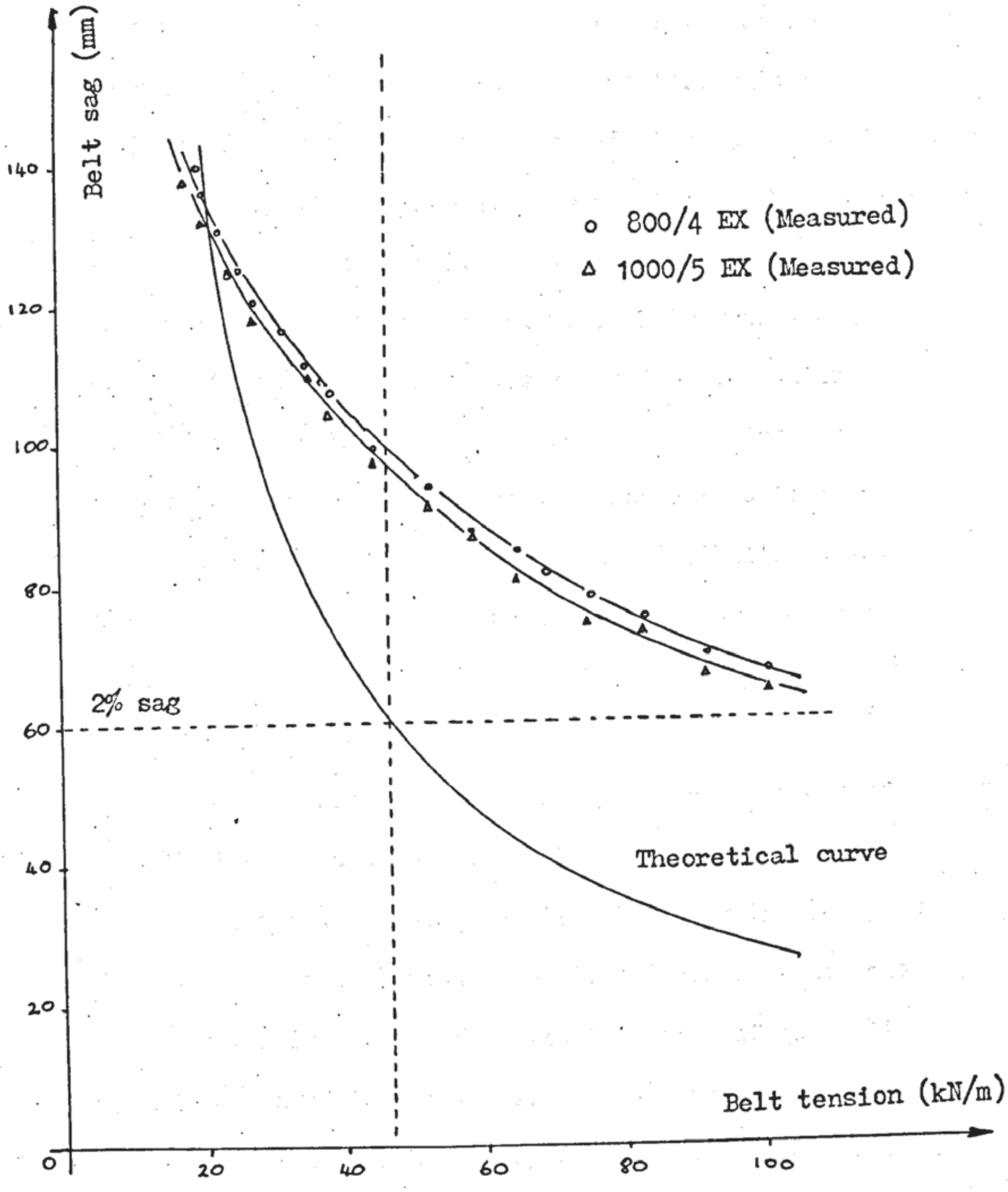


FIGURE 7.6.3

Comparison of measured and calculated belt sag values for two types of belt construction

Note:- 2% sag is the maximum acceptable sag according to most major belt manufacturers.

Test conditions:-

Load weight = 240 kg/m

Idler pitch = 3 m

and the distance between opposite belt edges decreased to a minimum. As the tension was increased further this parameter started to increase again. This was probably due to the belt tension reducing the sag and physically lifting the load, thus pushing the belt edges outwards. Decreasing belt sag would also reduce the belt wrap around the idler rollers. This wrap would cause anticlastic curvature (see Section 8.2) which would force the belt edges inwards. Increasing the belt tension and the subsequent decrease in belt wrap around the idler would decrease the anticlastic curvature and therefore tend to make the belt edges move outwards.

It can be seen from Figure 7.6.4 that the behaviour of both belt samples was very similar; the construction 1000/5 Extra always showed a slightly lower inter-edge distance than the construction 800/4 Extra but this was almost entirely due to the extra thickness of the belt type 1000/5 used in the tests. (As the measurements were always taken from the top of the belt, any increased belt thickness would reduce the inter-edge distance.) If an allowance was made for this factor, the two belt samples still exhibited slightly different behaviour, but not to any significant extent.

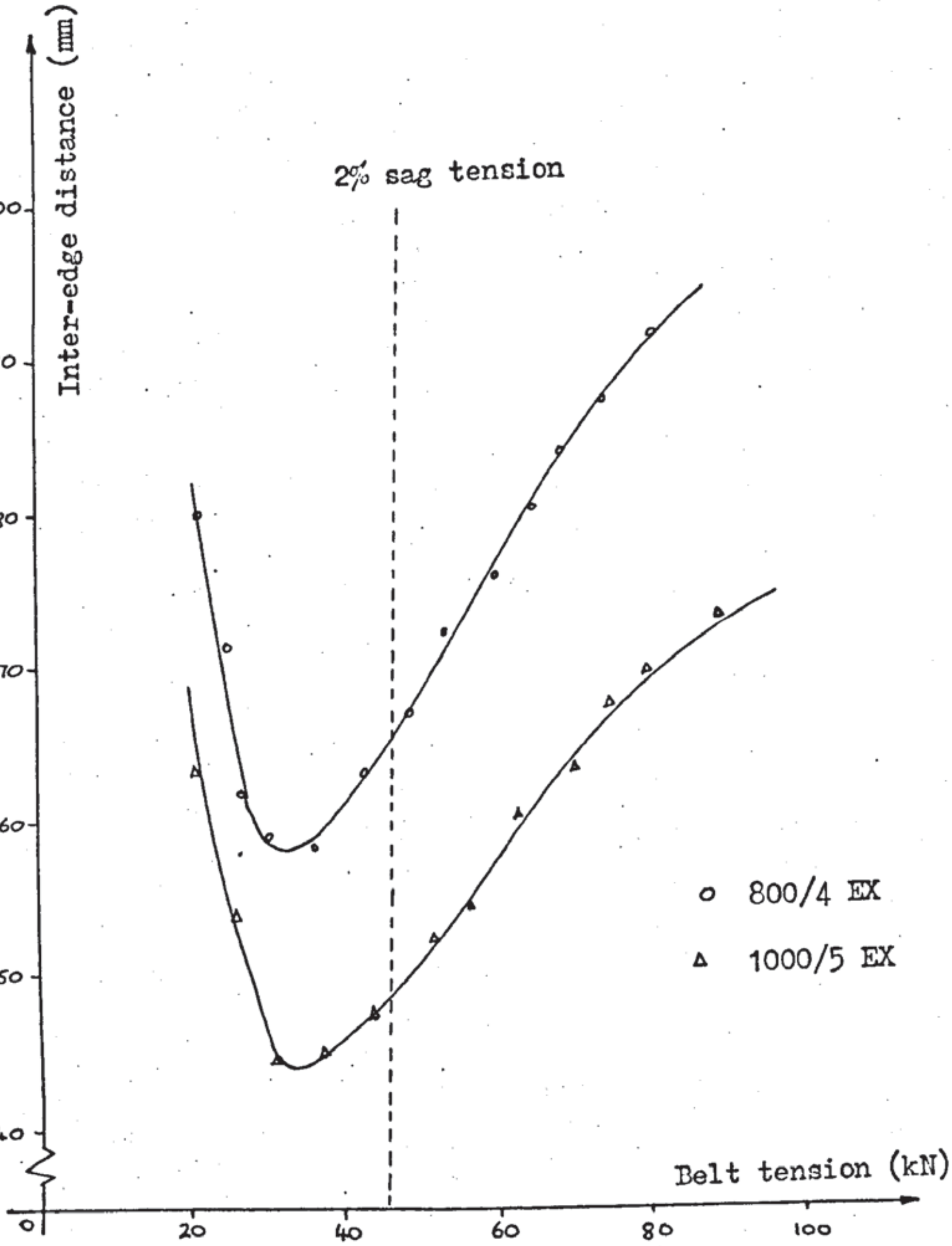


FIGURE 7.6.4

Comparison of measured distance between opposite belt edges at mid-point of idler pitch for two belt constructions

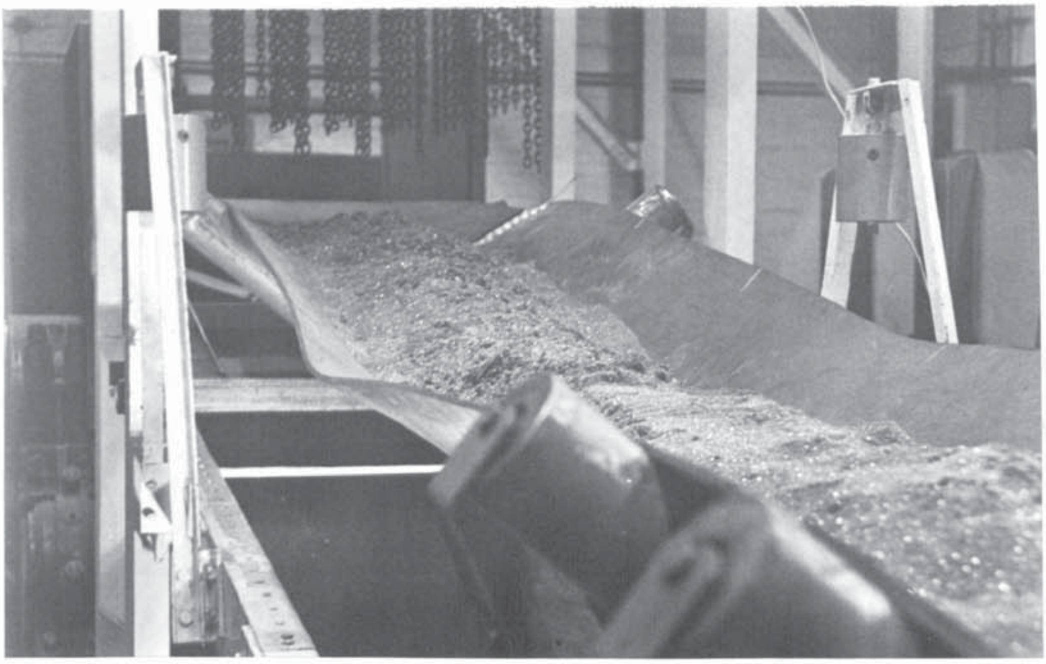


FIGURE 7.6.5

Test sample belt under low tension

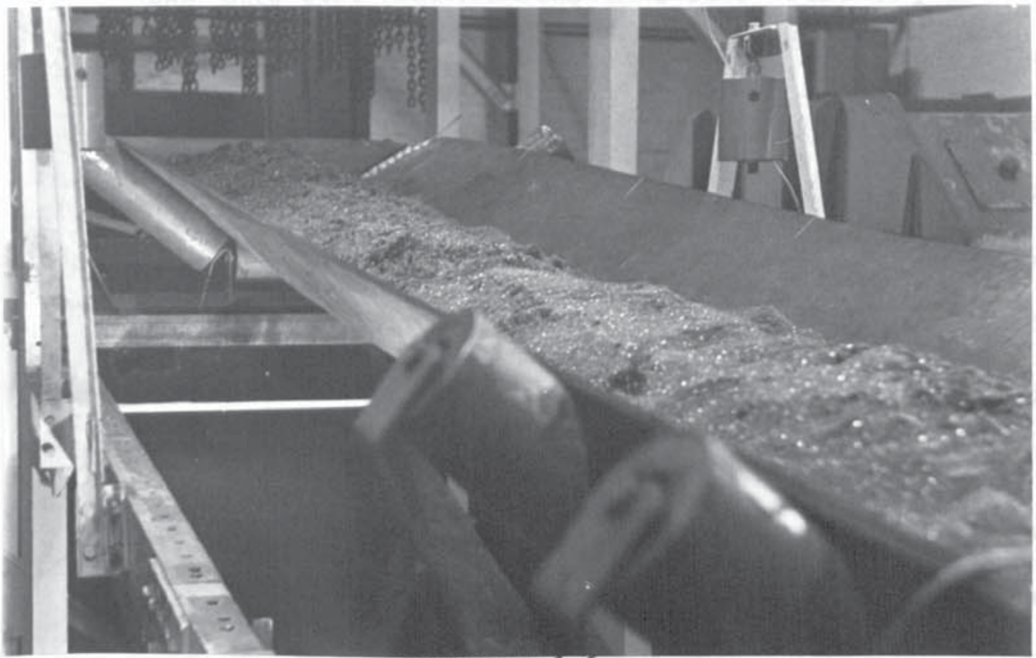


FIGURE 7.6.6

Test sample belt under high tension

NOTE:- In the belt under low tension (Figure 7.6.5) edge "folds" can be seen whereas under high tension the belt edges are straight.

(3) Strain Measurements

The measurements of the strain grid were not very successful. This was because the inaccessibility of the test rig prevented accurate measurement of the lengths between the staples. It was also found that the strain changes occurring, particularly those in the transverse direction, were not sufficient to be accurately measured on a grid length of only 250 mm. (That is, the changes in length occurring over such a distance were less than the errors introduced by the measurement technique.)

The only positive use served by the strain measurements was that they demonstrated that there were no significant differences in the strain pattern occurring in the two belt samples. They did, however, also suggest that the strain distribution in a belt is fairly complex and possibly merits further study. For example, compressive transverse strains were noted in the underside of the belt under certain conditions. This suggested that the belt shape was as shown, exaggerated, in Figure 7.6.7. This would agree with the studies by Oehmen and Alles [59] of conveyors having longer than usual idler pitches.

The three sets of results described above demonstrated that, as expected, changing the belt type on the conveyor in service would not significantly affect the phenomena observed. (This might not necessarily have

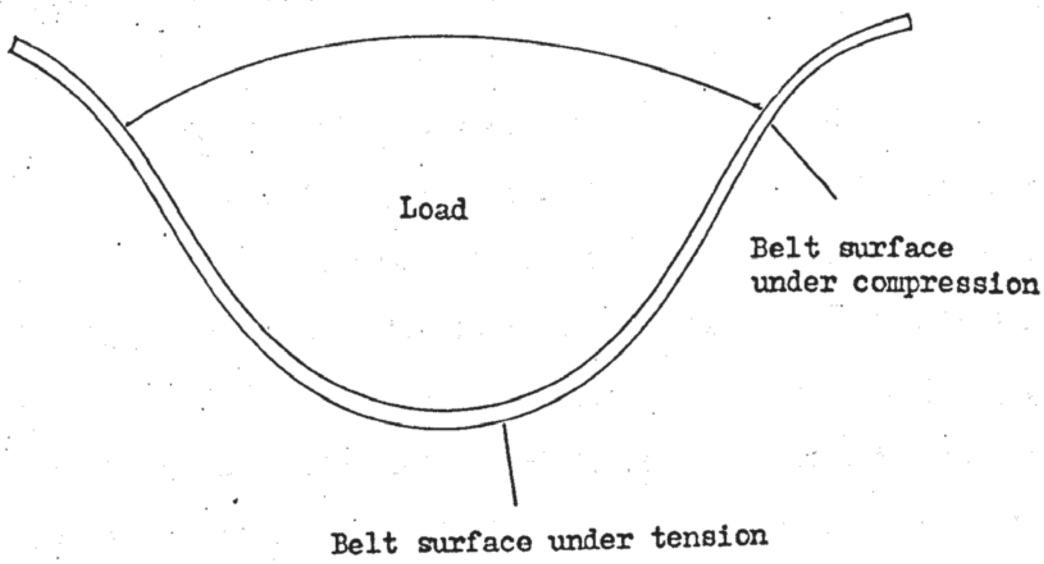


FIGURE 7.6.7

Exaggerated belt shape showing areas of belt bottom cover compression and extension observed mid-way between idler sets under certain conditions during test work been the case if the belt constructions tested had been radically different - for example, if a rubber/fabric belt had been compared with a steel-cord reinforced construction.) Some modifications were therefore proposed for the conveyor structure including suggestions for increasing the belt tension to prevent edge ripples and re-positioning some of the idlers on the moving tripper. Thus, the test work fulfilled its major purpose by showing that the belt construction installed on the working conveyor was not the cause of the observed phenomena.

Some additional small scale tests were carried out on the belt constructions used during the main experiments. These were intended to show the difference in stiffness characteristics between the two belt types. The only stiffness tests available were (1) the British Standard F/L test (see Figure 2.2.6) and (2) Stechert's pantograph test [109]. The F/L test uses a sample of the whole belt width and does not subject it to any severe deformation. Table 7.6.2 shows that results from the two belt samples for this test were very similar. The pantograph test uses a small belt sample (300 mm x 25 mm) and <u>does</u> subject it to severe deformation, bending it to a radius in the order of 40 mm. This test did show significant differences between the two belt types in both transverse and longitudinal directions. (See Table 7.6.3.) A formula given by Stechert to calculate the bending stiffness from the results of the pantograph test showed that, in the transverse direction, the belt type 1000/5 Extra was approximately three times as stiff as the construction 800/4 Extra. Both constructions were significantly stiffer in the longitudinal direction than in the transverse, as would be expected because the methods of fabric and belt manufacture result in the warp threads being straight and the weft threads being highly crimped. (See Figure 7.6.8.) Therefore, when a transverse belt sample

Belt Type		Sample deflection, F. (mm)	Sample width, L. (mm)	F/L
800/4 Extra	a	458	1200	0.38
	b	449	1200	0.37
1000/5 Extra	a	446	1200	0.37
	b	445	1200	0.37

TABLE 7.6.2

Results of British Standard Troughability Test for belts tested in connection with steelworks service problem

NOTE:- Minimum acceptable value for F/L for test conditions at hand is 0.14.

Belt Type	Sample	Distance between pantograph plates* (mm)	Bending stiffness, EJ_2+ (kN mm^2)	Average bending stiffness (kN mm^2)
800/4 Extra	Transverse A	50	14.9	14.2
	Transverse B	48	13.4	
	Longitudinal A	57	20.5	20.5
	Longitudinal B	57	20.5	
1000/5 Extra	Transverse A	87	48.2	43.8
	Transverse B	80	39.3	
	Longitudinal A	118	98.6	96.7
	Longitudinal B	116	94.8	

TABLE 7.6.3

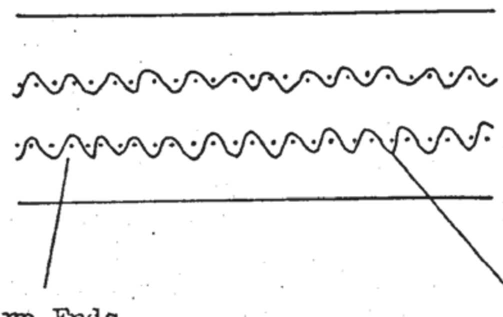
Results of Pantograph Test for belts tested in connection with steelworks service problem

* The distance between pantograph plates, x, was measured two minutes after the start of the test. It was found that a stable position had been reached by this time.

+ The bending stiffness values were calculated from Stechert's formula:

$$EJ = 0.3483 \ F \ (x - t)^2.$$

Where:- 'F' is the force applied to the pantograph plates (26.7 N in this case) and "t" is the sample thickness (mm).



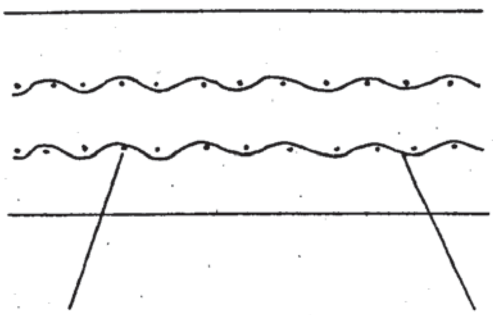


FIGURE 7.6.8

Cross-section of transverse and longitudinal belt samples is deformed it does not require much force because the actual threads are not extended, instead the crimp is removed. If, however, a longitudinal belt sample is deformed, the actual threads can be extended and as they have a relatively high modulus, this requires a much greater force than straightening out crimp.

These small scale tests demonstrated how belt stiffness is very much dependent on the degree of deformation. If only slight defromation occurs over a large belt sample, as in the F/L test, the differences between the behaviour of different rubber/fabric belt constructions are negligible. If, however, the deformation is severe and localised, appreciable differences are apparent. As the phenomena leading to the complaint discussed above were all concerned with the whole width of the belt, it was not surprising that they were exhibited by both belt constructions tested.

In addition to fulfilling its original purpose, the conveyor test work described above demonstrated several points that proved to be useful for the main investigation - that of transverse load support. These included:-

(1) The physical scale of experiments on the test rig was shown. (During the course of the work approximately two tonnes of material was lifted onto and removed from the belt.)

(2) Several days were required to set up an experiment and carry out a series of measurements.

(3) The reproducibility of results over several tension cycles was demonstrated.

(4) A new possible criterion for load support was demonstrated. This was the need to avoid belt edge "ripples", which could lead to material spillage. Measurement of inter-edge distance was therefore included in the test programme for future work. This is probably a criterion which would affect conveyor power consumption as obviously greatly increased movement of the load and belt occurs if edge ripples are allowed to exist.

(5) The observed behaviour of the belt edges lifting from the idler side rollers under some conditions as shown in Figure 7.6.9 was thought to be due to anticlastic curvature. It was decided that measurements of the distance of the belt edge from the roller should be included in the experimental work. It is interesting to note that this behaviour sometimes occurs in service as shown by Figure 7.6.10.

(6) The experiments demonstrated that direct measurement of the strain distribution was not suitable for the proposed work. Therefore, as mentioned in Section 7.3, a programme of development for a suitable strain transducer was initiated and this is described in Appendix 11.

(7) The observation of the belt pressing into the idler gap as shown in Figure 7.6.11, under the extreme load

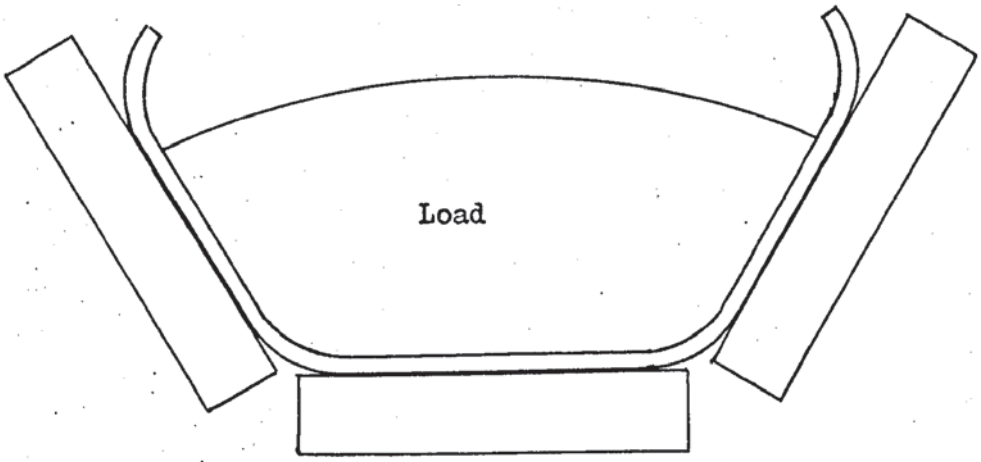


FIGURE 7.6.9

Observed phenomenon (exaggerated) of belt edges lifting from idler side rollers during test work concerned with steelworks service complaint

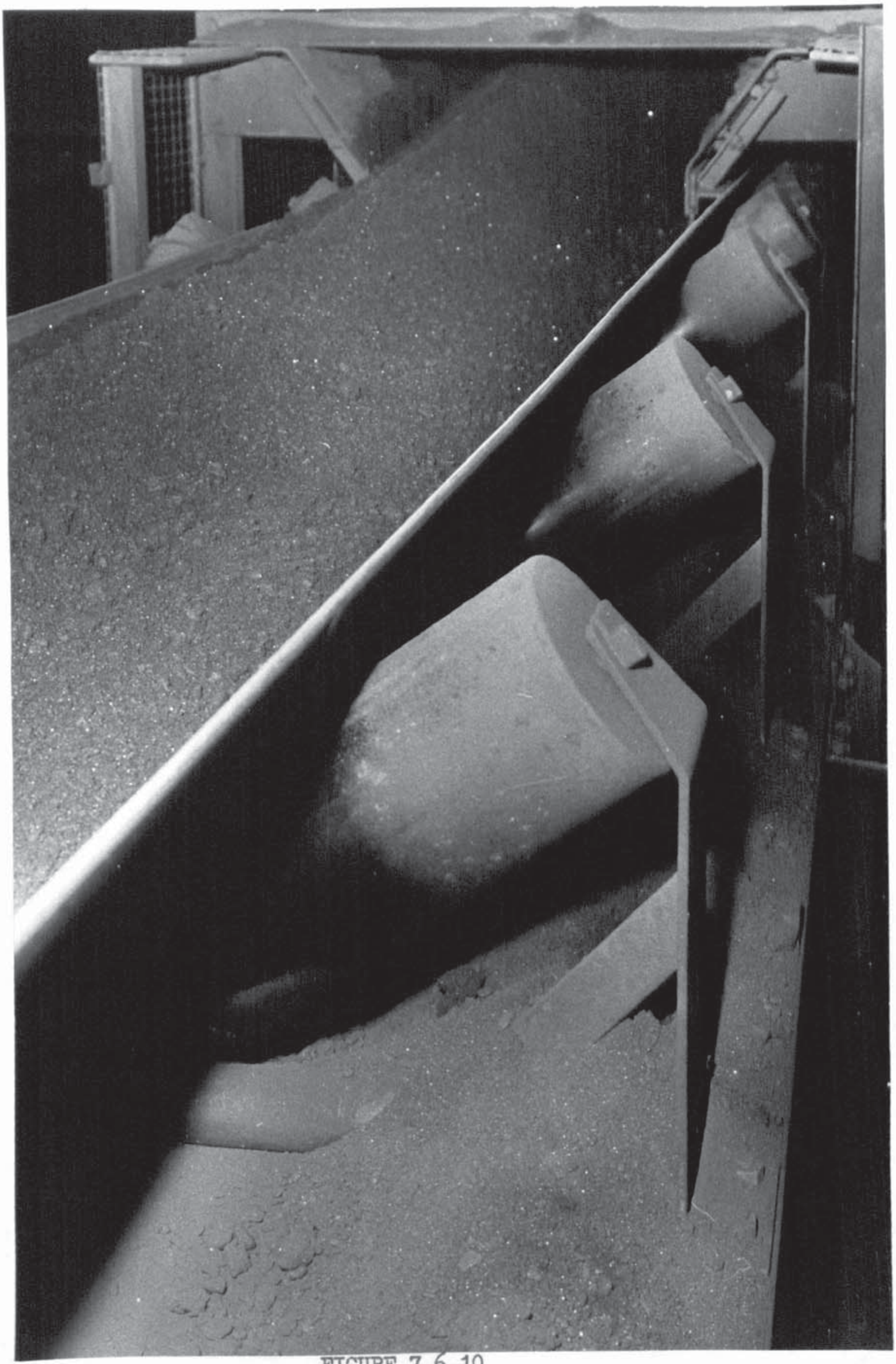


FIGURE 7.6.10

Example of belt edges lifting off side idler rollers on a working conveyor

FIGURE 7.6.11

Photograph of belt pressing into idler gap during test work concerned with a steelworks service problem conditions of the test work without any obvious damage to the belt such as a carcase fracture suggested that the strain levels occurring were not detrimental. However, on a working conveyor, damage would be caused by the moving idler rollers if the belt pressed into the idler gap, and so, it was considered that the most suitable transverse load support criterion was not that of strain occurring, but the length of non-contact between belt and idler, measured from the idler gap.

As well as demonstrating these points useful to the transverse load support investigation, the test work showed the need for both belt and conveyor parameters to be considered if an optimum design is to be obtained. The results of the test work therefore reinforced the fact that in any investigation of design properties of conveyor belts the influence of the conveyor structure must be considered. The service problem investigated thus served as an extremely useful basis for the design of an experimental procedure for the main test work. The initial experiments to develop such a procedure are discussed in the next section.

7.7 Development of an experimental procedure

The original proposals for the experiments and the additional observations made during the test work described above meant that the following measurements were to be made:-

(1) "Depths" of holes through idler "rollers". Analysis of these would give the length of non-contact between the belt and idler at the idler gap.

(2) Loads occurring at four idler supports.

(3) Distance of belt edge from idler due to anticlastic curvature.

(4) Belt shape and sag measurements at the mid-point of the idler pitch.

(5) The distance between opposite edges of the belt at the mid-point of the idler pitch.

In total, this meant that approximately ninety readings were to be taken and the variation in these was to be recorded for different belt tensions, idler angles, idler pitches, load densities and load weights. In order to reduce the total number of readings required to a more realistic figure and to develop an experimental procedure, some initial experiments were carried out on the test rig. The belt sample used for this work was an eight metre length of 1400 mm wide construction type 315/2 Heavy Duty (315/2HD). The vehicle belt, as mentioned in Section 7.1, was only 1000 mm wide, but it was thought that the joints between the two belts were far enough away from the test area for them to have negligible effect on the measurements taken. The main experiments were to be carried out with 1000 mm wide belts, but as the 1400 mm sample was available, it was considered worthwhile to carry out the initial experiments using this sample and perhaps compare the results with the other belts later. (It should be borne in mind that some manufacturers quote transverse load support data in terms of maximum permissible belt width.)

For the first experiments, the four available idler sets were positioned 0.80 metres apart with an idler angle of 35^{o}: the special idler set was one of the two centre ones. The belt sample was installed on the rig and measurements were started. The easiest parameter to vary was the belt tension and so a complete set of measurements was taken for the empty belt at two tension values. The belt was then loaded to full capacity with gravel in three equal increments, and at each stage measurements at two tensions were taken. (Photographs taken during the experimental work are given in Figures 8.1.1, 8.1.2 and 8.1.3.) When the belt was fully loaded it was left for five days and the measurements were repeated. The belt was then unloaded in stages. This procedure was repeated over two cycles of loading and it was found that no significant changes occurred in the readings taken at the same test conditions during each load cycle. It was also found that no significant changes occurred with time, thus showing that any variation in the belt deformation due to a change in the test conditions must take place within the period of approximately ten minutes taken to start measurements. Any subsequent changes which might occur due to belt creep were negligible. This was in agreement with the results found during the work described above concerned with the steelworks conveyor.

By this stage it had been confirmed that the most suitable parameter for load support was the non-contact length of belt and idler on the centre "roller" of the idler set, and this could be measured accurately by simply sliding a thin steel rule as far as possible along the centre "roller". The distance of this ruler from the idler gap could then be measured and the results were in complete agreement with those obtained by the more sophisticated technique of plotting the belt shape after having measured the effective depth of the holes through the idlers. It was also found that this parameter exhibited excellent symmetry about the centre of the conveyor. Typical results of the average value of this parameter measured at both ends of the centre "roller" at various load increments are shown in Table 7.7.1 to demonstrate the reproducibility of results over two load cycles.

The next stage of the experimentation was to repeat the measurements using the prepared bags of coal for the load. These tests demonstrated that the load weight and not the load density was the most important parameter to consider when specifying transverse load support requirements. This was shown by the fact that, at a given load weight, the non-contact length discussed above was found to be the same (within acceptable limits) for both load densities. (See Table 7.7.2.)

The initial experiments therefore provided some useful information and helped to reduce the number of measurements required because they demonstrated the following points:-

(1) There was no need to carry out several loading cycles on each belt in order to produce a stable situation before measurements could be started.

(2) There was no need to be concerned about time delays occurring between load increments as all significant changes in deformation took place within the time taken to start measurements and no variations were noted over

longer time periods.

(3) There was no need to carry out experiments with two different load densities.

These points were verified by later experiments on the first 1000 mm wide belt.

Further work with the 1400 mm wide belt demonstrated that changing the idler pitch required more time and physical effort than loading and unloading,and that the most difficult parameter to vary was the idler angle. Thus it was possible to establish a sequence in which the experiments should be carried out as follows:-

(a) Carry out measurements with empty belt at various tensions.

(b) Load belt in four stages, carrying out measurements at various tensions at each stage.

(c) Unload belt and change idler pitch.

(d) Repeat stages (a) and (b) for second idler pitch.

(e) Unload belt and change idler pitch.

(f) Repeat stages (a) and (b) for third idler pitch.

(g) Unload belt and change idler angle.

(h) Repeat all experiments in reverse idler pitch order for second idler angle.

This sequence required ninety full sets of measurements to be made for each belt sample.

The belt tension values and idler pitches actually used were chosen to represent situations likely to occur in service. For example, the three idler pitch values used were 0.80 m, 1.20 m and 1.60 m. Reference to Table A5.3 of Appendix 5 shows that these values cover the whole range of recommended idler spacings. Some doubts about the accuracy of the tension measurement have already been mentioned and so the possibility of re-calibrating the existing device to read the tension in the belt sample directly was considered. If this was to be done, a force measuring instrument capablo of withstanding approximately twelve tonnes force would have been required. As this was not available, it was not possible to carry out the calibration and the existing system had to be accepted. If further experiments requiring more accurate tension measurement are to be carried out in the future, it is suggested that a strain-gauged tie rod arrangement should be incorporated into the vehicle belt.

As the number of measurement cycles to be carried out was still very large, it was obvious that standardised results sheets would have to be developed. Two such sheets are shown in Figures 7.7.1 and 7.7.2. The sheet shown in Figure 7.7.1 was used to record the experimental conditions, the output from the load cells, the distance between opposite belt edges at the mid-point of the idler pitch ("inter-edge distance"), the belt/roller non-contact lengths SA (Side roller, side 'A'),

Load Weight (kg/m)	Non contact lengths (mm) Zero	100	200	300
Load increasing (1st cycle)	71	28	20	9
Load decreasing (1st cycle)	70	24	16	
Load increasing (2nd cycle) (After two days)	70	27	15	0
Load decreasing (2nd cycle)	68	23	14	

TABLE 7.7.1

Average of two lengths of non-contact between belt and centre idler roller showing reproducible results over two load cycles

Measurement conditions:-

Idler pitch	=	0.80 m
Idler angle	=	35^{o}
Belt width	=	1400 mm
Belt tension	=	17.8 kN
Load		Gravel

	Non contact length (m)			
Load weight	100 kg/m		200 kg/m	
Belt tension	17.8 kN	44.5 kN	17.8 kN	44.5 kN
GRAVEL ($1.5\ t/m^3$)	26	35	16	24
COAL ($0.7\ t/m^3$)	28	35	20	27

TABLE 7.7.2

Comparison of average lengths of non contact between belt and centre idler roller over two load cycles for two different loads

Measurement conditions:-

Idler pitch = 0.80 m
Idler angle = 35^{o}
Belt width = 1400 mm

SB (Side roller, side 'B'), CA (Centre roller, side 'A') and CB (Centre roller, side 'B') and the effective depths of the holes through the idlers. For the hole "depth" measurements, two figures were required; the first was the total length of the dial gauge probe and the second was the dial gauge reading when pushed up against the belt. If the second figure was subtracted from the first and then the actual thickness of the roller was also subtracted, the resulting figure would be the distance of the belt from the idler surface. The thickness of the idler roller at each hole was measured accurately and recorded on the standard results sheet as shown. Accuracy to the nearest half millimetre was possible for these measurements of the distance of the belt from the idler. The holes on the idler set were numbered so that multiplying the hole number by ten gave the distance, in millimetres, of that hole from the idler gap.

Figure 7.7.2 shows the results sheet used to record the measurements from the wooden structure positioned at the mid-point of the idler pitch. This structure was used to measure the transverse shape of the belt and also to measure the sag at the belt centre line over a short distance in the longitudinal direction. (This was done by using the holes coded LA1, LA2 etc.,) The results sheet was used to record the distance of the belt from the wooden structure in the same way as the previous sheet was used at the idler set. From these figures the actual sag of the centre of the belt could be calculated by subtracting the distance between belt and wooden structure from the length equivalent to zero sag. These measurements showed that the value of belt sag could be regarded as virtually constant over a longitudinal portion of the belt at the mid-point of the idler pitch and so, even though the test rig had a slight gradient and the actual point of maximum sag

DATE .... TEST REF:...................... BELT .... BELT WIDTH 1000 .... LOAD DENSITY ....t/m
LOAD WEIGHT ....kg/m IDLER ANGLE .... IDLER PITCH ....m TENSION READ OUT ....kN/m
NOTES:- Ambient Temperature = °C, °
Belt Surface Temperature = °C.

Multiplying idler hole number by 10 gives the distance of the hole from the end of the idler in millimetres

CENTRE IDLER 'A'		CENTRE IDLER 'B'		SIDE 'A'		SIDE 'B'	
1,	- - 53.4 = ____ mm	1,	- - 53.8 = ____ mm	1,	- - 53.2 = ____ mm	1,	- - 53.3 = ____ mm
2,	- - 53.5 = ____ mm	2,	- - 53.9 = ____ mm	2,	- - 53.2 = ____ mm	2,	- - 53.3 = ____ mm
3,	- - 53.6 = ____ mm	3,	- - 53.8 = ____ mm	3,	- - 53.2 = ____ mm	3,	- - 53.4 = ____ mm
4,	- - 53.6 = ____ mm	4,	- - 53.8 = ____ mm	4,	- - 53.3 = ____ mm	4,	- - 53.4 = ____ mm
5,	- - 53.6 = ____ mm	5,	- - 53.8 = ____ mm	5,	- - 53.4 = ____ mm	5,	- - 53.4 = ____ mm
6,	- - 53.5 = ____ mm	6,	- - 53.8 = ____ mm	6,	- - 53.4 = ____ mm	6,	- - 53.2 = ____ mm
7,	- - 53.6 = ____ mm	7,	- - 53.8 = ____ mm	7,	- - 53.4 = ____ mm	7,	- - 53.2 = ____ mm
10,	- - 53.6 = ____ mm	10,	- - 53.7 = ____ mm	8,	- - 53.4 = ____ mm	8,	- - 53.2 = ____ mm
11,	- - 53.6 = ____ mm	11,	- - 53.8 = ____ mm	9,	- - 53.4 = ____ mm	9,	- - 53.3 = ____ mm
				10,	- - 53.4 = ____ mm	10,	- - 53.2 = ____ mm
				11,	- - 53.4 = ____ mm	11,	- - 53.2 = ____ mm
				12,	- - 53.4 = ____ mm	12,	- - 53.2 = ____ mm
				13,	- - 53.3 = ____ mm	13,	- - 53.2 = ____ mm
				14,	- - 53.2 = ____ mm	14,	- - 53.1 = ____ mm
				15,	- - 53.2 = ____ mm	15,	- - 53.1 = ____ mm

LOAD CELL READINGS

1, READ OUT = ... V = ... N = ...kgf
2, READ OUT = ... V = ... N = ...kgf
3, READ OUT = ... V = ... N = ...kgf
4, READ OUT = ... V = ... N = ...kgf

LOAD DETAILS

INTER - EDGE DISTANCE = ____ mm
NON CONTACT LENGTH SA = ____ mm
NON CONTACT LENGTH CA = ____ mm
NON CONTACT LENGTH SB = ____ mm
NON CONTACT LENGTH CB = ____ mm

FIGURE 7.7.1

Standardised results collection sheet 'A' for experimental investigation (Reduced Size)

BELT PROFILE MEASUREMENT

DATE .... TEST REFERENCE .... ZERO SAG FOR MOLE '0' = ...49.2 mm.........

Mole Number		Actual Sag	
0,	.... - .... - 20.6 .... = ________ mm,	________ mm	
LA 1,	.... - .... - 20.8 .... = ________ mm,	________ mm	
LA 2,	.... - .... - 20.6 .... = ________ mm,	________ mm	
LA 3,	.... - .... - 20.7 .... = ________ mm,	________ mm	
LB 1,	.... - .... - 20.5 .... = ________ mm,	________ mm	
LB 2,	.... - .... - 20.4 .... = ________ mm,	________ mm	
LB 3,	.... - .... - 20.0 .... = ________ mm,	________ mm	
CTA 1,	.... - .... - 20.5 .... = ________ mm		
CTA 2,	.... - .... - 20.7 .... = ________ mm		
CTA 3,	.... - .... - 20.4 .... = ________ mm		
CTA 4,	.... - .... - 20.7 .... = ________ mm		
CTA 5,	.... - .... - 20.3 .... = ________ mm		
CTA 6,	.... - .... - 20.3 .... = ________ mm		
CTA 7,	.... - .... - 19.8 .... = ________ mm		
CTA 8,	.... - .... - 20.1 .... = ________ mm		
CTA 9,	.... - .... - 20.0 .... = ________ mm		

Mole Number	
STA 1,	.... - .... - 19.2 .... = ________ mm
STA 2,	.... - .... - 19.1 .... = ________ mm
STA 3,	.... - .... - 19.4 .... = ________ mm
STA 4,	.... - .... - 19.4 .... = ________ mm
STA 5,	.... - .... - 19.6 .... = ________ mm
STA 6,	.... - .... - 20.0 .... = ________ mm
STA 7,	.... - .... - 19.8 .... = ________ mm
STA 8,	.... - .... - 19.9 .... = ________ mm
STA 9,	.... - .... - 20.0 .... = ________ mm
STA 10,	.... - .... - 20.1 .... = ________ mm
STA 11,	.... - .... - 20.0 .... = ________ mm
STA 12,	.... - .... - 20.0 .... = ________ mm
STA 13,	.... - .... - 20.1 .... = ________ mm
STA 14,	.... - .... - 20.2 .... = ________ mm
STA 20,	.... - .... - 19.0 .... = ________ mm
STA 21,	.... - .... - 18.8 .... = ________ mm
STA 22,	.... - .... - 18.4 .... = ________ mm
STA 23,	.... - .... - 18.1 .... = ________ mm

FIGURE 7.7.2

Standardised results collection sheet 'B' for experimental investigation (Reduced Size) probably would be displaced slightly from the idler pitch mid-point, the reading at the mid-point could still be regarded as the maximum for the purposes of this work.

It can be seen that many measurements were required if these two sheets were to be completed. It was not feasible to do this for every single conveyor condition and so a shortened test method was devised for those conditions where measurement of the belt shape was not required. The results sheet for this shortened test is shown in Figure 7.7.3. This form was used to record the test conditions, the load washers' outputs, the belt sag at the centre of the belt mid-way between idlers, the belt/roller non-contact lengths, the inter-edge distance and the distances of the belt edges from the idler surfaces. As it was found that the changes in belt shape at each load increment were not severe, it was decided that this shortened form of results sheet should be used for "intermediate" load conditions. (That is, for loads in-between the first load increment and the fully loaded belt.) This meant that the time taken to complete a load cycle and subsequently analyse the results was reduced to a more workable level. Even so, testing each belt sample fully required two people working for six days and involved lifting, in all, 7.2 tonnes of material on and off the conveyor as well as moving idlers at various stages during the experiments. As the investigation progressed typical patterns of behaviour were noted and so it was possible to reduce the number of experiments required for later belt samples.

The results of this work and their analysis are discussed in the next Chapter.

RESULTS SHEET

DATE ....

BELT ....

IDLER PITCH = ...m

TEST REFERENCE ....

TENSION READ OUT = ...

BELT WIDTH = ...1000 mm................

LOAD = ...

IDLER ANGLE = ...

LOAD WEIGHT = ...kg/m

AMBIENT TEMP = ... °C

SURFACE TEMP = ... °C

LOAD CELL READINGS:-

1, READ OUT = ... x 10^{-5} V = ... N = ... kgf

2, READ OUT = ... x 10^{-5} V = ... N = ... kgf

3, READ OUT = ... x 10^{-5} V = ... N = ... kgf

4, READ OUT = ... x 10^{-5} V = ... N = ... kgf

SAG = ... − .... + 69.8 = ... mm = ... %

INTER-EDGE DISTANCE = ... mm

NON CONTACT LENGTH SA = ... mm

NON CONTACT LENGTH CA = ... mm

NON CONTACT LENGTH SB = ... mm

NON CONTACT LENGTH CB = ... mm

BELT EDGE – IDLER SURFACE, SIDE 'A' .... mm

BELT EDGE – IDLER SURFACE, SIDE 'B' .... mm

LOAD DETAILS

FIGURE 7.7.3

Standardised results collection sheet for shortened form of experimental investigation (Reduced Size)

CHAPTER 8

RESULTS AND CONCLUSIONS FROM EXPERIMENTAL INVESTIGATION OF TRANSVERSE LOAD SUPPORT

The method by which an experimental procedure was developed for the investigation of transverse load support has been described in Chapter 7; the main objectives of the investigation were (a) to establish a criterion by which transverse load support could be assessed and (b) to study the variation of load support with changes in certain belt, load and conveyor parameters. As the setting-up of any experiment on the available test rig was to be an expensive and time consuming exercise, it was decided that other facets of belt behaviour should also be investigated. Accordingly, belt sag measurements and a study of the phenomenon of the belt edges lifting clear of the side rollers of an idler set were made.

This chapter describes how the original investigation objectives were fulfilled and gives examples of the belt behaviour recorded during the work.

8.1 Experimental Details

The basis for the experimental conditions has already been described; the actual values of the parameters varied during the main test programme are shown in Table 8.1.1. Details of the belt samples obtained for the work are shown in Table 8.1.2 and Figures 8.1.1, 8.1.2 and 8.1.3 are general-view photographs taken during the work.

Idler angles	35° and 45°
Idler pitches (m)	0.8, 1.2, 1.6
Load increments (with gravel) (kg/m)	0, 50, 100, 150, 200
Belt tensions (kN/m)	17.8, approximately T_r, one intermediate value

TABLE 8.1.1

Values of belt and conveyor parameters used during the main test work

NOTE:- T_r = Rated working tension of belt construction.

Belt type	315/2HD	630/4EX	630/4HD
Belt width (mm)	1000	1000	1000
Tensile strength (kN/m)	315	630	630
Interply amount code	Heavy Duty	Extra	Heavy Duty
Number of plies	2	4	4
Fabric code	8814	8814	8814
Top cover thickness (mm)	3.0	5.0	5.0
Bottom cover thickness	1.5	1.5	1.5
Approximate carcase thickness (mm)	3.5	4.5	5.5
Overall thickness (mm)	8.0	11.0	12.0
Belt weight (kg/m)	8.9	11.7	13.1
Selling price of sample (8m length) July, 1976	£140	£221	£249

TABLE 8.1.2

Details of belt samples used during main test work

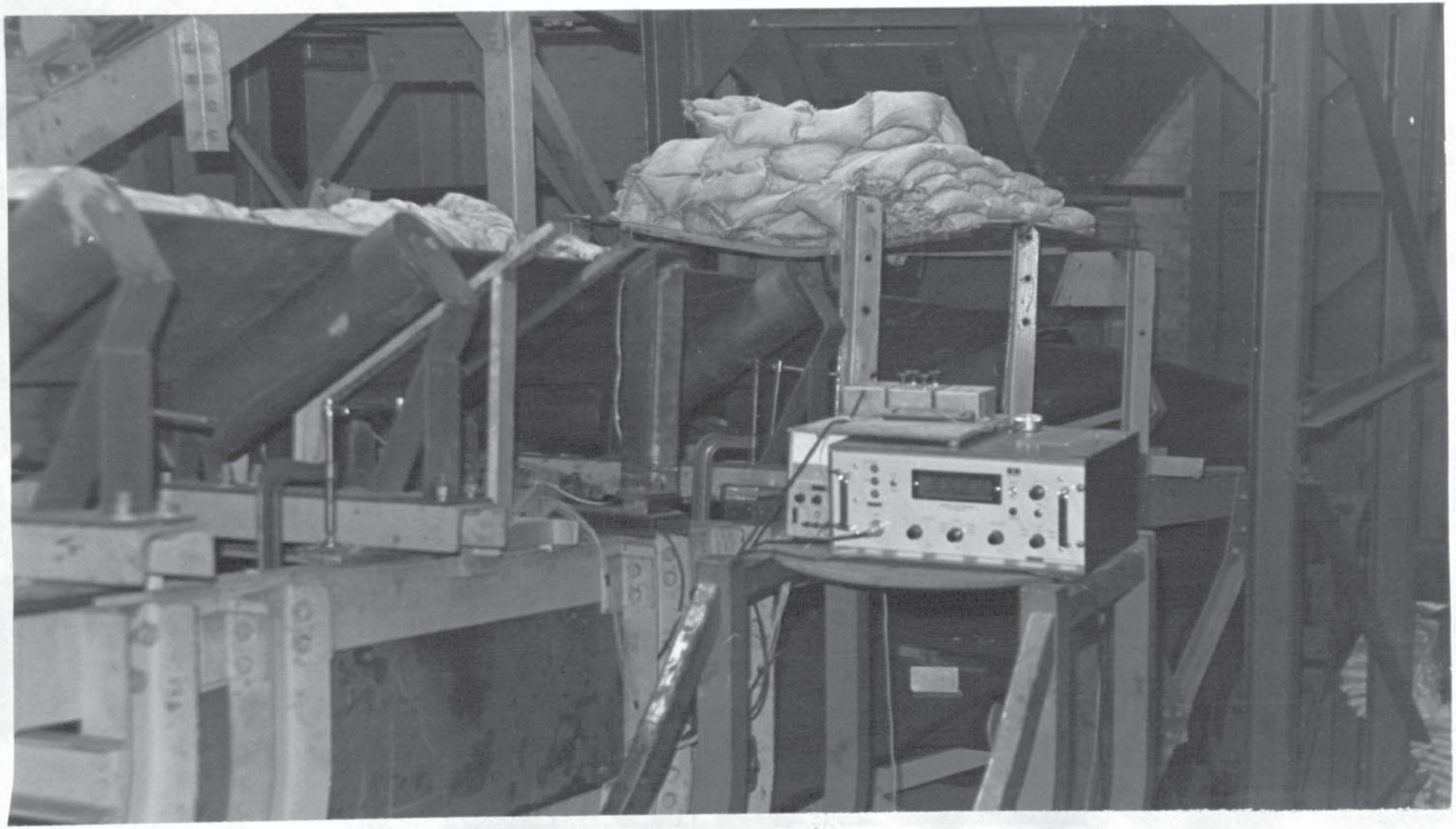


FIGURE 8.1.1

General view of test equipment

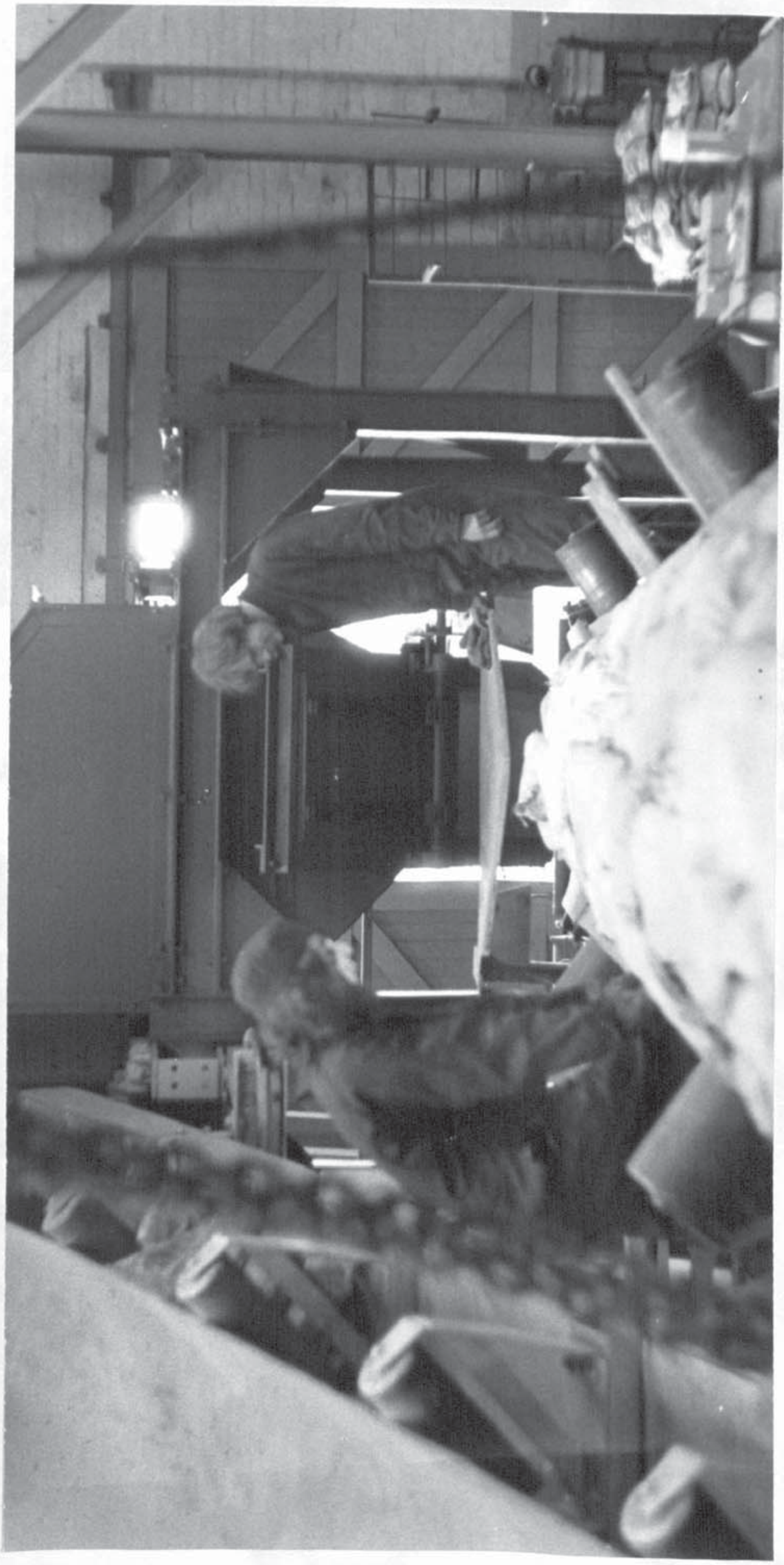


FIGURE 8.1.2

General view of test belt in fully loaded condition

FIGURE 8.1.3

Measurement of belt "inter-edge distance" at half capacity loading

It has been stated in Chapter 7 that setting-up and carrying out a test for each belt sample required two persons working for six days. It was estimated that the labour and overheads cost of each test was therefore in the order of £300: the average "opportunity cost" of the belt samples, equal to their selling price, was approximately £200, and so it can be seen that the investigation, even using an existing test rig, was expensive; any large-scale experimental study of conveyor belting must therefore be carefully prepared to ensure that it is carried out as efficiently as possible and must be justified by the possible benefit to be derived from the results. The possible benefit of the present work was, of course, a better understanding of the mechanism of transverse load support which could lead to a more efficient design of belt construction providing more economic materials handling.

During the test work some minor improvements that could be made to the special idler set were noted; these included:-

(a) The holes through the side "rollers" of the idler set only extended to approximately one third of the "roller" length from the idler gap. This was intended to reduce production costs to a minimum, and it was anticipated, anyway, that the belt would be touching all the remainder of the idler. In practice it was found that the belt edge lifted off the roller as shown in Figure 7.6.9. If the holes had been along the complete length of the side "roller", it would have been possible to measure the actual shape of the belt near its edge. This is a relatively easy modification to the structure that could be made if any further test work was carried out using the equipment. It should be remembered, however, that increasing the number of holes would have increased the time for measurement and result analysis and would only have resulted in the study of "tubing", an interesting facet of belt behaviour which has no obvious detrimental effect on service performance.

(b) The idler design was such that the idler angle was fixed at 35° or 45°. (An additional set of supports corresponding to a troughing angle of 55° was also made for future work if the other idlers on the test rig are adapted to this angle.) It was found that the supports that should have corresponded to 45° had been manufactured in such a way that the idler angle was slightly less than this value (by about 2°). This was not, of course, a design fault of the idler set and did not prove to be significant, but resulted in the unloaded belt samples not quite touching the side rollers of the 45° troughing idler when a high tension was induced in the belt. As soon as any load at all was placed on the belt, it deformed enough to touch the side roller.

If the idler set had been designed so that the troughing angle was not fixed but could be varied, this minor problem would not have arisen.

Some other problems that arose, such as the doubts about the accuracy of the tension measurement and the poor output of the load washers, have been discussed previously. However, neither these nor the problems discussed above invalidated the study of transverse load support; the results of which are discussed in the next section.

8.2 Results of Transverse Load Support Investigation

The first 1000 mm wide belt sample to be placed on the test rig was the construction type 630/4EX. Initial experiments with this confirmed the results obtained with the 1400 mm wide belt originally installed. That is, it was found that no significant differences occurred in the belt deformation measured during the first and second cycles of loading to full capacity and unloading, that load density had little effect on the investigation results, and that all significant changes in belt deformation occurred within the five to ten minutes taken to start measurements after a change in load conditions. All further tests were therefore carried out using only one load material and no load cycles prior to measurement were considered to be necessary to establish a stable situation.

The criterion by which transverse load support should be assessed had already been suggested by the previous tests, and work with the construction 630/4EX confirmed that measurement of the length of non-contact between the idler centre "roller" and the belt represented the most sensitive relevant parameter for this. The reasons for deciding this were as follows:-

(a) Visual examination of the belt suggested that the transverse strains occurring in the belt were not severe even when the belt was pressed into the idler gap. This was confirmed by simple theory (see Appendix 12) showing that the maximum strain levels that could occur in a belt before the length of centre-roller/belt non-contact became zero were not likely to be detrimental to the service life of the belt for constructions at present in common use.

(b) The cyclic flexing that occurs in a belt as it passes through an idler pitch was shown not to be severe from the measurements of belt shape at and between idler positions. This can be seen from Figure 8.2.1.

(c) If a belt is pushed into the idler gap, it will rub against the rotating edge of the idler rollers. This will eventually lead to the breakdown of the belt - as was observed when all-synthetic belt constructions were initially introduced. It was found during the experiments that the belt, at any given load condition, approached nearer the idler gap on the centre roller than on the side roller. This would be expected because most of the load weight is taken by the centre roller of an idler set and so greater forces act in this area than at the sides of the idler set. This was confirmed by the output of the load washers which showed that the percentage of the load taken by the centre roller was at least as much as that shown in Figure 7.2.3.

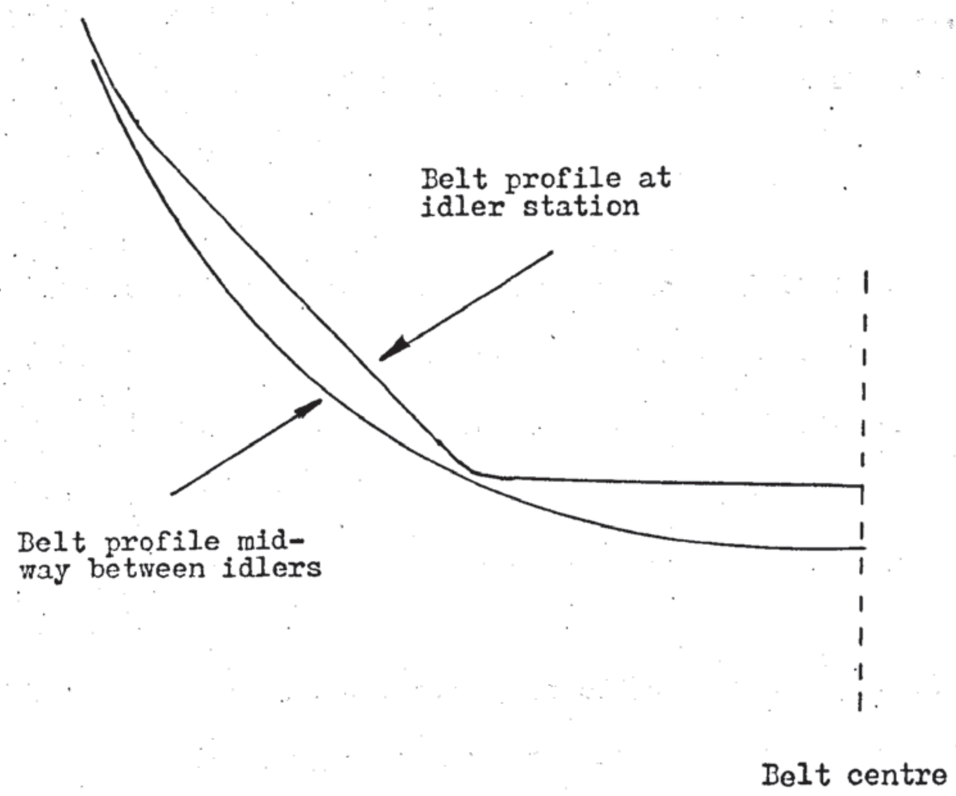


FIGURE 8.2.1

Belt profile at idler station and mid-way between adjacent idler sets (Scale 1:4)

Test conditions:-

Belt type = 630/4 HD

Belt tension = 17.8 kN/m

Load weight = 200 kg/m

Idler pitch = 1.60 m

Idler angle = 35°

The centre-roller/belt non-contact length was therefore established as being the most relevant criterion by which transverse load support could be assessed for the conditions being studied. It must be remembered that if different forms of belt constructions were being tested it might have been found that other criteria required consideration. For example, if all-cotton belts were under investigation, it might be found that the strain levels occurring would, in fact, be detrimental to the belt. This is suggested by the Architect's Journal which states [7] :-

> "Belts of cotton or canvas construction are limited to angles of 30 degrees as the carcase is not flexible enough to trough at steeper angles without damage. Synthetic fibre carcase belts can trough at angles up to 60 degrees, and can carry a bigger load."

It should be noted that the latter part of this statement could be misleading because troughing angles are now commonly limited to only 45° in practice.

Having decided that the most suitable criterion of transverse load support was the length of non-contact discussed above, its variation with changes in various belt parameters was studied. Typical results obtained from this work are shown in Figures 8.2.2 to 8.2.8; in each of these the average non-contact length obtained from measurements at both ends of the centre "roller" is plotted. It was found that the measured parameter was, as would be expected, symmetrical about the belt centre and the discrepancy between the two "roller" ends was less than ten millimetres at all values of the parameter being measured.

As experiments were only carried out at three values of some conveyor parameters, it was difficult to plot a "curve shape" from only one set of measurements. However, as so many experiments were carried out it was possible to establish general patterns of belt behaviour.

Figure 8.2.2 shows the observed variation of belt/centre"roller"non-contact length with belt tension. From this, it can be seen that as belt tension increases the load support capability of the belt and conveyor is improved. This was because the higher belt tension meant that the belt itself carried a greater proportion of the load weight and so the idler "rollers" were not so heavily loaded; thus the length of non-contact between belt and "roller" increased. However, as only a relatively small variation in this parameter was noted over a large tension range,it was realised that belt tension need not be regarded as a major factor in the provision of transverse load support. It must, however, be noted that a minimum tension must be present to avoid the edge folds observed between idler sets during the test work described in Section 7.6. These edge folds could lead to spillage of material and so, by definition, could cause inadequate load support. Edge folds were not observed under any conditions during the work described in this chapter and so it was concluded that, under normal conditions, only a fairly low tension is required to ensure that they do not occur.

Figure 8.2.3 demonstrates that a much greater variation of non-contact length was observed with increasing load weight; thus showing that the load weight per metre of belt is a very important factor to be considered when deciding the amount of transverse load support capability required by a given conveying situation. Figure 8.2.4 is a curve of non-contact length against idler pitch. As idler pitch increases

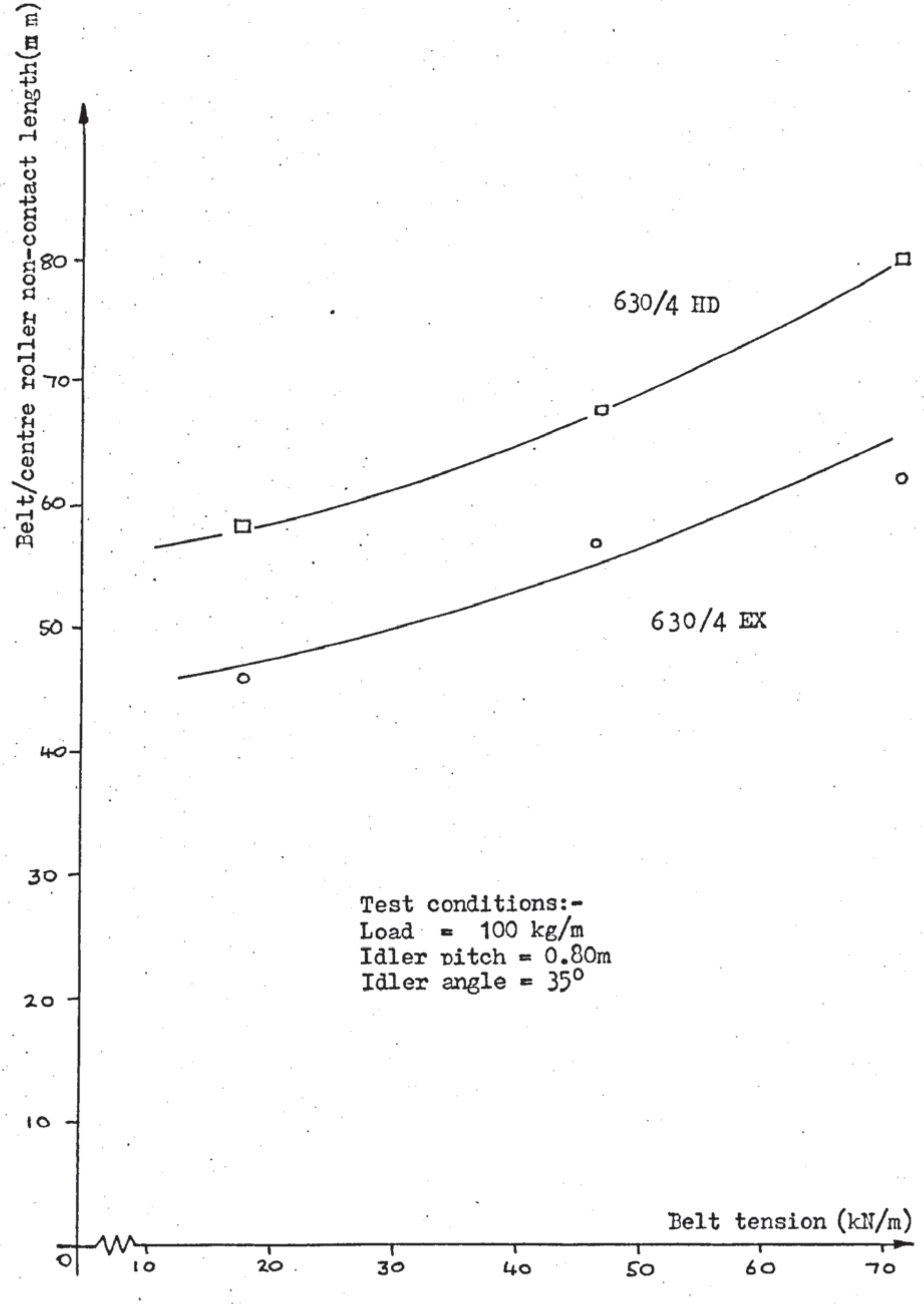


FIGURE 8.2.2

Variation of belt/centre roller non-contact length with belt tension for two belt constructions

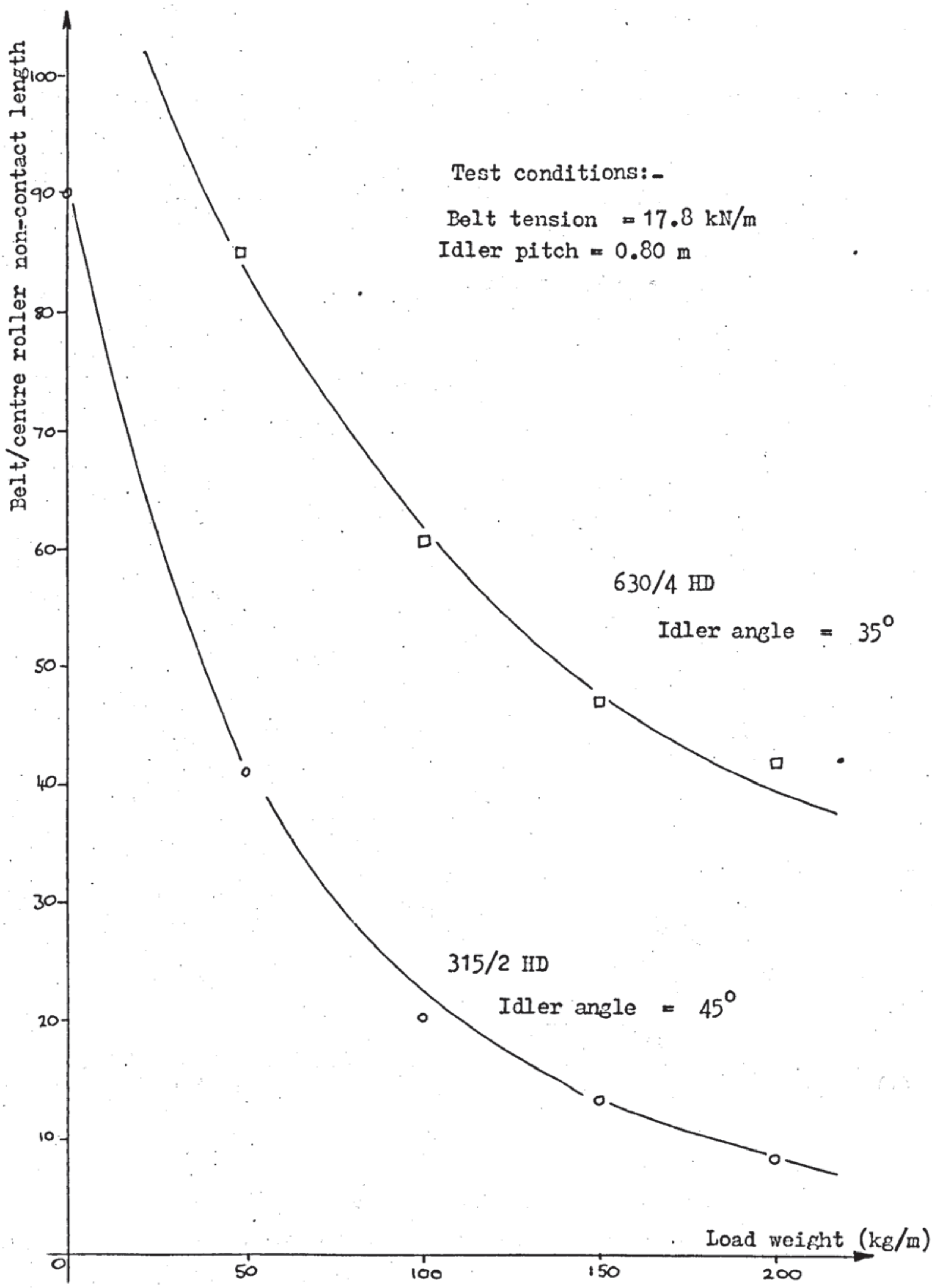


FIGURE 8.2.3

Variation of belt/centre roller non-contact length with load weight for two belt constructions

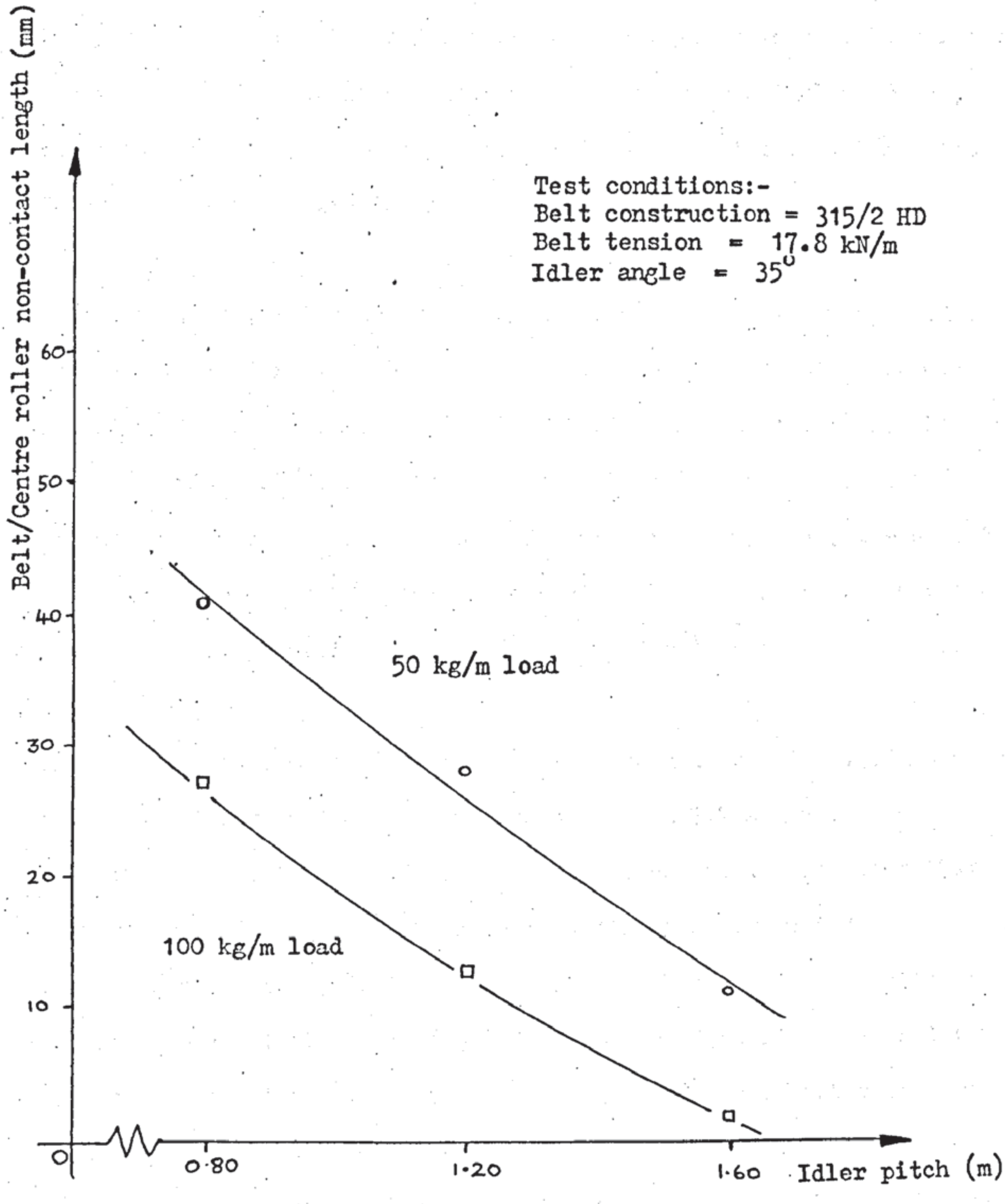


FIGURE 8.2.4

Variation of belt/centre roller non-contact length with idler pitch at two belt load weights the load per idler set obviously also increases for a certain belt loading per metre. The observed results showed that the non-contact length was very dependent on idler pitch and that, for practical purposes, the relationship could be regarded as linear. That is, doubling the idler pitch would halve the non-contact length.

Figure 8.2.5 shows the influence of idler angle on transverse load support by a plot of non-contact length against belt load weight for two different troughing angles with all other conditions the same. It can be seen that at 45° troughing the non-contact length tended to be slightly greater than at 35°, but, for practical purposes, the effect of idler angle could be regarded as negligible. This contradicts the claimed advantage of improved load support for deep trough conveyors.

Figure 8.2.6 is a comparison of results obtained from the 1400 mm wide belt tested previously as described in Section 7.7 and a 1000 mm wide belt of the same type tested during the present work. It shows that only minor differences in the non-contact length were noted at a given loading condition. This is almost certainly because the majority of the load weight is taken by the centre roller of an idler set and thus, for normal situations where the load is spread over a belt width greater than the centre roller length, the overall width of the belt has no effect on the transverse load support capability of the belt and conveyor. It must, however, be remembered that increasing the belt width will increase the carrying capacity of a conveyor and so greater inherent belt transverse load support is required to cope with this.

The above results show the influence of the various conveyor parameters on the criterion which had been chosen as the most relevant measure of

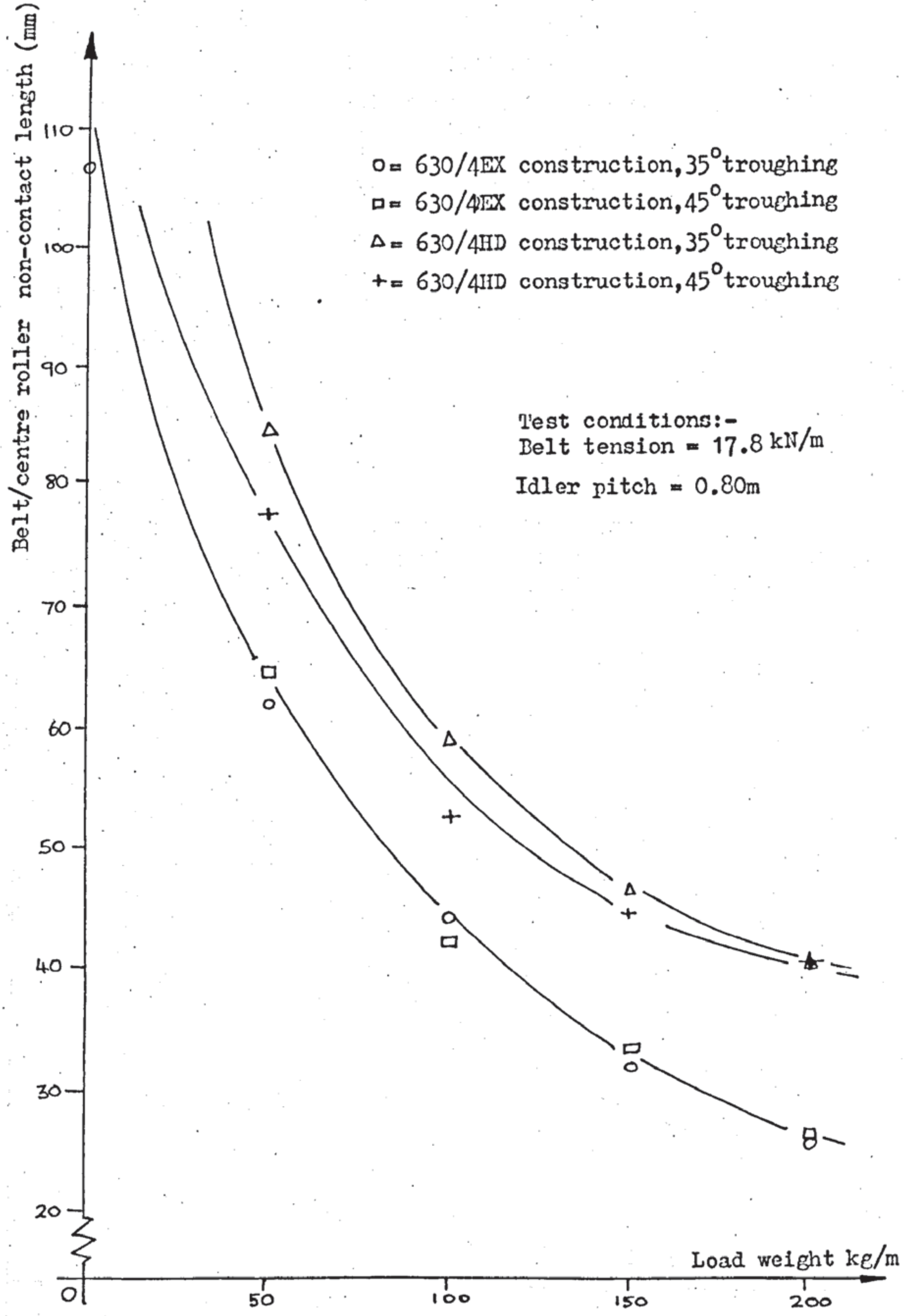


FIGURE 8.2.5

Plots of belt/centre roller non-contact length against load weight to demonstrate the effect of idler angle

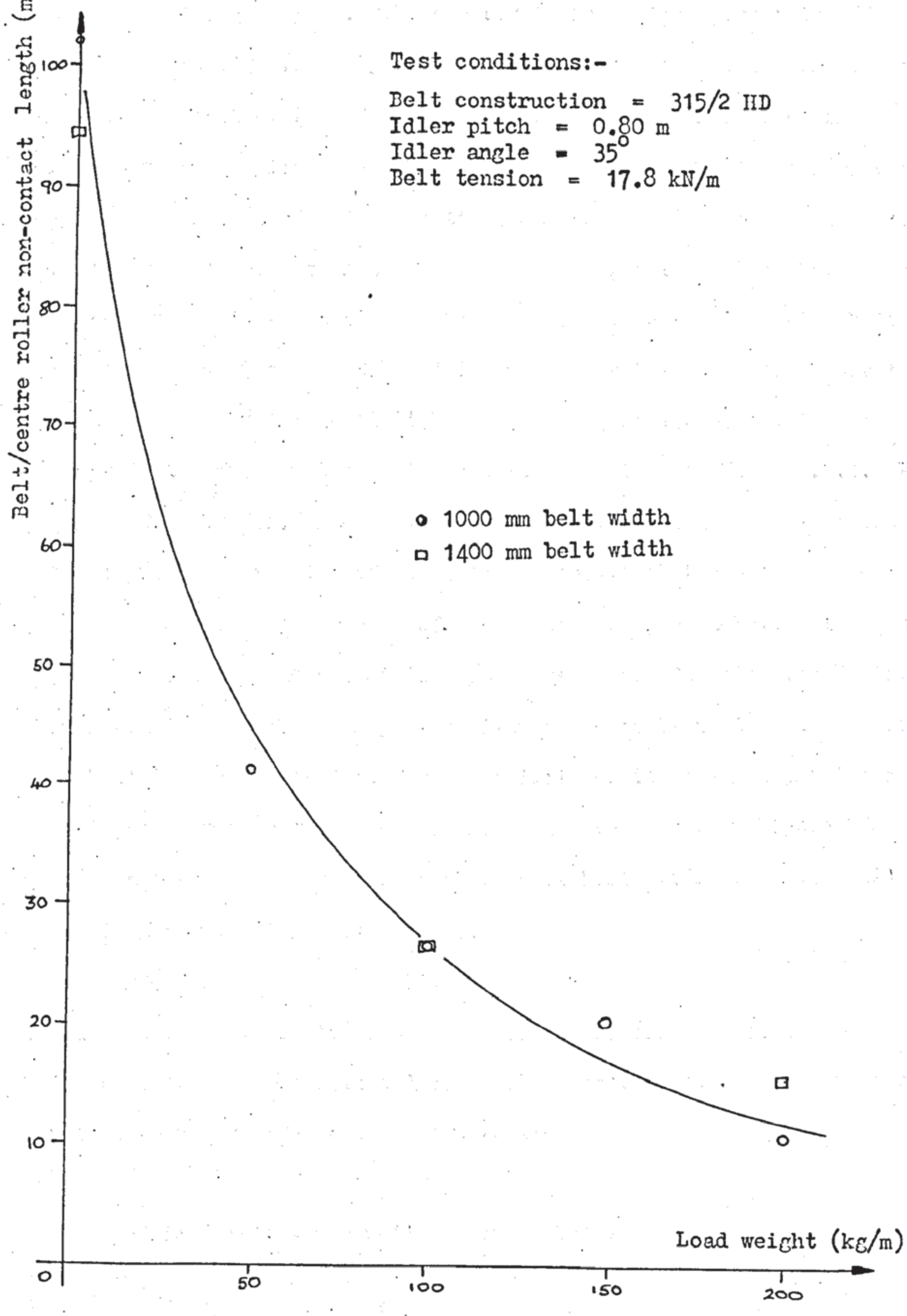


FIGURE 8.2.6

Plots of belt/centre roller non-contact length against load weight to demonstrate the effect of belt width transverse load support. The one remaining factor thought to affect load support significantly was that of belt construction. The variation of non-contact length with belt construction can be seen from some of the diagrams already described but in order to provide a more useful demonstration of this variation, an assessment of transverse belt stiffness was made as this is the belt property thought to have most effect on the inherent transverse load support capability of a belt construction. The methods used for measuring belt stiffness were the British Standard Troughability Test (see Figure 2.2.6) and the Pantograph Test. The results of these tests are given in Table 8.2.1. Figures 8.2.7 and 8.2.8 respectively show the variation of non-contact length with the results of these two tests. Both these figures demonstrate the strong dependence of non-contact length on belt stiffness, but Figure 8.2.7 shows that the F/L value recorded during the British Standard test is not really sensitive enough to be used as a measure of belt transverse load support capability as it does not provide adequate scatter of results. However, as can be seen from Figure 8.2.8, the Pantograph test provides a measure of belt stiffness that could possibly be used for this purpose.

The experimental results summarised in the figures described above demonstrate that the belt and conveyor parameters having the greatest effect on transverse load support were:-

(a) Load weight per unit length of belt.

(b) Idler pitch.

(c) Belt stiffness.

Belt construction	Average F/L value sample	Average longitudinal bending stiffness (kN mm^2)	Average transverse bending stiffness (kN mm^2)
315/2HD	0.49	6.4	5.1
630/4EX	0.36	20.8	15.5
630/4HD	0.35	36.0	21.4

TABLE 8.2.1

Results of British Standard Troughability Test and Pantograph Test for belt constructions tested during experimental investigation

NOTE:- The Pantograph Test was carried out as described under Table 7.6.3.

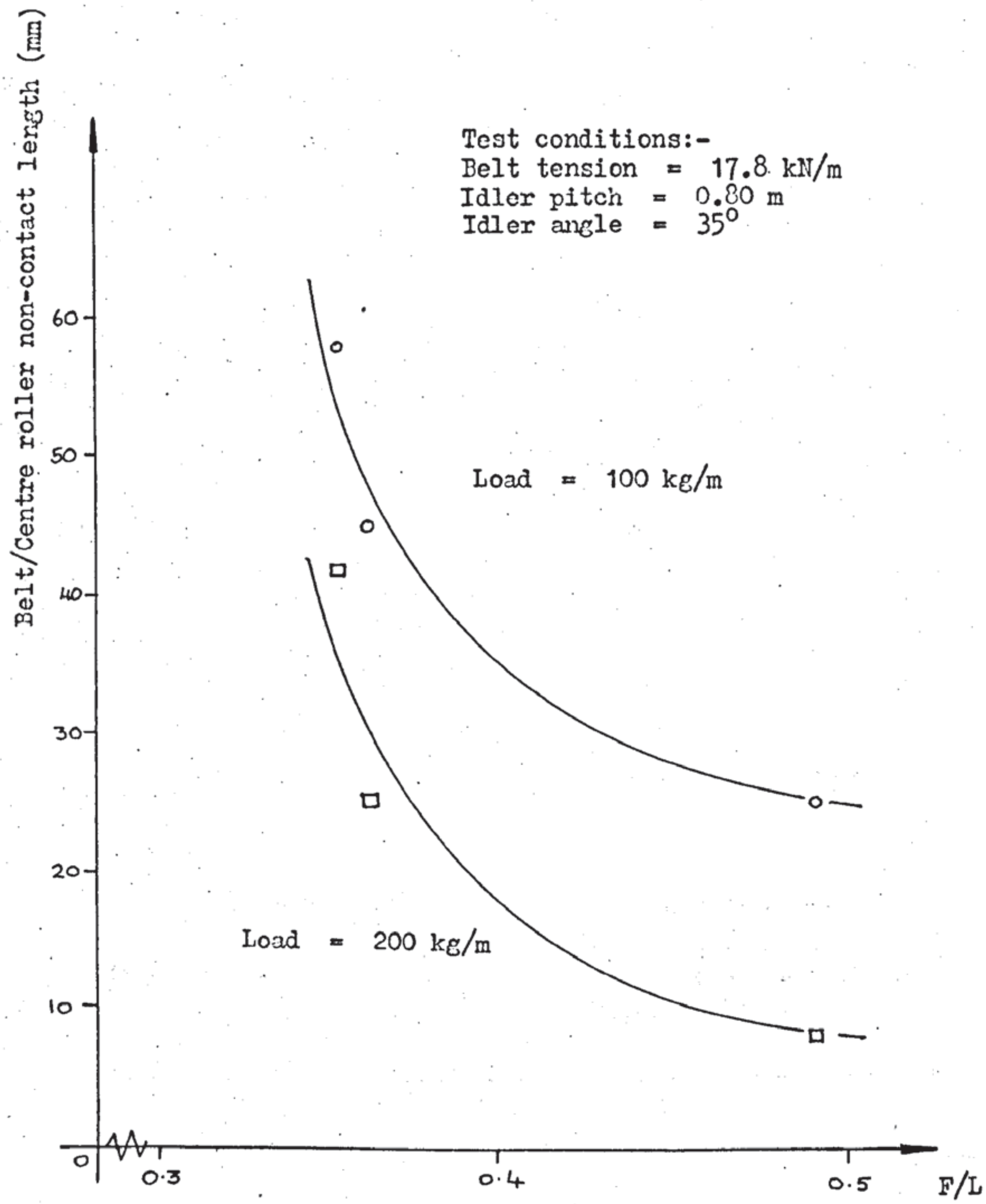


FIGURE 8.2.7

Variation of belt/centre roller non-contact length with F/L value recorded from the British Standard Troughability Test

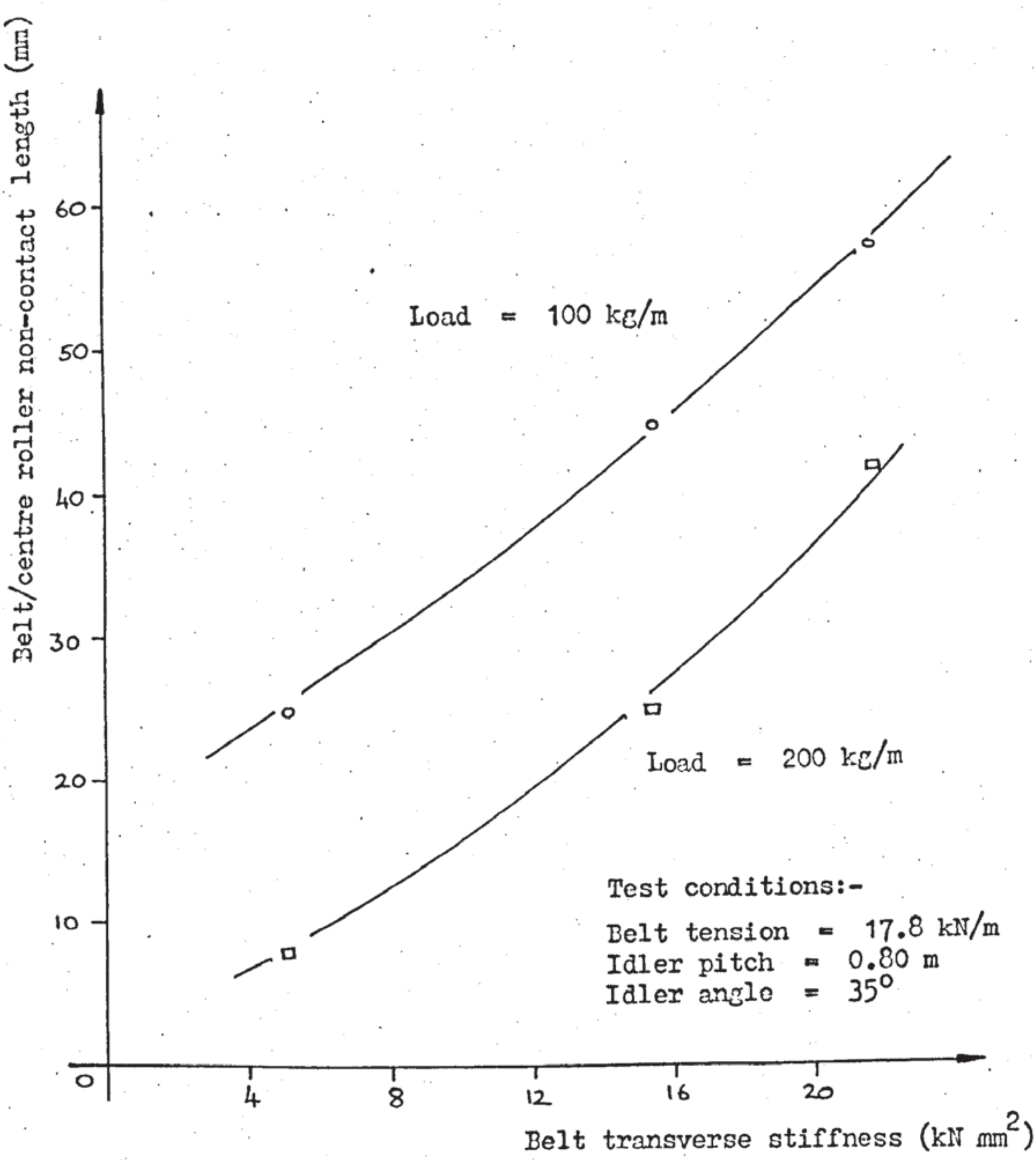


FIGURE 8.2.8

Variation of belt/centre roller non-contact length with belt transverse stiffness recorded by Pantograph Test

Others, such as belt tension, idler angle, load density and belt width were shown to have negligible effect; but it should be remembered that certain of these affect the total load weight that can be carried on a belt. That is, increasing belt width, load density or idler angle without altering other conveyor parameters will increase the maximum possible load weight per unit length of belt. Comments on the significance of these results are made below in Section 8.3 as part of the conclusions drawn from the experimental work.

The remainder of the experimental work could be divided into three sections:-

(1) Belt sag measurements.

(2) Inter-edge distance measurements.

(3) Belt shape measurements.

The results of these investigations are given, in order, below.

Belt Sag Measurements

It was possible, with the wooden structure described in Section 7.1, to measure the sag at several points along the centre line of the belt. It was found that no significant variation of this parameter occurred along a length extending approximately 150 mm either side of the point mid-way between idler sets. Therefore the sag at the centre point only was measured even though the slight gradient of the conveyor would cause a small displacement of the point of maximum sag from this position. The results obtained were compared with those obtained from the use of the accepted formula given in equation 2.2.10. Typical examples of the behaviour observed are given in Figures 8.2.9, 8.2.10 and 8.2.11 and in each of these figures a theoretical behaviour curve is shown. It can be seen from equation 2.2.10 that belt weight is considered to affect the amount of sag; it was found that the variation in the theoretical sag values caused by this factor for the different belt constructions tested was negligible, and so only one theoretical curve corresponding to the average belt weight was drawn.

Figure 8.2.9 shows the observed variation of inter-idler belt sag with belt tension. It can be seen that, as would be expected from the formula, belt sag decreased with increasing tension. However, the observed behaviour did not follow the theoretical curve exactly. This pattern was in agreement with that recorded previously in the test work concerned with the steelworks service complaint described in Section 7.6 and the results of Behrens [28] shown in Figure 2.2.5. However, typical behaviour of sag with two different idler angles is also shown in Figure 8.2.9 and this does not agree with that observed by Behrens. In the present work, it was found that sag was slightly greater at 45° than at 35° whereas Behrens found that sag at 45° was slightly less. The differences occurring with idler angle are small and can probably be regarded, for practical purposes, as negligible, but it is interesting to note that the increased beam effect claimed as an advantage for deep trough conveying does not materialise under all conditions. Behrens work was carried out using a fine material on a steel-cord belt construction. This particular type of belt has virtually no transverse stiffness and so the increased beam effect caused by increasing the

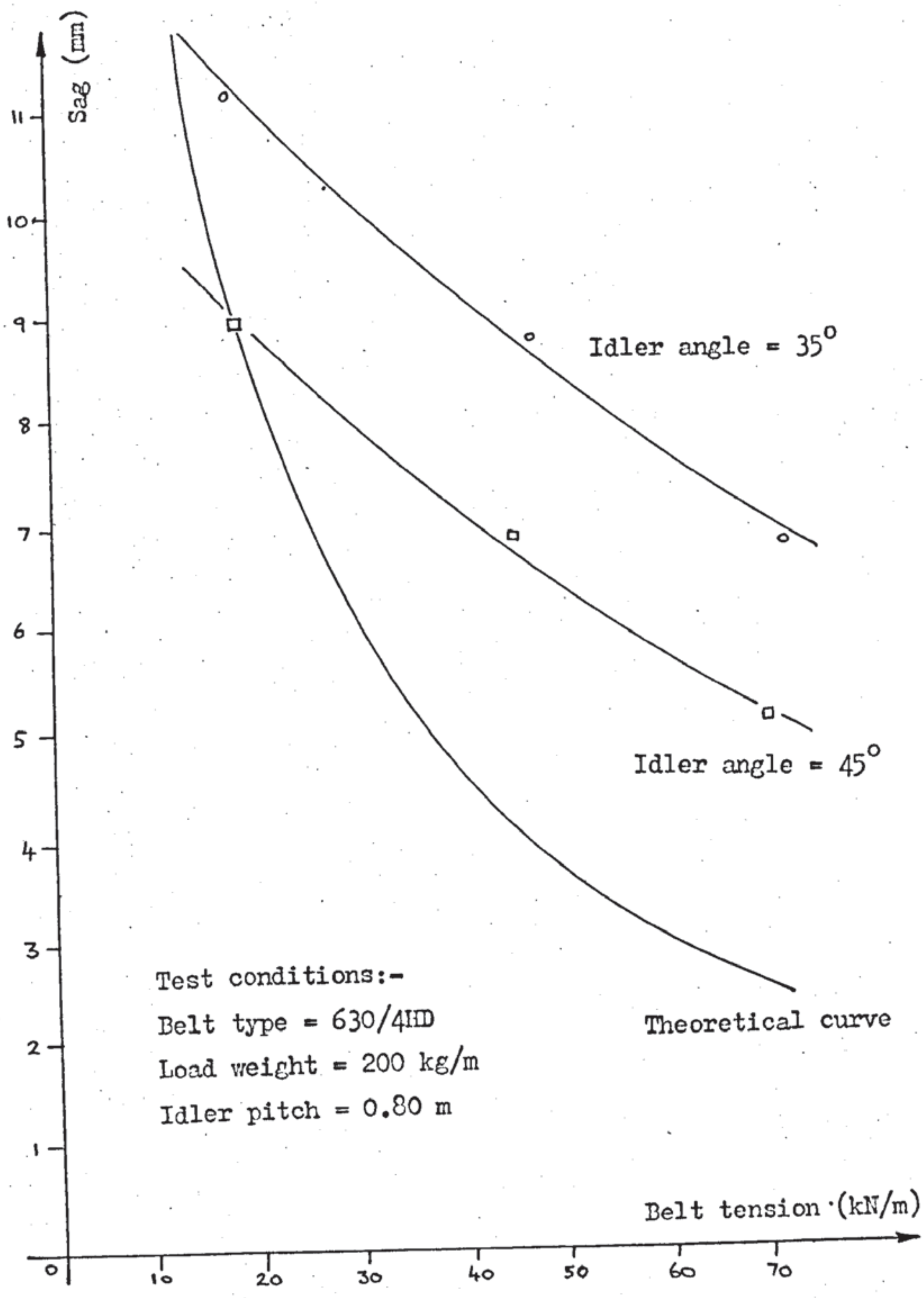


FIGURE 8.2.9

Variation of inter-idler belt sag with belt tension troughing angle would probably cause a significant change in the overall effective stiffness of the belt trough. In addition, as the material carried was fine and unconstrained it is likely that the load distribution across the belt was more even than in the present work. In the present work the load material was constrained in bags and the belt types tested had a relatively large transverse stiffness. Therefore, increasing the idler angle would cause no significant increase in trough stiffness,but would change the load distribution such that a greater proportion was acting over the central area of the belt and would thus increase the belt sag slightly. It can therefore be seen that the increased beam effect claimed for deep troughing conveyors does not decrease the amount of inter-idler sag under all conditions. This is in agreement with the statement made in the Engineering Manual of the B. F. Goodrich Company [98] :-

> "Deep troughing idlers can result in increased belt support between idlers. There is more "beam effect" between idlers which could conceivably permit wider idler spacing. This may be true with fine materials, but due to the concentrated or localised belt distortion with lumps, wider idler spacing is not generally recommended."

Figure 8.2.10 shows the variation of inter-idler belt sag with load weight for the three different belt constructions tested. It was found that no significant differences occurred between the behaviour patterns exhibited by the different belt types and although deviations from the theoretical curve were recorded,these were minor in the critical region approaching full-capacity loading.

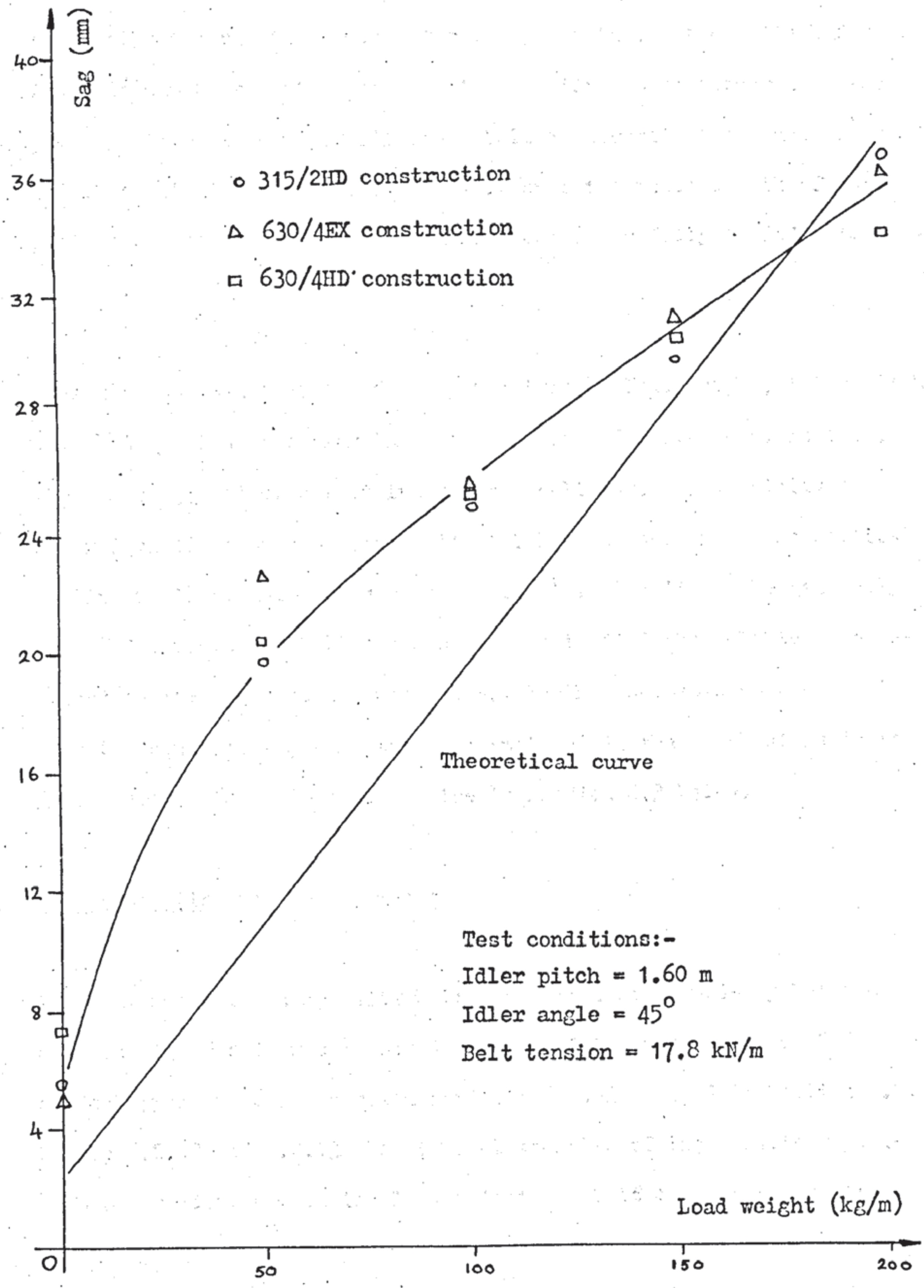


FIGURE 8.2.10

Variation of inter-idler belt sag with load weight

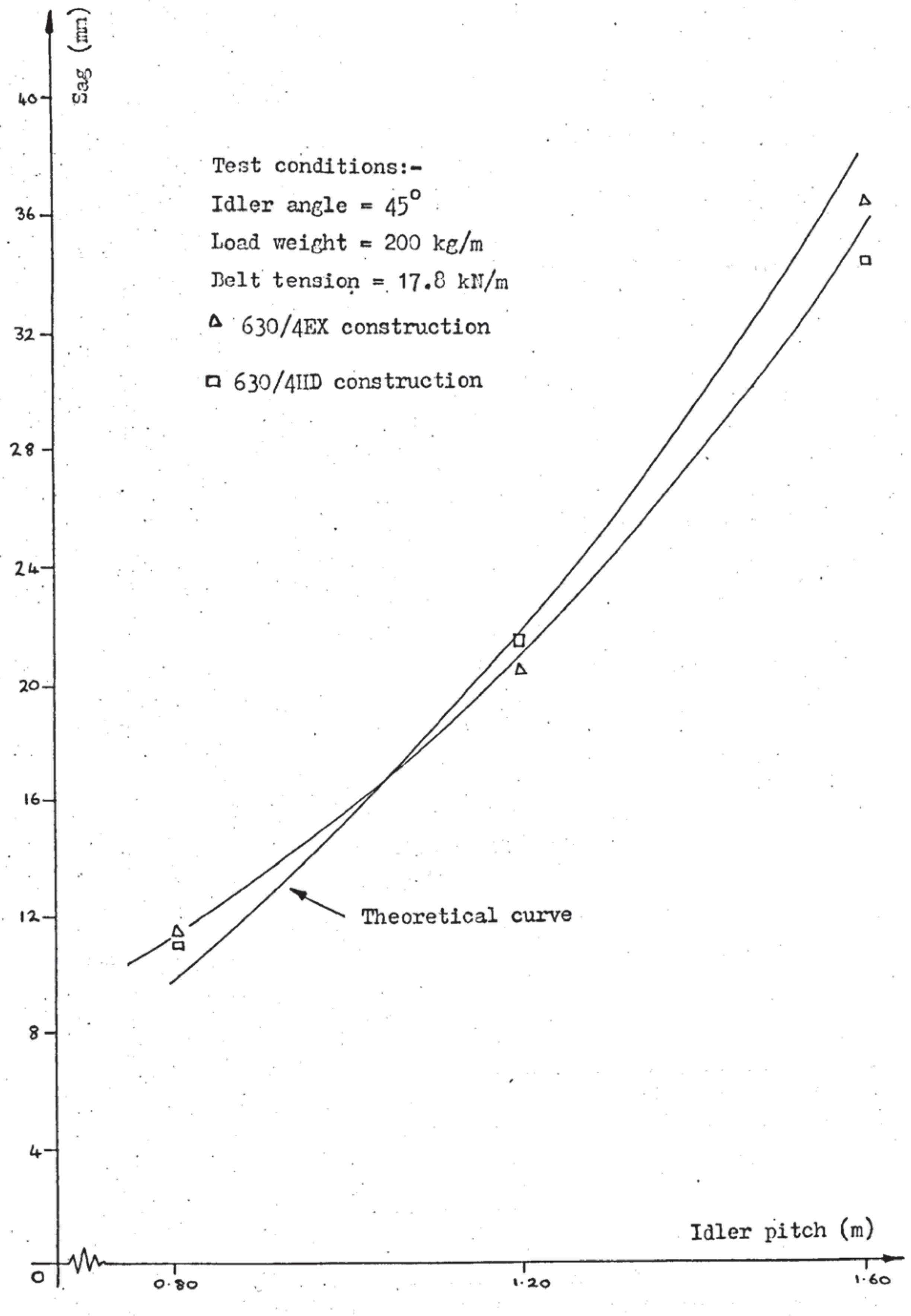
Figure 8.2.11 shows the variation of inter-idler belt sag with idler pitch (the one remaining parameter to appear in the theoretical formula). Again it can be seen that no significant differences occurred between the behaviour patterns of different belt constructions but now observed and theoretical curves agree very well over the whole length of curves plotted because the tests were carried out under fully-loaded belt conditions.

It was thus shown that the three parameters of load weight, idler pitch and belt tension appearing in the theoretical formula were, in fact, those having most effect on inter-idler belt sag. Only relatively minor deviations from the theoretical behaviour were noted at critical loading regions, and so for most practical applications the generally accepted formula for belt sag can be used to find the belt tension required to limit sag to a given value. Further comments on the results from the study of sag behaviour are included in the conclusions drawn from the experimental investigation in Section 8.3 below.

Inter-edge Distance Measurements

The distance between opposite belt edges at a point mid-way between adjacent idler sets was measured because of the phenomenon of "edge folds" appearing under certain conditions as described in Section 7.6. Figure 8.2.12 and 8.2.13 show typical examples of the behaviour patterns observed with changing load weight and belt tension respectively.

Figure 8.2.12 demonstrates that the inter-edge distance decreased as loading started but increased slightly at full load because of the load material physically pushing out the sides of the belt. It can



<u>FIGURE 8.2.11</u>

<u>Variation of inter-idler belt sag with idler pitch</u>

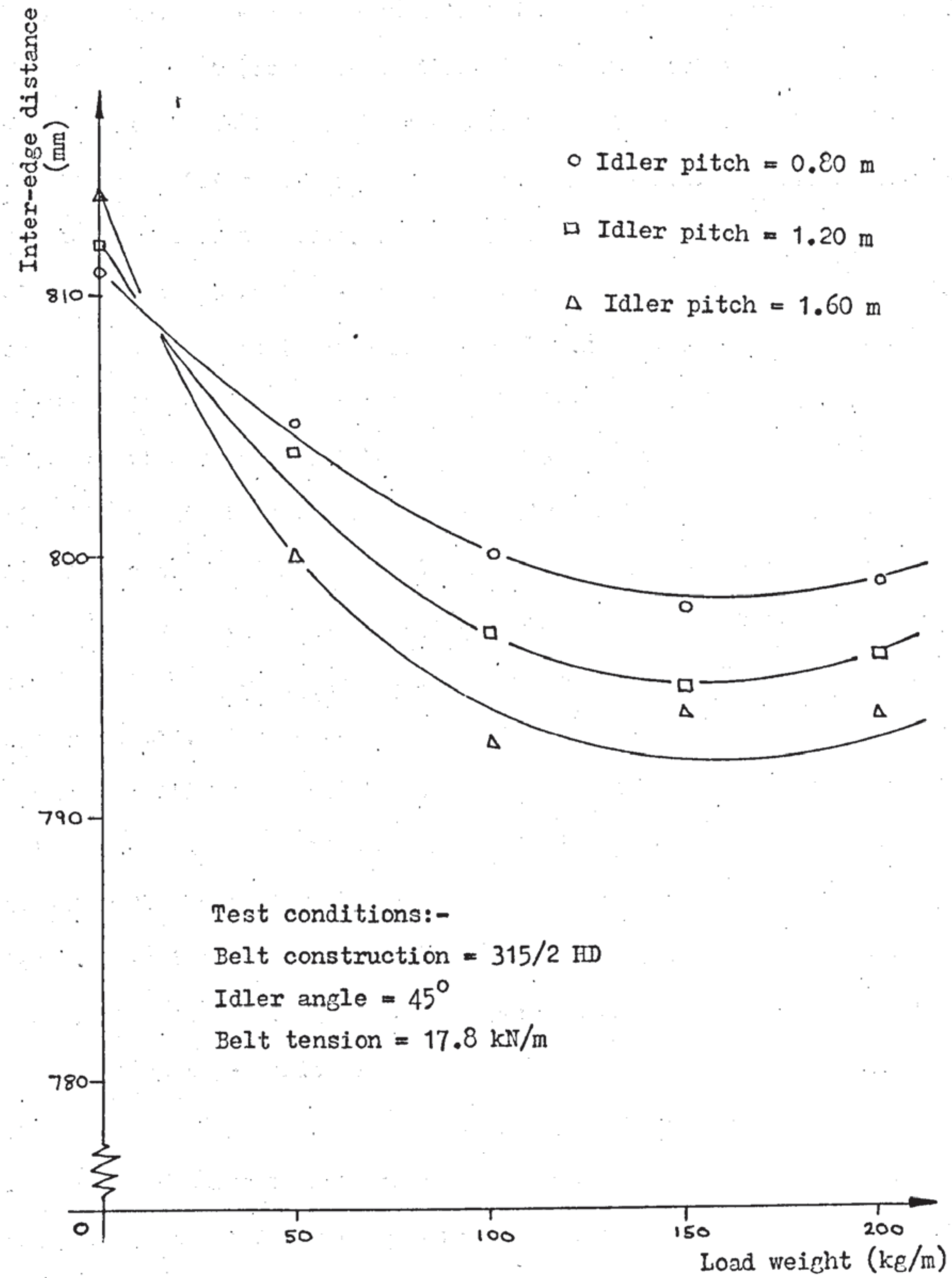


FIGURE 8.2.12

Variation of inter-edge distance mid-way between idler sets with load weight for three idler pitches be seen that the overall change in this parameter was in the order of only two percent (or sixteen millimetres; that is, each belt edge moved only eight millimetres). No significant variations in this parameter at different idler pitches were noted. Figure 8.2.13 shows that, under the conditions tested, the inter-edge distance increased slightly with tension but could be regarded as virtually constant. No behaviour patterns such as that shown in Figure 7.6.4 were observed, thus showing that no edge folds occurred during the test work. As described previously, it was not possible to record a belt tension of less than 17.8 kN/m, but it is likely that if this had been possible, edge folds would have appeared at a very low tension. It was thought that the probable shape of the inter-edge distance against belt tension over a complete tension range at all load conditions would be as shown in Figure 8.2.14. The test work described here demonstrated that,under normal conditions of belt loading on standard idler pitches,no edge folds appeared in the belt even at relatively low belt tensions, and it was thus shown, from the work described in this chapter and in Section 7.6, that the tension required from formula 2.2.10 to limit belt sag to the acceptable limit of two percent was always adequate to ensure that no edge folds occurred.

It was interesting to note that the behaviour of the inter-edge distance (both at the idler sets and mid-way between them) was consistent with anticlastic bending behaviour. Antiolastic bending occurs in any material sample of finite thickness when bending occurs. It is the phenomenon of bending also occurring in a direction at right angles to the original bending (see Figure 8.2.15), due to the different levels of strain that occur on the two sample surfaces when bent. During the test work it appeared that, as the belt was bent round the idler "rollers",

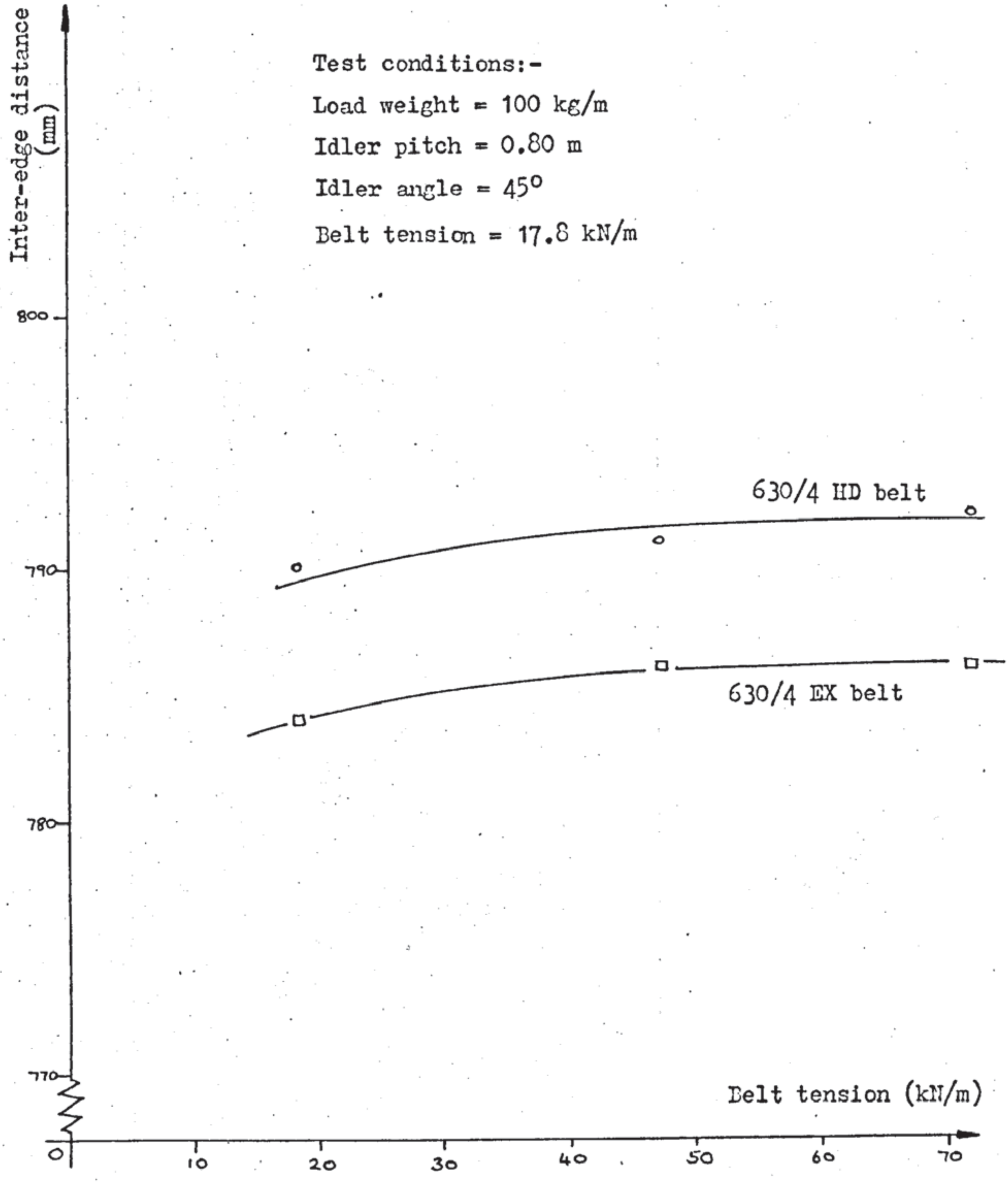


FIGURE 8.2.13

Variation of inter-edge distance mid-way between idler sets with belt tension for two belt constructions

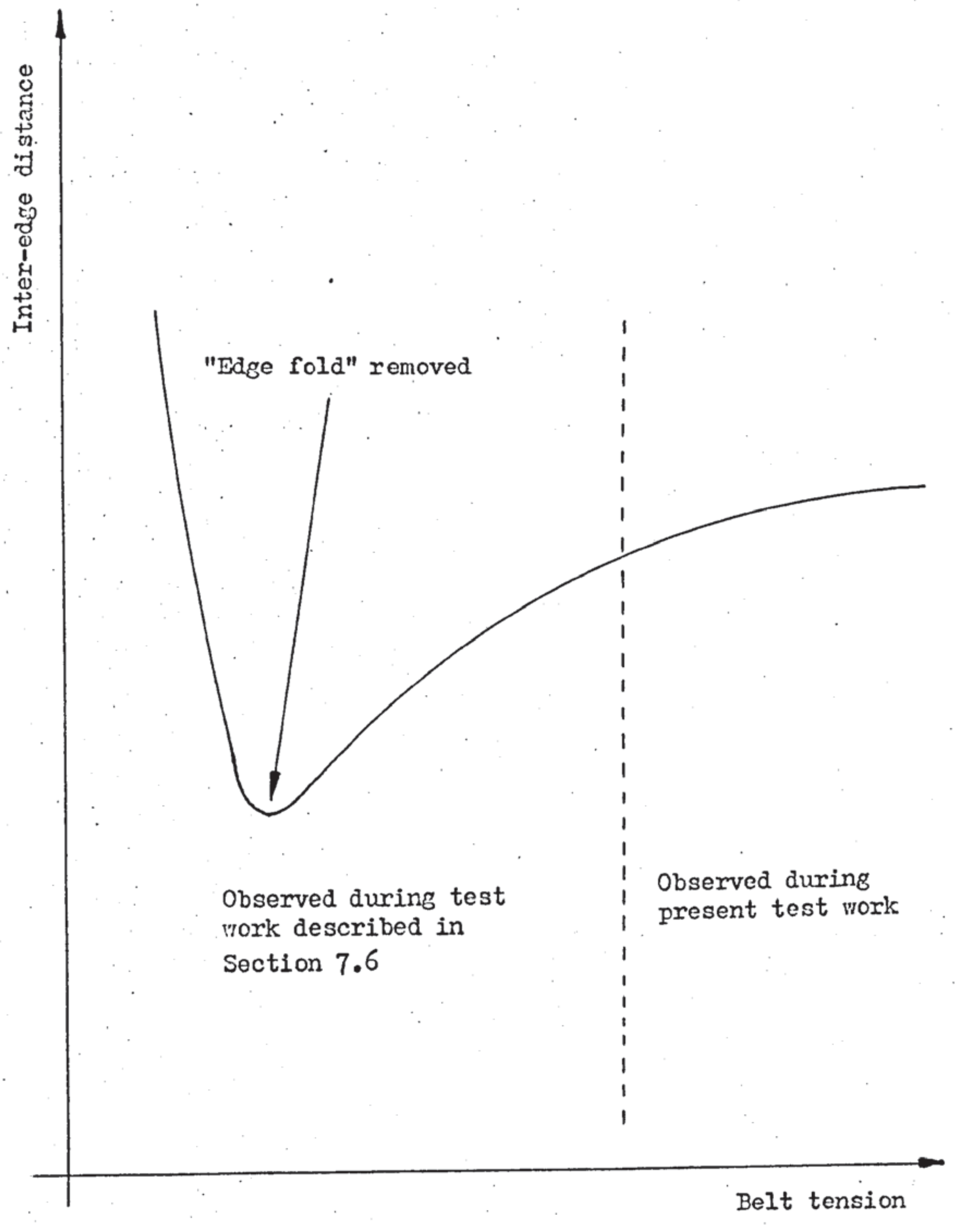


FIGURE 8.2.14

Probable overall behaviour pattern, at all load conditions, of inter-edge distance mid-way between idler sets

NOTE:- The behaviour at the low tension end of the curve was noted in the extreme conditions of high load and long idler pitch but was not noted during the work described in Chapter 8 because low enough tension values could not be recorded.

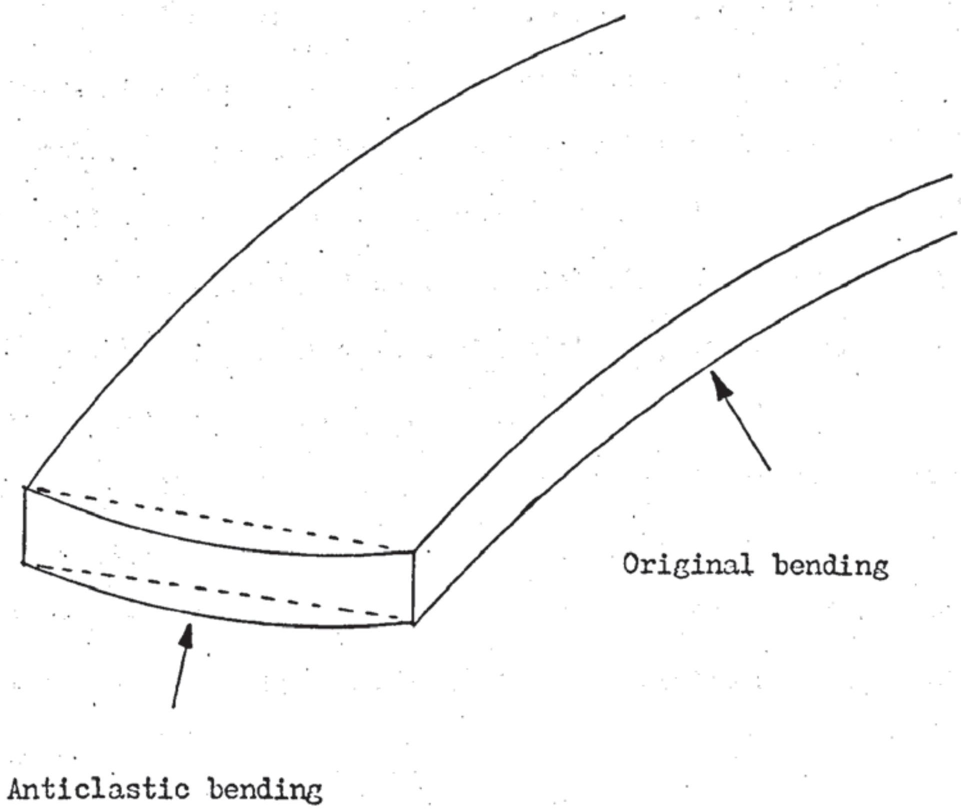


FIGURE 8.2.15

Anticlastic bending behaviour

its edges lifted slightly from the side "rollers." Thus, bending was occurring in the plane at right angles to the original bending and anticlastic behaviour was thought to be the cause. The degree of anticlastic curvature increases as the Poisson ratio of the bent material increases. (Poisson ratio being the ratio of percentage width decrease to percentage length increase when a material sample is extended. For metals the Poisson ratio is in the order of 0.3.) It was realised that if the observed behaviour of the belt edges was due to anticlastic curvature, the effect was much greater than that normally observed with metals. This could have been due to the fact that the belt shape, particularly at the idler sets, was constrained over most of its width by the load and idlers and so the anticlastic bending was compounded into only the outer belt areas. In addition, the fairly large anticlastic behaviour could have been due to a high Poisson ratio. Therefore, some small scale experiments were carried out to estimate the Poisson ratio of the belt samples used. These tests were carried out on a tensometer and consisted simply of measuring the decrease in width of a belt sample as it was extended. The results are shown in Table 8.2.2.

It can be seen that all the results were in the order of 0.5; the maximum possible value for Poisson ratio. This demonstrated that the observed behaviour was almost certainly due to anticlastic curvature. Thus, as the belt tension decreased or the load weight increased causing greater belt sag and hence greater bending of the belt around the idler, the belt edges should have moved further off the side "rollers" at the idler station. This was the observed behaviour pattern. As the degree of belt longitudinal bending is much greater at an idler set than at any other position along an idler pitch, it is the behav-

Belt Type	Poisson ratio		
	Sample 1	Sample 2	Average
315/2 HD	0.52	0.43	0.47
630/4 EX	0.48	0.50	0.49
630/4 HD	0.54	0.50	0.52

TABLE 8.2.2

Poisson ratio values for the belt types tested

Note:- The theoretical maximum value for Poisson ratio if no volume change is to take place when the sample is extended is 0.5. Three of the results quoted above are greater than 0.5, but the errors involved in the fairly crude experimental technique were large, and so the results show that for belting, the poisson ratio appraoches the maximum possible value, 0.5.

iour at this point that determines the distance of the belt edges from the idler side "roller." As the belt moves between idlers the belt edges will move outwards slightly but will still be restrained, especially at small idler pitches, by the belt shape at the idler set.

Having shown that this particular phenomenon of belt edge movement or "tubing" could be explained by anticlastic curvature, it was realised that the deformations occurring because of this were so small that the belt constructions at present in use were unlikely to be damaged by them. Therefore, it was decided that no practical use could result from a more elaborate study of this behaviour.

Belt Shape Measurements

Measurement of the belt shape was used, as has already been shown in Figure 8.2.1, to demonstrate that the cyclic changes in the shape of a belt as it moves through an idler pitch length are not severe enough to cause belt breakdown by flexing. The belt shape measurements could also have been used to study the validity of formulae to find the cross-sectional area of the belt trough and hence derive the geometric load carrying capacity of the conveyor. However, for the reasons given in Section 2.2, this was not thought to be a worthwhile exercise at present, but the basic data for this study is now available if it is required at a later date.

The importance of the length of non-contact between belt and idler centre roller and the role that belt stiffness plays in keeping this to an acceptable value was established during the experimental work. It was thought that, if a small-scale service related test was to be designed to measure the belt stiffness in such a way as to be useful for assessment of transverse load support capability of different belt constructions, it was important to study the shape of the belt in the idler gap so that it could be reproduced in the test. Typical results of the shape obtained are shown in Figure 8.2.16 and Figure 8.2.17 shows the holes through the idler used to measure this shape. Figure 8.2.18 is a photograph of the belt area near the idler gap and should be compared with that in Figure 7.6.11. The figures mentioned show that the level of belt deformation near the idler gap is not as high as that experienced by the sample in the Pantograph test. Therefore, it is thought that, although the Pantograph test would appear to be the most suitable belt stiffness test for the assessment of load support capability of those available at present, it does not really reproduce the conditions that occur in practice, and so another method must be devised. Further comments on this are included in the conclusions drawn from the work and discussed in Section 8.3 below.

8.3 Conclusions from the Experimental Investigation

The study described above which was carried out on a large conveyor test rig demonstrated several interesting features of belt behaviour. The most useful of these, and, in fact, the original objective of the work, was the insight gained into the mechanism of transverse load support. It was shown that the criterion by which load support should be assessed for the type of belt construction tested was the length of non-contact between roller surface and belt at each end of the idler set centre roller. Another criterion, that of avoiding "edge folds" by adequate belt tension, was shown to be met under all normal and

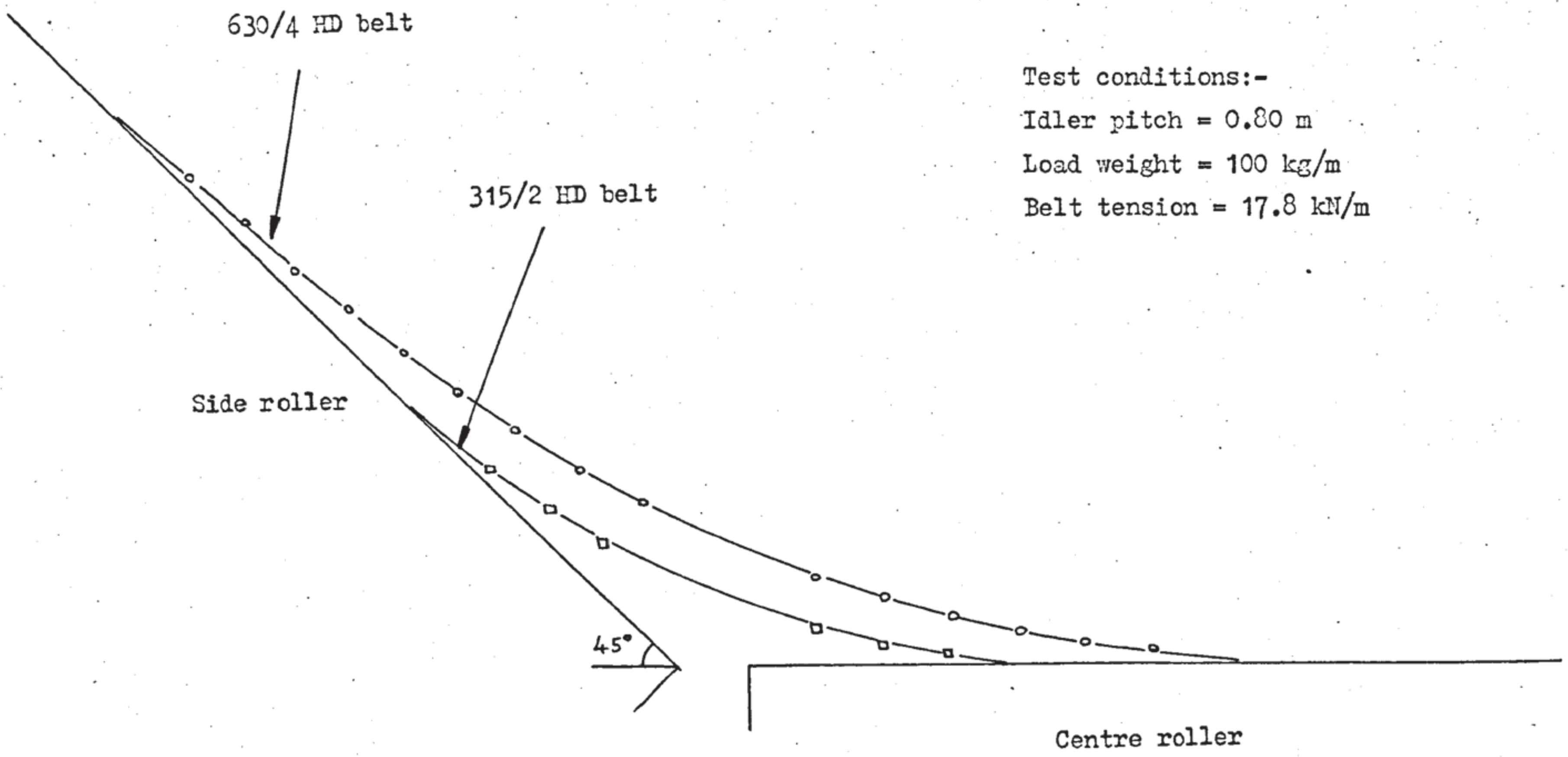


FIGURE 8.2.16

Measured belt shape at idler gap for two different belt constructions (Full scale)

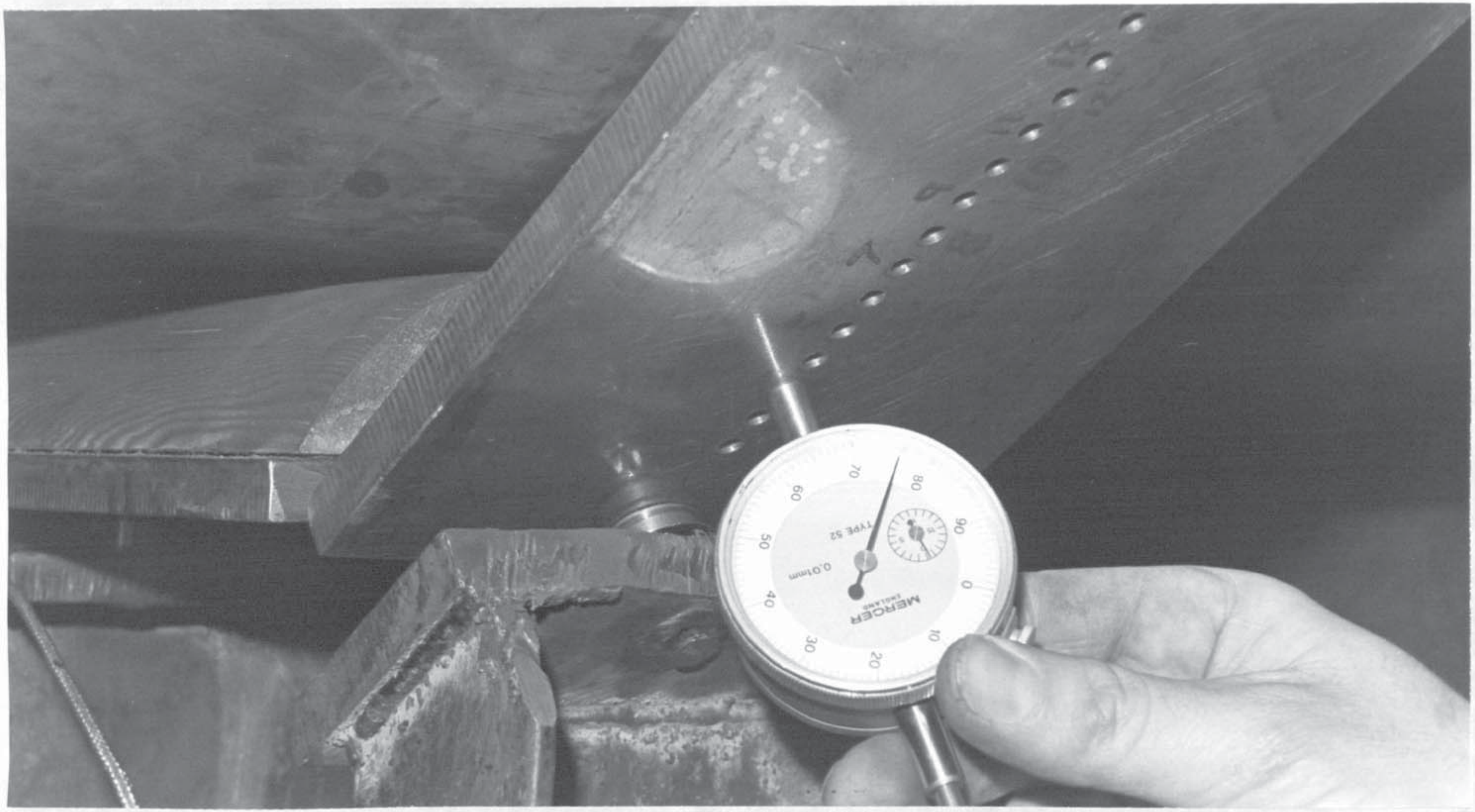


FIGURE 8.2.17

Measurement of the distance of the belt from the idler surface

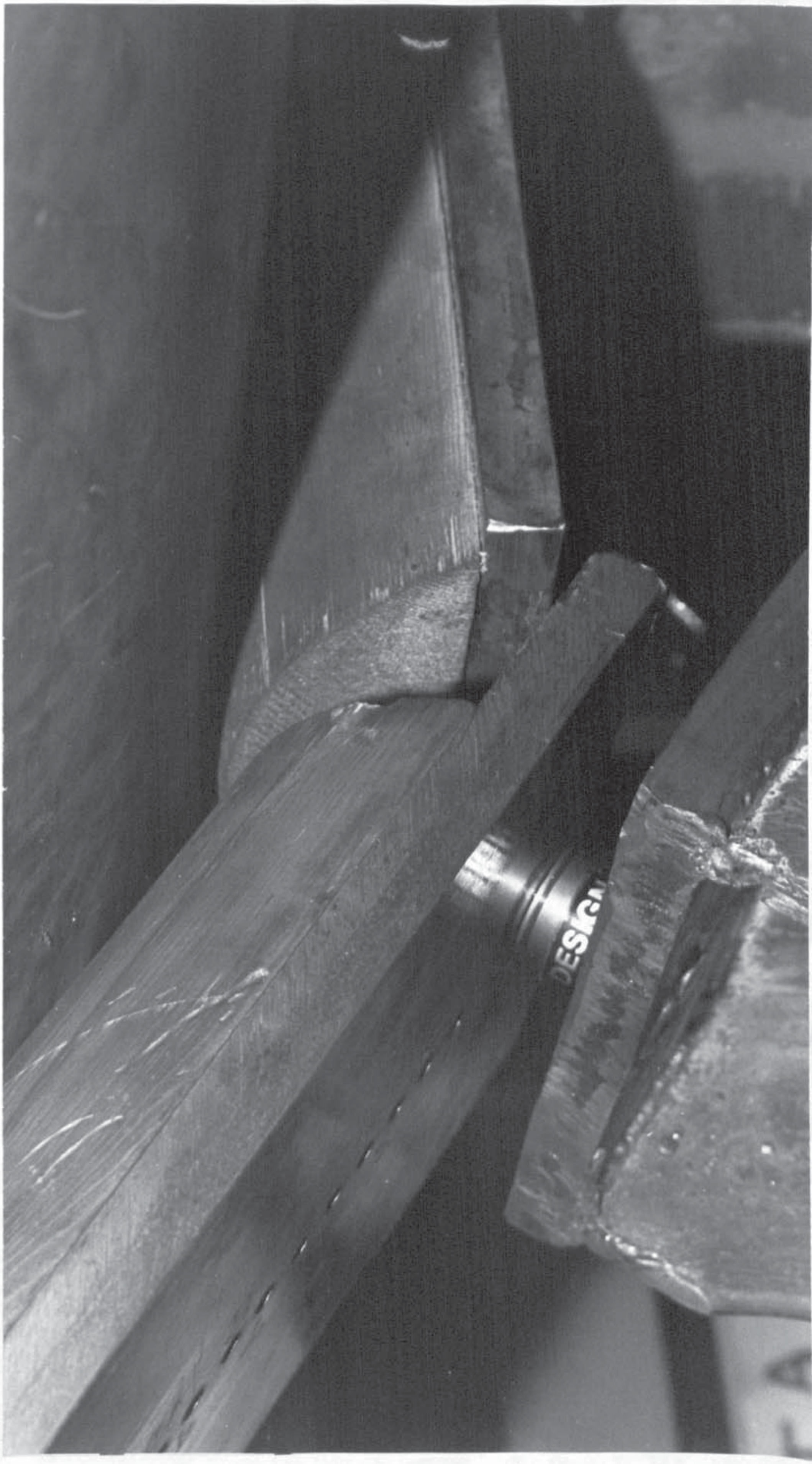


FIGURE 8.2.18

Belt type 630/4EX at idler gap area under fully loaded conditions some extreme conditions by ensuring that the tension required to limit belt sag to an acceptable limit was present. However, if abnormally long idler pitches and high load weights are being considered, this particular aspect should also be investigated.

The three most important parameters to consider when attempting to find the optimum combination of conveyor and belt properties to supply the required load support were shown to be:-

(1) Load weight per metre of belt, M (kg/m).

(2) Idler pitch, I (m).

(3) Belt transverse stiffness, EJ (kN mm^2).

The variation in belt/idler non-contact length can be seen in the figures presented in the previous section. Other parameters such as idler angle, load density and belt width appeared to have insignificant effect, but it must be remembered that varying these will after parameter (1). Other combinations of load density and belt width could be studied in the future to check that the statements made above are valid even in extreme situations. (For example, high density material being conveyed on very wide belts.) Parameter (1) is very easy to calculate, and, as can be seen in Section 3.3, is required for belt tension calculations anyway. It is dependent on two pieces of conveyor design information; the tonnage rate and the belt speed. Parameter (2), for existing conveyors, is a fixed quantity and parameters (1) and (2) can be combined to give a "load weight per idler set". Thus the determination of adequate load support for normal present-day conveying practice can effectively be reduced to finding a suitable belt stiffness for a given load weight per idler set. It is interesting to note that none of the major belt manufacturers mention idler pitch when tabulating their load support data (see Section 3.4).

Having demonstrated that belt stiffness and load weight per idler set are the major factors affecting load support, it is necessary to find permissible combinations of them. Therefore, an acceptable limit for the minimum length of non-contact between belt and centre roller must be decided. Obviously the belt does not actually fail until it reaches the very end of the centre roller, but a certain "safety factor" must be included to allow for off-centre loading, variations in the load weight on the conveyor and changes in the belt tension. After considering the results of the experimental work, it was decided that the minimum length of non-contact should be set at 10 mm as this would then allow for the factors mentioned. However, a minimum non-contact length generally accepted by belting engineers from service experience is in the order of 3 mm. The test work has shown that this would be acceptable only if it is observed at maximum possible loading conditions and cannot be accepted if measured under the conditions at which the conveyor is normally run. (That is, as most conveyors are designed to allow for surge loading, normal running conditions do not represent maximum possible loading.) It could, of course, be argued that surge loading should not be considered for this particular design aspect because, even if the belt did start to press into the idler gap under these circumstances, this condition should not last for long and would be unlikely to damage the belt considerably.

If the test results are studied, the load required to deform the belt constructions investigated to this minimum acceptable limit can be found. The results of such an exercise are given in Table 8.3.1. However, the value of these figures is rather limited because they have not been correlated with service. Having carried out a service-related investigation to establish belt behaviour patterns, it is now important to collect some service information and correlate the non-contact lengths measured during the work and the minimum acceptable value with those occurring in practice. For example, although the rotation of the idlers on a working conveyor will reduce the frictional forces along the side rollers occurring between the belt and side rollers and hence, perhaps, allow the belt to move towards the idler gap, it is possible that the inertial effect of the forward rotation of the rollers will force the belt to lift away more from the idler surface. This work should, ideally, be carried out on a specially designed conveyor rig, where loading conditions can be controlled. However, none is available at present although some suggestions about the equipment that should be incorporated on such a rig have been discussed at the beginning of Chapter 7.

Another aspect that must be studied is the changes occurring in belt stiffness with service life. Vulkan [87] has carried out some work on this and his results are reproduced in Table 8.3.2. The flexural rigidity was measured using the Pantograph Test but the dimensions of the apparatus and the belt samples are not given. It can be seen that, in general, the flexural rigidity of the belts tested decreased with service life and the major changes occurred within the first sixteen weeks of service. It is interesting to note that tests were made at two different temperatures by Vulkan. During the present work the

Belt construction	Approximate load weight/idler set (kg) to give belt/roller non-contact length of 10 mm.
315/2 HD	75
630/4 EX	180
630/4 HD	320

TABLE 8.3.1

Maximum permissible load weights
per idler set for the belt
constructions tested

Temperature	Direction	Belt	Flexural rigidity (kN mm^2)			
			Unused	16 weeks	32 weeks	64 weeks
80°C	Warp (Longitudinal)	A	31.6	24.4	25.3	25.5
		B	52.5	57.1	51.1	59.1
		C	110.8	115.4	-	-
		D	147.8	137.5	122.0	-
		E	41.3	40.5	38.7	46.2
		F	70.6	66.0	64.9	75.5
	Weft (Transverse)	A	45.1	40.5	36.2	31.6
		B	91.8	67.2	57.4	65.4
		C	139.2	103.6	-	-
		D	193.7	184.0	182.5	-
		E	41.3	18.1	21.5	21.2
		F	53.7	47.1	43.3	46.8
27°C	Warp (Longitudinal)	A	26.4	22.4	25.3	20.1
		B	37.3	33.6	29.0	29.3
		C	47.1	49.1	-	-
		D	68.3	63.7	66.3	-
		E	25.5	23.0	22.7	21.8
		F	29.3	23.0	24.4	23.5
	Weft (Transverse)	A	40.2	35.3	33.6	27.0
		B	58.3	44.2	37.9	38.5
		C	66.0	51.1	-	-
		D	120.0	96.4	107.9	-
		E	16.1	14.1	15.8	10.6
		F	25.5	21.2	20.7	17.2

TABLE 8.3.2

Changes in belt flexural rigidity during service recorded by Vulkan ambient temperature during the test work was recorded, but only temperature differences of four or five Centigrade degrees were noted and so no account was taken of this in the analysis of the results. Vulkan's work showed that, over fairly large temperature ranges, changes of approximately twenty percent can occur in the belt stiffness value. It is therefore important to take account of temperature if the belt is to be operating in conditions of extreme heat or cold.

Belt flexural rigidity changes with the degree of sample deformation, and so, if a bending test is to be related to service conditions, it should reproduce the type of deformation occurring in service. The Pantograph Test does not do this, but, as can be seen from Figure 8.2.8, it is sensitive enough to demonstrate the different transverse load support capabilities of various belt constructions. However, it is thought that a more suitable test method could be evolved and the transverse deformation shape that should be brought about in the test is that measured during the present test work and shown in Figure 8.2.16. A short review of different bending tests that have been, or could be, used on conveyor belting is included as Appendix 13 in order to help limit the choice for a small-scale service related technique. It is concluded in this that a "4-point bending test" should be adopted as a standard measure of inherent belt transverse load support capability or another technique causing bending in both longitudinal ("around the idler") and transverse ("to form a trough") directions could be devised specifically for the task.

If such a test was devised and a relation between it and the results of the present test work was established there would no longer be a need to carry out further time consuming and expensive tests on the large rig. Instead, the small scale test could be used directly to assess the load support capability of any other type of belt construction compared to those tested in this work. It must be emphasised that at present such a test would only be comparative and service information as described above would be required to establish the test as an absolute measure of load support capability. The need for such an absolute test has already been discussed and the present test work has established information that is essential to the development of such a standard test.

Now that the relation between belt stiffness and transverse load support has been empirically established, a bending test could be used in a development of any new form of belting to assess its load support capability. Initially, an absolute measure of load support is not required for this and a comparative test demonstrating a new belt's capabilities against that of the existing range would probably be adequate. (It could, in fact, be argued from the results shown in Figure 8.2.8 that the Pantograph Test, as used during the experimental investigation, would be suitable for this purpose.) A programme should now be initiated to find methods of building the necessary transverse stiffness into belt constructions without overspecification of other design properties. Attempts to do this have been carried out previously [110], but now a more rational basis for development work has been established.

The test work has demonstrated that the most rational method of choosing a belt construction for an existing conveyor from the point of view of load support would be to calculate the load weight occurring per idler set and then choose a belt construction having adequate transverse stiffness for the task. This suggests that for new conveyors a method of design to select the optimum combination of conveyor design and belt properties could be introduced. That is, the choice of idler pitch and belt construction should be made from a study of the costs of various suitable combinations of idler pitch and belt construction. Thus it might be found to be more economic, from the point of view of load support, to increase the number of idler sets in order to be able to use a cheaper belt. Several factors would have to be taken into consideration during such an investigation. These include:-

(a) The conveyor structure usually outlasts the belt and so belt replacement costs in the future must be considered.

(b) Fewer idler sets would mean less maintenance costs.

(c) Longer idler pitches can cause increased power consumption because of greater inter-idler sag [28].

It can be seen that every case would have to be judged individually and the advantages of using increased idler pitches with higher transverse stiffness belts would have to be weighed against their disadvantages. However, this discussion does demonstrate the possibility of devising a method of conveyor/belt selection which results in the optimum combination of parameters. There is, of course, a considerable amount of further work to be done before this can be fully achieved. For example, a study of the effect on power consumption of belt sag, idler angle and idler pitch should be carried out. The acceptability of the sag formula has been demonstrated during the experimental work, but what has not been established (and, in fact, cannot be without service information) is the validity of the 2% sag limit imposed in the tension calculations as described in Section 3.3. It might be found that a lower or higher value represents the optimum compromise between avoiding load spillage and reducing power costs.

However, even though there is still a great deal of work to be carried out before a perfect belt selection procedure can be devised, the present experimental work has, at least, fulfilled its original objectives within its original budget and has demonstrated the most rational method by which one particular section of the procedure should be carried out.

CHAPTER 9

OVERALL DISCUSSION OF FINDINGS AND CONCLUSIONS

The present work has involved a full study of all aspects of belt design and selection and has demonstrated that modern belt selection proceduresoften rely on rules based on experience with older types of belting such as those with all-cotton construction. The introduction of strong synthetic belt reinforcement has allowed belt conveyors of high capacity and virtually unlimited length to become technically feasible, but these, in turn, require a consideration of design parameters which previously were not important whilst other factors, formerly significant, no longer play a critical role.

The studies described in this thesis, relating to practical belt design and specification experience and an examination of the scientific literature, have shown that the provision of adequate load support is a significant factor in modern design and that satisfaction of this criterion by means of existing constructions often causes overspecification of other design properties. Accordingly, an experimental investigation into this property "load support" was set up to study the contribution to it from various conveyor parameters. The investigation, described in Chapters 7 and 8, has established that the length of non-contact between the belt and the centre roller of an idler set provides a criterion by which this property can be assessed rationally for the all-synthetic plied belt constructions at present in common use. In addition, the parameters having most influence on transverse load support have been shown to be the load weight per idler set and the belt transverse stiffness; the dependence of load support on these parameters has been demonstrated and a basis for a test method, relevant to service conditions, for measuring belt stiffness has been proposed. Once such a test has been established it could provide specification data for use in a published belt selection procedure and be used as an assessment method for a full study of the dependence of inherent belt transverse load support on belt parameters such as amount of interply rubber. This could result in a more efficient method of providing inherent load support without having to make any radical changes in belt construction techniques. Further experimental studies to include other belt constructions and different belt widths on the large test rig could be carried out along the lines of the present investigation with a view to establishing a code of practice for specifying inherent load support capability, but this would be of limited value because a more urgent need, at present, is for the results already obtained to be correlated with service. Field studies are required for this work, which should include investigations into the applicability of the minimum belt/centre roller non-contact length of ten millimetres suggested as acceptable in the present work and the changes in belt stiffness during service. These studies could be carried out as part of an organised service information collection system and would result in a data base from which optimum combinations, from the point of view of load support, of idler pitch and belt stiffness could be found.

It has been shown at various stages throughout the thesis that some belt and conveyor design parameters have been "optimised" without considering the resultant effect on other design factors. This situation is not desirable because it can lead to those design properties being given more attention than they merit at the expense of others, and so the effect of optimum load support conditions on such items as power consumption, idler bearing life and belt life should also be considered. The final objective is therefore an optimum design of the whole conveying system and so a knowledge of the conditions which represent the overall most efficient combinations of conveyor and belt parameters is required. The total cost, including depreciation of conveyor machinery, per tonne of material carried by a conveyor during one belt's life is the absolute measure of that particular belt/conveyor combination's efficiency and performance. If such figures were derived for a great many working conveyors, it should, eventually, be possible to form belt selection procedures with a more rational basis - that of end-use selection and total cost to a customer over a period of time: that is, end-use selections rules would be used to find several belt constructions suitable for the task and then the customer would be able to choose that which came nearest to the economic constraints of the situation; he might, for example, decide to choose a more expensive belt in order that repair costs are kept to a minimum and replacement is unlikely for some time. There will always be some uncertainty in predictions of belt life and running costs because of the possibility of random damage. However, as some belt constructions will obviously have better resistance to random damage than others, test methods should be derived to assess their relative resistance so that the "random damage factor" could be taken into account in any cost predictions.

The situation discussed above should be the long term objective of belting industry, and, because of the vast amounts of service data necessary and the fact that it requires knowledge of both conveyor machinery and belt design, it is possible that it could only be attained by several belt manufacturers working co-operatively through organisations such as the Mechanical Handling Engineers' Association or the British Rubber Manufacturers' Association. In order to obtain this objective efficiently, the current literature on belting studies must always be reviewed so that no work is reproduced needlessly. A comprehensive review of literature published up to the time of the present work has been included as part of this thesis, and this should continually be updated. In addition, an efficient system for recording and analysing service data is required and that suggested in Appendix 6 would serve as a basis, for development, of the information that should be collected. It is likely that some form of computer storage facility would be necessary to deal with the vast amount of data generated and, as the work described in Chapter 5 has shown that the use of existing belt selection procedures could be simplified by programming, a computer system incorporating belt selection and the storage of conveyor specification and performance data could be devised.

Although reaching this final objective of more rational belt selection on an economic basis must be regarded as a very long term result of a systematic data collection service, there are several other factors, apart from the correlation of load support testing mentioned above, that could profitably be investigated in the short term by using the data. Examples of these have been mentioned at various stages in the thesis and include:-

(a) Some studies of belts in service could be used to establish more rational empirical design rules for the relatively minor design features of edge margin and permissible maximum lump size.

(b) Working belt horsepower requirements could be measured to check the validity of the various friction factors used in the tension calculations for modern belting. The variation, if any, of power consumption per tonne of material carried with changing conveyor parameters such as belt speed and idler angle could be studied.

(c) Some work on the co-efficients of friction between belt and drive drum could be carried out to establish drive factor values to be used under various working conditions.

(d) The relevance of the 10:1 factor for the ratio of belt tensile strength to permissible maximum working tension could be investigated.

(e) The accepted maximum amount of inter-idler belt sag is two percent of the idler pitch in the Dunlop belt selection procedure. As German test work, described in Chapter 2, has demonstrated that sag can affect power consumption, a study could be carried out to find the optimum value that will avoid material spillage and minimum power requirements.

(f) Correlation with belt performance of any tests used now or developed in the future and thought to be service related, such as impact resistance testing machines, could be attempted.

Point (e) in the above list relates to one stage that would be required to establish the overall optimum combination of belt and conveyor properties. However, the work described in Sections 6.1 and 6.2 has shown that limiting sag is unlikely to cause as much increased cost to the belt user as providing transverse load support, and so greater importance should be attached to establishing optimum conditions for load support than for sag limitation.

The need for the type of work described under (f) above has been emphasised throughout the thesis, and a series of tests with proven relation to service would be of great value in the development of any new belt constructions. (The ability of past and present belt constructions to withstand various damaging factors should be assessed from such tests so that acceptable standards for the results from new constructions could be derived.) The original experimental work described in the present work has given the basis for a small-scale service related test for the important property of load support and demonstrates the method by which such tests could be evolved: that is, large scale test work correlated with service measurements or controlled field trials should be used to establish the factors affecting the property in question, and then a small scale test should be devised to reproduce service conditions as nearly as possible. For many properties, such as fatigue life, fastener holding and splice efficiency, it is likely that dynamic tests would be required and the test methods used would have to accelerate the damaging factors that occur in service in order to obtain results within a reasonable time. In such cases, it must be ensured that the "acceleration" does not bring other factors which could affect the results into play: for example, if a short length of belting was run very quickly around some small diameter pulleys in order to accelerate flex fatigue, it is possible that heat build-up, a factor not likely to be too critical in most service applications, could cause the belt's failure rather than the property which is being studied.

It is not only a greater understanding of the technical aspects of belt behaviour and an improvement in the methods by which design parameters are assessed that would result from a systematic data collection service: Section 6.3 has demonstrated that the development of a standard product range more in line with present requirements would be facilitated by a greater knowledge of the conditions which are met by belts operating within various market segments.

Thus, at present, the needs of the conveyor belting industry are not so much for pure scientific research involving studies such as comprehensive investigations of the stress/strain patterns occurring in a belt (which, although interesting, would have limited value in belt design rules), but for a controlled, logical, "problem-solution" approach to the development of belt design and selection. This could lead to a selection procedure which is (a) more simplified and easy to use in all cases; (b) relevant to the needs of those industries requiring efficient bulk material transfer; (c) based on belt design data assessed from a service related test method for each property influencing performance; and (d) reviewed continually in the light of any developments made in belt or conveyor technology. Comparing this selection procedure with that described at the beginning of Chapter 4 demonstrates that if it could be achieved, the "ideal" situation would then have been reached.

APPENDICES

APPENDIX 1

Details of the industrial situation in which the present research work was carried out

The Technical Manager of Dunlop-Angus Belting Group (D.-ABG), Speke, Liverpool, provided the industrial supervision for the research project. D-ABG is responsible for the technical development, engineering expertise and marketing information of several Dunlop Divisions dealing with belting or belting-like materials. The major Division in terms of labour force and production capacity is the Speke based Dunlop Belting Division (DBD) which produces and sells conveyor belting mainly for handling bulk materials. The standard product range consisting of all-synthetic fabric belts is called the "Starflex" range. A technical department within DBD is responsible for quality control and tests belts both on the production line and in the laboratory to ensure that a consistent product is being manufactured but no research or development work is carried out by this department. DBD also have a team of belting engineers who collect conveyor installation details for belt selection purposes, carry out the belt selection procedure, make recommendations to customers about correct belt running practice and deal with any minor complaints about belt performance. If, however, the complaint proves to be due to some fundamental fault in belt design such as inadequate load support, the complaint is referred to D-ABG for test work to be carried out. (One example of such work is described in Chapter 7.)

The facilities available to D-ABG for research and development consist of a laboratory for production and evaluation of rubber compounds and a test house containing machines for dynamic and static testing of belt properties thought to relate to service performance. The resources of the Dunlop Central Research Department in Birmingham are also available to D-ABG Development Section if any specialist machinery not installed at Speke is required. There is also close technical liaison between D-ABG and several other Dunlop belting companies operating in other countries.

APPENDIX 2

A Short History of Belt Conveyor Development

According to Hetzel and Albright [30] the earliest reference to belt conveyors was by O. Evans in his "Miller's Guide" published in Philadelphia, U. S. A. in 1795. The conveyor was described as "a broad endless strap of thin pliant leather or canvas revolving over two pulleys in a case or trough". At that time, screw conveyors were commonly used to convey grain and other light materials; belts were only used for the transmission of power. Early experimental work was almost all concerned with this aspect of belting and not with conveying but the experience and results obtained during this early work were relevant to both applications. Morin probably became the first serious researcher into the properties of belting to document any work when he published his findings on the frictional properties of leather belts in 1834. In 1858 a patent for belting made up of several plies was taken out by S. T. Parmalee [111], one of the founders of the North British Rubber Company.

Over the next few years most of the basic pieces of belt conveyor equipment were designed, and work was carried out to establish design criteria such as power requirements. In 1863 O. C. Dodge was granted a U. S. patent for the "tripper*" - a device allowing the load to be discharged at any point along the conveyor. In 1865 the first conveyor belt in Great Britain was used by P. G. B. Westmacott and G. F. Lyster, engineers for the Birkenhead and Waterloo Docks at Liverpool.

* See glossary of terms at the beginning of this thesis.

They carried out experiments with a 12" wide belt and showed that it was capable of carrying grain with much less power consumption than a screw conveyor. In 1866 they installed a more sophisticated system using 18" wide belts running at up to 500 feet per minute. Westmacott was granted a patent in 1866 for a moving tripper, a device showing great improvements on that invented by Dodge. In 1868 Lyster described his work to the British Engineers' Society [112] and in 1869 Westmacott presented a paper to the Institute of Mechanical Engineers. Lyster's further work included the introduction of spool-shaped wooden idlers to increase the carrying capacity of the conveyor but these were discarded in favour of straight faced idlers because the higher linear speeds at the edge of the spool compared with the speed at its centre caused severe damage to the belt. Spools also made it very difficult to run the belt centrally. [33]. He then experimented with two roll troughing idlers and found it impossible to prevent the belt from centering on one roll or the other.

By this time much experimental and theoretical research work had been carried out, again mainly on power transmission belts, to study the efficiency of belt drive units. Investigations by Towne in 1868 and by Leloutre in 1878 made it possible to adopt satisfactory values for the permissible ratio between the tensions in the tight side (T1) and the slack side (T2) of a working belt, and thus to design belt drives in a more rational way. Osborne Reynolds advanced knowledge of the mechanics of belt action when, in 1874, he showed that there must be a change of speed with an elastic driving belt when the tension in it alters, and that this change in speed must increase as the tension difference becomes greater. (An effect verified experimentally by Lewis and Lanza in 1886.)

On the conveyor engineering side, the work of Lyster and Westmacott was taken up initially by W. B. Reaney. He introduced a grain belt conveyor in Baltimore, U. S. A. in 1873. This belt, being 30" wide, was much wider than those used previously. The belt was made up of four plies of fabric bonded together by rubber and it was run at 550 feet per minute. Webster continued experiments with grain handling belts, and for a time the Webster Manufacturing Company was the leader in the field. By 1885 belts were being used to handle heavier and more dense loads such as ores on an experimental basis,but it was not really until 1892 that belts successfully entered this field when the troughing of conveyor belts was introduced by Thomas Robins Jnr. to increase substantially their carrying capacity. His first attempt to achieve this and to eliminate the difficulty inherent in the two roller troughing idler system was to add a horizontal pulley just back from two inclined wing idlers. He followed this with the three-in-line unit idler which is commonly in use today and prevented oil from the idlers leaking onto the belt by an elaborate bearing system [112]. Initially the wing idlers were inclined at 45° to the horizontal, but it was noticed, with the cotton belts then in use, that longitudinal splitting of the belt occurred at the gap between the idler rollers. The connection between the steep troughing angle and the longitudinal splitting was quite apparent [30] and so the idler gap was reduced to as small a value as possible and the idler angle was decreased from 45° to 35°. Later, troughing idlers were reduced still further to 20°.

Robins not only considered the mechanical aspects of a conveyor, but also made a significant contribution to the developments of the conveyor belt itself. His first design of belts consisted of two or more fabric plies with a covering of rubber. He then reduced the thickness of the bottom cover and increased the rubber on the carrying side in order to secure longer life. About 1896, because of the then common practice of partial loading, he originated the stepped ply belt in which the plies were stepped off towards the belt centre so that there was an effective increase in cover thickness in the central zone. (This type of construction is rarely used now because of the high costs caused by not being able to produce several belts from one slab.)

By this time, Robins, Westmacott and Lyster had, between them, developed the major features of the present day belt conveyor. There have obviously been improvements in the equipment used, such as larger capacity drives and the introduction of sealed bearings for the idlers, but essentially the modern belt conveyor is the same as that of the late nineteenth century. (That is, a troughed fabric reinforced rubber belt moving over idler sets consisting of three rollers.)

The major developments that have occurred in recent years have nearly all been concerned with changes in the construction of the belt itself and the associated belt manufacturing methods. These changes have meant that longer and longer conveyors carrying heavier and heavier loads have been built. This can be seen when the first underground belt conveyor installed by Richard Sutcliffe in Glasshoughton Colliery in 1906 is compared with one recently installed at Silverdale Colliery. Sutcliffe's conveyor was 20" wide and 100 m long. The Silverdale belt was 763 m long and other underground conveyor systems now transport mineral over distances of 8 km to 13 km without any transfer points [14].

This increase in the size of belt conveyors was made possible by the introduction of synthetic fabric carcase reinforcement. Twenty to thirty years ago, the strongest belts were multi-layered sandwiches of rubber and 32 oz cotton duck with rubber edge added to prevent the ingress of moisture which would lead to degradation of the cover during service (see Figure A2.1a). Up to 1955 99.9% of all Dunlop conveyor belting was manufactured from cotton duck [100] and it was common for seven or eight plies to be used to reinforce a heavy duty belt with up to fifteen plies sometimes being used. This led to thicker and thicker belts with increasing lateral stiffness which gave rise to troughability problems and the need for larger and larger pulley diameters because of the lack of longitudinal flexibility. Attempts to reduce the lateral stiffness included the introduction of cotton tyre cord having warp (longitudinal) strength but virtually no weft (transverse) members [113]. The situation was further exacerbated as conveyor users and manufacturers again tried to increase capacities and reduce spillage by increasing the wing idler angle. (A 53° angle was used at Billingham by I. C. I.)

In September, 1950, the Cresswell Colliery fire disaster occurred and an investigation showed that it was due to the ignition of conveyor belting. A development programme, set up to produce a fire resistant belt, showed up many facets of belt behaviour (see Chapter 2) and the work led to the production of P. V. C. (fire resistant) belting. Impact testing machines clearly demonstrated that if two belts of identical construction, one of rubber and the other of P. V. C., were subjected to the same impact forces, then the strength retained by the rubber belt was always higher than the fire resistant type. This was due to the rubber cover absorbing more energy than P. V. C., thus causing the carcase of the rubber belt to be subjected to blow of lower intensity [114].

Changes in the carcase construction were needed to overcome this effect; one of the first was the introduction of a nylon weft in a standard 32 oz cotton fabric. This improved the resilience of the carcase and increased the lateral flexibility of the belt, thus allowing the increasing use of deep troughing conveyors with side idlers at 45° or more. The next step was to double the synthetic fibre for both warp and weft. The inclusion of cotton was necessary to ensure adequate adhesion between fabric and compound because the bonding was due to physical entanglement of the fibrous cotton with the rubber, and it was not possible to reproduce these adhesion levels with a chemical bond between nylon and rubber. The rapid advances which were made to overcome this problem can be seen from the following:- in 1966 [115] it was still essential to bulk nylon with other fibres, but less than two years later virtues of all-synthetic belts could be expounded [116]. This had been made possible by the development of chemical bonding techniques. The main advantages of all-synthetic belts are their high tensile strength, their high longitudinal and lateral flexibility and their inherent resistance to bacterial and fungal attack. This has meant that, as well as requiring fewer fabric plies, modern belt constructions need no edge strips to protect the carcase from moisture. (See Figure A2.1b.) This has also meant more efficient manufacturing methods; it is no longer necessary to make each belt individually, instead, for example, a slab of all synthetic belt can be made 1800 mm wide which can then be cut into two or more narrower belts as desired.

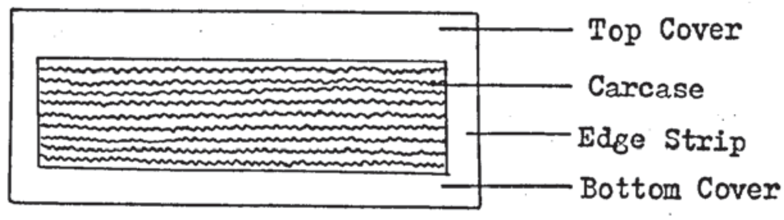


FIGURE A 2.1a

Typical construction of all-cotton belts

Note closely packed fabric plies and edge strip to prevent ingress of moisture.

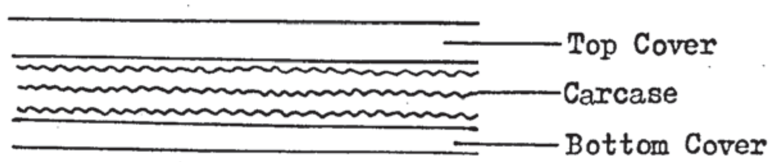


FIGURE A 2.1b

Typical construction of all-synthetic belt

Note low number of plies separated by interply rubber and lack of edge strip.

Thus, the development of belt conveying has led to the present situation of all-synthetic carcase belts being used on deep troughing installations. (The term deep troughing is generally regarded as applying to idler angles equal to or greater than 45°.) The actual choice of reinforcing fabric varies from place to place: all nylon carcasses are used extensively in the United Kingdom; the mainland of Europe favours fabric having a polyester warp and a nylon weft (steel cord belting is also very popular); rayon/nylon is favoured in the United States; polycarbonate yarns in Japan; Australia prefers carcasses made of a polyvinyl alcohol fibre Kuralon [117].

There is always development work being carried out to increase the variety of tasks for which a conveyor belt could be suitable. Thus, various different cover compounds demonstrating such properties as oil resistance, heat resistance or suitability for food transportation are available.

Few changes have occurred to the basic carcase construction comprising several plies of fabric, although solid woven carcases [118] and steel cord reinforced belting [119] are now manufactured. Probably the major recent development of the actual belt is the introduction of passenger conveyor belts having, of course, high lateral stiffness [120].

APPENDIX 3

Bibliography and sources of conveyor belt information

A major problem when the present work was started was the lack of readily-available published information on conveyor belting. An attempt was therefore made to collect as many details as possible of papers written on conveyor belt technology. After having approached the major belt and conveyor equipment manufacturers throughout the world it was obvious that none of them continually reviewed abstract lists and indexes to obtain details of articles written about their products.

Other non-manufacturing organisations such as the Plastics and Rubber Institute, the Rubber and Plastics Research Association, the Malaysian Rubber Producers' Research Association, the American Chemical Society Division of Rubber Chemistry and the National Coal Board were therefore approached to find out if they could provide more useful information. The American Chemical Society and the National Coal Board Library and Information Service both supplied reading lists which were useful but were somewhat out of date. The National Coal Board reading list mainly consisted of articles containing straightforward descriptions of conveyor installations whilst that of the American Chemical Society provided details of patents concerning belt manufacture. Both of these organisations pointed out that there were no plants to up-date their reading lists.

As the reading lists mentioned did not form an adequate or up-to-date review for the diverse nature of the present project, a full literature search on all aspects of belt design and selection was carried out. This involved approaching academic institutions which had been involved in belt studies; these included the Pennsylvania State University, the University of Wisconsin, the Department of Mining and Mineral Technology of Imperial College, London and the Hannover Technical High School - the institution which proved to be mose useful. In addition to this, abstract lists were studied; Engineering Index, Rubber and Plastics Research Association Abstracts and British Technology Index providing most articles. The result of this was that many papers and articles were reviewed and categorised according to their subject matter. (This had only been done in very vague terms in the reading lists mentioned above. For example, the subject headings of the National Coal Board List were (1) Conveyor belts and Installations, (2) Conveyor belt control and (3) Conveyor belt accessories.) Most of these articles have been referred to in the thesis text, but, in order to provide a comprehensive bibliography on conveyor belts, a full list of those reviewed is presented here. Each article is numbered and the articles giving information on a particular subject are listed. It should be noted that this bibliography only contains papers and articles which have been reviewed fully. There is still a great deal of useful literature published in German which could be translated and incorporated into the bibliography; any such literature encountered by the present author is listed at the end of this appendix.

List of Articles and Papers

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<u>Subjects covered by the above articles and papers</u>

1. General conveyor belt information:-

   4, 8, 11, 16, 19, 20, 21, 32, 33, 38, 39, 44, 50, 51, 52, 53, 63, 67, 69, 72, 73, 74, 97, 106, 109, 116.

2. History of conveyor belting:-

   8, 19, 33, 38, 50, 51, 52, 72, 73, 106, 116, 118.

3. Manufacture of conveyor belting:-

   9, 25, 52, 53, 71, 106, 112.

4. Economics of conveyor belting:-

   7, 8, 19, 30, 35, 39, 44, 45, 57, 63, 69, 70, 74, 88, 98, 116.

5. Belt and belt conveyor design and selection:-

2, 3, 5, 6, 7, 9, 13, 14, 15,
17, 18, 19, 20, 21, 22, 23,
26, 27, 30, 32, 33, 34, 35,
36, 37, 39, 41, 42, 44, 49,
50, 52, 55, 61, 63, 65, 67,
68, 69, 70, 71, 74, 78, 81,
82, 84, 85, 87, 90, 93, 96,
97, 98, 109, 114, 116, 117.

6. Rubber compounds used in conveyor belts:-

13, 15, 33, 38, 50, 51, 52,
71, 77, 106, 109, 110, 111,
118, 119, 120.

7. Fabrics used in conveyor belts:-

19, 21, 30, 32, 33, 38, 39,
50, 52, 63, 71, 72, 79, 83,
90, 99, 100, 101, 106, 107,
109, 112.

7a. The changeover from cotton to synthetic fabrics:-

15, 16, 19, 21, 30, 38, 39, 43, 50, 51, 63, 71, 72, 98, 100, 101, 106, 108, 109.

8. Conveyor belt constructions and properties:-

1, 3, 13, 16, 19, 21, 25, 26, 27, 30, 32, 33, 37, 39, 40, 44, 50, 51, 52, 53, 55, 71, 72, 79, 87, 88, 89, 92, 97, 101, 108, 109, 115.

9. Conveyor equipment:-

3, 4, 7, 8, 20, 21, 22, 31, 32, 33, 34, 35, 44, 50, 52, 57, 63, 65, 66, 69, 70, 71, 75, 96, 97, 98, 106, 116, 117.

9a. Idlers and idler loading:-

2, 14, 15, 17, 32, 34, 36, 46, 55, 63, 69, 70, 75, 82, 84, 97, 98, 101.

10. Field trials and service information:-

1, 2, 13, 17, 19, 21, 32, 33, 34, 37, 38, 40, 41, 42, 46, 54, 60, 66, 73, 77, 87, 88, 90, 91, 94, 95, 103, 108, 118, 119, 120.

11. Laboratory and service related conveyor belt testing:-

1, 5, 6, 9, 18, 19, 21, 23, 26, 27, 37, 38, 39, 40, 42, 48, 55, 60, 66, 73, 76, 83, 87, 88, 90, 91, 94, 95, 99, 100, 102, 103, 118, 119, 120.

12. Conveyor belt standards and specifications:-

8, 17, 23, 26, 27, 33, 37, 40, 41, 51, 52, 64, 76, 88, 90, 99, 100, 102, 105.

13. Conveyor maintenance:-

2, 4, 7, 11, 31, 32, 34, 35, 37, 52, 62, 67, 74, 97, 116.

14. Splicing and fasteners:-

16, 21, 22, 32, 39, 43, 50, 52, 62, 87, 88, 90, 97, 99, 114.

15. Stress and strain in conveyor belting:-

9, 18, 26, 27, 28, 43, 47, 48, 56, 64, 78, 82, 85, 86, 92, 93, 104, 113.

16. Examples of conveyor belt applications and installations:-

16, 19, 23, 32, 35, 52, 57, 67, 75, 79, 105, 116, 117.

17. Special types of conveyor belting:-

(i) Steel cord:-

21, 30, 39, 51, 52, 63, 71, 72, 75, 101.

(ii) Solid woven:-

2, 25, 89, 106.

(iii) Fire resistant:-

32, 38, 83, 91, 106.

(iv) Passenger conveyor:-

29, 80.

18. Comparison of belt conveying with other transportation systems:-

8, 9, 11, 12, 14, 24, 58, 59, 107, 115, 117.

<u>German articles not fully reviewed</u>

E. Bahke and T. Weigel	:	Cost comparison of various ore transport systems. Stahl Eisen, <u>94</u>, n19, p881-886, September 12, 1974.

H. P. Lachmann : The deflection resistance of rubber belts. Forschung auf dem Gebiete des Ingenieurwesens, 20, n4, p97-107, n5, p145-149, 1954.

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APPENDIX 4

Derivation Of Inter-Idler Sag Formula

During the belt selection procedure, described in Chapter 3, a formula is used to find the belt tension required to limit the belt sag to an acceptable level. It is claimed that this formula is derived from the catenary - the shape taken up by a string of negligible stiffness hanging under its own weight. This appendix demonstrates how the derivation is carried out and shows how a much simpler analysis assuming a parabolic belt shape gives the same result.

The situation assumed in this derivation is shown in Figure A4.1; in which the symbols used have the following meaning:-

I	=	Idler pitch
S	=	Belt sag
L	=	Belt length between idlers
ℓ	=	Belt length between axes' origin and point 'B' (see Figure)
Ψ	=	Angle between belt and x axis at the idler station.
ψ	=	Angle between belt and x axis at point 'B'
T_e	=	Belt tension at idler station
T_o	=	Belt tension at origin

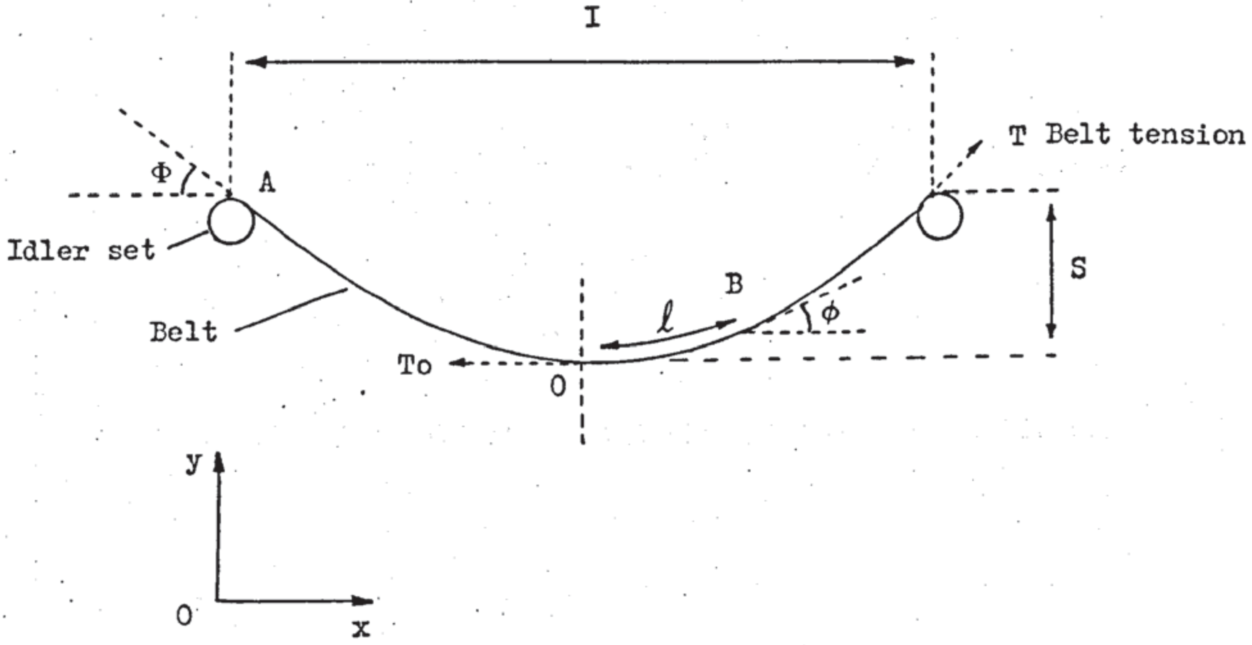

FIGURE A4.1

Inter-idler belt sag

(a) Catenary derivation

For a catenary,

$$c \frac{d\,y}{d\,x} = (y^2 + 2yc)^{\frac{1}{2}} \tag{A4.1}$$

where $$c = \frac{\ell}{\tan\psi} \tag{A4.2}$$

and $$\tan\psi = \frac{d\,y}{d\,x} \tag{A4.3}$$

so, at B,

$$\frac{\ell}{\tan\psi} \times \tan\psi = (y^2 + 2y\,\ell/\tan\psi)^{\frac{1}{2}} \tag{A4.4}$$

and $$\tan\psi = \frac{2\,y\ell}{\ell^2 - y^2} \tag{A4.5}$$

Considering half of the belt length between the idlers,

$$y = s$$

and $$\ell = \frac{L}{2}$$

So $$\tan\Psi = \frac{2S(L/2)}{(L/2)^2 - S^2} \tag{A4.6}$$

$$= \frac{S\,L}{(L/2)^2 - S^2} \tag{A4.7}$$

Three forces act on the belt length considered; T, T_o and $(W\,L/2)$ caused by a total load weight of belt and material equal to W per metre. $W\,L/2$ acts vertically; T and T_o subtend the angle Ψ as shown, and so, for equilibrium,

$$\tan \Psi = \frac{W\ L}{2\ T_o} \qquad (A4.8)$$

For small sag, $\frac{L}{2} \simeq \frac{I}{2}$ (A4.9)

and $T \simeq T_o$ (A4.10)

So, by substituting equations A4.7, A4.9 and A4.10 into A4.8:-

$$\frac{W\ L}{2\ T} = \frac{S\ L}{(L/2)^2 - S^2} \qquad (A4.11)$$

But $(L/2)^2$ is very much greater than F^2 and L is approximately equal to I, so:-

$$S = \frac{W \times I^2}{8\ T} \qquad (A4.12)$$

This is the quoted inter-idler sag formula. The finals units of S obviously depend on the units used for the other variables.

(b) Parabola Derivation

For a parabola with origin at point 0,

$$y = a\ x^2 \qquad (A4.13)$$

$$\text{So} \quad \tan \psi = \frac{d\ y}{d\ x} = 2\ a\ x \qquad (A4.14)$$

So at point A,

$$\tan \Psi = \frac{2\ a\ I}{2} \qquad (A4.15)$$

$$\text{or} \quad a = \frac{\tan \Psi}{I} \qquad (A4.16)$$

Also, from the parabola formula,

$$S = a\ (I/2)^2 = \frac{\tan \Psi \times I}{4} \qquad (A4.17)$$

As above, $\tan \Psi \simeq \frac{W\ L}{T}$ and $L \simeq T$

$$\text{So} \quad S = \frac{W \times I^2}{8\ T} \qquad (A4.18)$$

Formulae A4.18 and A4.12 are the same, and so using a much simpler derivation gives the same result as considering the belt shape to be that of a catenary. These formulae do not take into account any effect due to belt stiffness and consider inter-idler sag to be limited purely by tension. The validity of this has been studied in this work as described in Chapters 7 and 8 and in some earlier work which is reviewed in Chapter 2.

APPENDIX 5

Conveyor Belt Design Data from The Dunlop "Starflex Conveyor" Manual

At several points throughout the present thesis, reference is made to the Dunlop Starflex range of belts. For convenience the complete set of values for the belt design parameters quoted in the Starflex manual is given here. This shows the source of data used in several of the examples of design calculations which appear in the thesis. The technical data for the common belt constructions in the Starflex range is also given as this, too, is used in various parts of the thesis.

Notes On Table A5.1 (See Overleaf)

1) The capacities given in the table are based on three equal length roll idlers. For long centre idlers the capacity is approximately 80% of that shown.

2) For free flowing material such as grain, dry cement etc., the maximum surcharge angle will be of the order of 5° and the capacity can be estimated as 60% of the values shown if 20° troughing is used and 80% of the values shown if 45° troughing is used.

3) For belts operating at angles of inclination in excess of 15°, the capacities obtained from the above table should be reduced by 10%.

Belt width (mm)	Rec. max. lump (mm)		Troughing angle (degrees)	Area of load (m^2)	Average weight of material (tonnes/m^3)							
	Sized	Unsized			0.5	0.75	1.0	1.25	1.5	2.0	2.5	3.0
500	100	160	20	0.0221	40	60	80	100	119	159	199	239
			27½	0.0248	45	67	89	112	134	179	223	268
			30	0.0255	46	69	92	115	138	184	230	275
			35	0.0268	48	72	96	121	145	193	241	289
			45	0.0281	51	76	101	126	152	202	253	303
650	125	200	20	0.0401	72	108	144	180	217	289	361	433
			27½	0.0450	81	122	162	203	243	324	405	486
			30	0.0463	83	125	167	208	250	333	416	500
			35	0.0485	87	131	175	218	262	349	437	524
			45	0.0509	92	137	183	229	275	366	458	550
800	160	250	20	0.0631	114	170	227	284	341	454	568	681
			27½	0.0707	127	191	254	318	382	509	636	764
			30	0.0727	131	196	262	327	393	523	654	785
			35	0.0762	137	206	274	343	411	549	686	823
			45	0.0798	144	215	287	359	431	575	718	862
1000	200	300	20	0.1015	183	274	365	457	548	731	913	1096
			27½	0.1138	205	307	410	512	615	819	1024	1229
			30	0.1170	211	316	421	527	632	842	1053	1264
			35	0.1225	221	331	441	551	662	882	1103	1323
			45	0.1282	231	346	462	577	692	923	1154	1385
1200	250	400	20	0.1499	270	405	540	675	809	1079	1349	1619
			27½	0.1680	302	454	605	756	907	1210	1512	1814
			30	0.1727	311	466	622	777	933	1243	1554	1865
			35	0.1808	325	488	651	814	976	1302	1627	1953
			45	0.1891	340	510	681	851	1021	1362	1702	2042
1400	275	450	20	0.2077	374	561	748	935	1122	1495	1869	2243
			27½	0.2327	419	628	838	1047	1257	1675	2094	2513
			30	0.2393	431	646	861	1077	1292	1723	2154	2584
			35	0.2505	451	676	902	1127	1353	1804	2255	2705
			45	0.2618	471	707	942	1178	1414	1885	2356	2827
1600	325	500	20	0.2737	493	739	985	1232	1478	1971	2463	2956
			27½	0.3067	552	828	1104	1380	1656	2208	2760	3312
			30	0.3153	568	851	1135	1419	1703	2270	2838	3405
			35	0.3299	594	891	1188	1485	1781	2375	2969	3563
			45	0.3448	621	931	1241	1552	1862	2483	3103	3724
1800	350	600	20	0.3501	630	945	1260	1575	1891	2521	3151	3781
			27½	0.3923	706	1059	1412	1765	2118	2825	3531	4237
			30	0.4032	726	1089	1452	1814	2177	2903	3629	4355
			35	0.4220	760	1139	1519	1899	2299	3038	3798	4558
			45	0.4410	794	1191	1588	1985	2381	3175	3969	4763
2000	400	700	20	0.4360	785	1177	1570	1962	2354	3139	3924	4709
			27½	0.4884	879	1319	1758	2198	2637	3516	4396	5275
			30	0.5021	904	1356	1808	2259	2711	3615	4519	5423
			35	0.5254	946	1419	1891	2364	2837	3783	4729	5674
			45	0.5489	988	1482	1976	2470	2964	3952	4940	5928

TABLE A5.1

Capacities of troughed belt conveyors (expressed in tonnes/hour based on speed of 1 m/s and load surcharge angle of 20°)

Material To Be Conveyed	Belt Width (mm)							
	500	650	800	900	1000	1200	1400	1600
	m/s	m/s	m/s	m/s	m/s	m/s	m/s	m/s
Grain and other light flowing non-abrasive materials	2.5	3.0	3.5	3.75	4.0	4.0	4.0	4.0
Coal, crushed stone and similar lump materials	2.0	2.5	2.75	3.0	3.25	3.5	3.5	3.5
Heavy and abrasive ore, coarse size sharp stone etc.,	1.75	2.0	2.25	2.5	2.75	3.0	3.0	3.0
Prepared or damp foundry sand without ploughs	1.5 (for all belt widths)							
Prepared foundry sand and similar abrasive materials with ploughs	0.75 (for all belt widths)							
Non-abrasive materials with ploughs	1.0 (for all belt widths)							
Shake out foundry sand with small cores and castings	1.0 (for all belt widths)							

TABLE A5.2

Recommended maximum belt speeds for normal use

NOTE:- In the case of belts loaded on inclines of 10° or more it may be necessary to reduce the above speeds in order to achieve maximum capacity.

Belt Width	Troughing Idlers								Return Idlers
	Density of material (tonnes/m^3)								
	0.5	0.75	1.0	1.25	1.5	2.0	2.5	3.0	
mm	m	m	m	m	m	m	m	m	m
500	1.65	1.65	1.65	1.65	1.5	1.5	1.43	1.43	3.0
650	1.65	1.65	1.65	1.65	1.5	1.5	1.43	1.43	3.0
800	1.5	1.5	1.5	1.5	1.35	1.35	1.28	1.28	3.0
900	1.5	1.5	1.5	1.5	1.35	1.35	1.28	1.28	3.0
1000	1.35	1.35	1.35	1.35	1.2	1.2	1.13	1.13	3.0
1200	1.2	1.2	1.2	1.2	1.0	1.0	0.93	0.93	3.0
1400	1.2	1.2	1.2	1.2	1.0	1.0	0.93	0.93	3.0
1600	1.2	1.2	1.2	1.2	1.0	1.0	0.93	0.93	3.0
1800	1.2	1.2	1.2	1.2	1.0	1.0	0.93	0.93	3.0

TABLE A5.3

Recommended idler spacing

NOTE:- The recommended pitch or idler sets given in the table conforms to that quoted in BS 2890 : 1973.

Symbol	Description	Units
F_B	Friction factor for empty belt	-
F_L	Friction factor for load on level	-
L_A	Length adjustment	m
L_1	Adjusted length	m
P	Weight of moving parts	kg/m
M	Weight of load = $\frac{T}{V}$ x 0.278	kg/m
K	Drive factor	-
G	Sag factor (6.25 for 2% sag)	-
B	Approximate belt weight	kg/m

TABLE A5.4

Design parameters obtained from tables

Friction Factor		Operating Conditions			
		Average	Above Average	Exception-ally good	Regener-ative
F_B	Empty Belt	0.03	0.025	0.022	0.012
F_L	Load on level	0.03	0.025	0.022	0.012

TABLE A5.5

Friction factors (F_B and F_L)

NOTE:- Where sticky materials are conveyed, or the peak load rate is high relative to the rated capacity of the belt, a higher value should be used for F_L.

Conveyor Installation	Length Adjustment (L_A) (m)
Normal driving conditions	45
Regenerative conditions	150

TABLE A5.6

Length Adjustment (L_A)

Belt Width (mm)	Weight of moving parts (kg/m)				
	Light Duty Idlers	Medium Duty Idlers	Medium Duty Idlers	Heavy Duty Idlers	Heavy Duty Idlers & Belt
	102 mm	127 mm	152 mm	152 mm	
500	24	28	36		
650	30	38	47	52	62
800	39	48	60	65	77
900	45	55	70	79	91
1000	49	60	77	88	102
1200	63	71	95	110	128
1400	71	84	110	129	148
1600		96	127	151	173
1800			142	170	198

TABLE A5.7

Weight of moving parts (P)

Angle of wrap (degrees)	Drive Factor			
	Gravity Take-up		Screw Take-up	
	Bare Drum	Lagged Drum	Bare Drum	Lagged Drum
160	2.00	1.60	2.45	2.05
170	1.92	1.55	2.32	1.97
180	1.85	1.50	2.20	1.90
190	1.78	1.46	2.08	1.85
200	1.72	1.42	2.00	1.78
210	1.67	1.38	1.94	1.73
220	1.62	1.35	1.90	1.68
230	1.58	1.32	1.85	1.65
240	1.54	1.30	1.80	1.60
250	1.50	1.28	1.77	1.57
360	1.26	1.13	1.52	1.32
380	1.23	1.11	1.50	1.30
400	1.21	1.09	1.48	1.28
420	1.19	1.08	1.45	1.26
440	1.17	1.07	1.43	1.24
460	1.15	1.06	1.41	1.21

TABLE A5.8

Drive Factor (K)

NOTES:-

1) When calculating the driving tension required for geared tandem units, the drive factor selected must correspond with the total angle of driving wrap. When independently driven dual drum units are installed, each unit must be considered separately and the drive factor selected to correspond with the angle of driving wrap on each drum.

2) The drive factors quoted for gravity or automatic take-up systems are based on the theoretical formulae showing the relationship between angle of wrap and coefficient of friction between belt and drum necessary to give drive without slip. In the case of screw take-up units, an

adjustment has been made to the drive factor to allow for the extra tension which may be induced in the belt either:

a) To compensate for the effect of belt elongation on the drive, or

b) Due to the difficulty in measuring the amount of tension applied.

3) In those cases where an electrically or hydraulically loaded winch type take-up is used, where the induced tension can be pre-set and controlled, the drive factor should be selected to correspond with a gravity take-up system.

Belt Width	Operating Conditions		
	Light Duty	Average Duty	Heavy Duty
mm	kg/m	kg/m	kg/m
500	4.1	6.2	10.3
650	5.3	8.0	13.3
800	6.6	9.9	16.4
900	7.4	11.0	18.5
1000	8.2	12.3	20.5
1200	9.8	14.8	24.6
1400	11.5	17.2	28.7
1600	13.1	19.7	32.8
1800	14.7	22.2	37.0

TABLE A5.9

Estimated belt weight (B)

NOTE:-

The values given in the table are estimated values for use in the calculation of maximum belt operating tension which has to be carried out to make the correct belt selection. When the belt specification has been determined the weight should be checked more accurately from Table A5.17 If the actual weight of the specification differs considerably from the approximate value obtained from the above table the tension calculation should be re-checked using the more accurate belt weight.

Belt Designation		Maximum recommended working tension	
		Vulcanised Joints	Mechanical Fasteners
Type	Rating	kN/m	kN/m
STARFLEX HEAVY DUTY	200/2	20	20
	315/2	31	31
	315/3	31	31
	500/3	50	50
	630/3	63	63
	630/4	70	63
	800/4	90	80
STARFLEX EXTRA	400/4	44	40
	500/5	55	50
	630/4	70	63
	800/4	90	80
	800/5	90	80
	1000/4	110	Mechanical fasteners not recommended for continuous operation
	1000/5	110	
	1000/6	110	
	1250/4	140	
	1250/5	140	
	1250/6	140	
	1400/5	155	
	1400/6	155	
	1600/4	180	
	1600/6	180	
	2000/5	220	
	2500/6	280	

TABLE A5.10

Recommended maximum belt working tensions for the Starflex range

NOTE:- Where special environmental conditions pertain, for example, operating temperature in excess of 120°C for long periods, the belt should be selected so that the maximum operating tension does not exceed 75% of the maximum recommended tension shown.

Belt Designation		Maximum belt width (mm)		
		Material density (tonnes/m^3)		
Type	Rating	Up to 1.0	Up to 1.5	Up to 2.5
STARFLEX HEAVY DUTY	200/2	900	650	500
	315/2	1000	800	650
	315/3	1200	1000	800
	500/3	1400	1000	800
	630/3	1400	1200	900
	630/4	1800	1400	1200
	800/4	1800	1600	1400
STARFLEX EXTRA	400/4	1200	1000	800
	500/5	1600	1200	1000
	630/4	1400	1000	900
	800/4	1400	1200	900
	800/5	1800	1400	1200
	1000/4	1800	1400	1200
	1000/5	1800	1600	1400
	1000/6	1800	1800	1600
	1250/4	1800	1600	1400
	1250/5	1800	1800	1600
	1250/6	1800	1800	1800
	1400/5	1800	1800	1800
	1400/6	1800	1800	1800
	1600/6	1800	1800	1800
	1600/4 2000/5 2500/6	For details please refer to Dunlop Belting Division		

TABLE A5.11

Maximum belt width for satisfactory load support

Belt Designation		Minimum Belt Width (mm)		
Type	Rating	Troughing Angle (°)		
		20°	30°	45°
STARFLEX HEAVY DUTY	200/2	500	500	500
	315/2	500	500	500
	315/3	500	500	500
	500/3	500	500	650
	630/3	500	500	650
	630/4	500	650	800
	800/4	650	800	900
STARFLEX EXTRA	400/4	500	500	500
	500/5	500	500	650
	630/4	500	500	650
	800/4	500	500	650
	800/5	500	650	800
	1000/4	500	650	800
	1000/5	500	650	800
	1000/6	650	800	900
	1250/4	500	650	800
	1250/5	650	800	900
	1250/6	650	800	900
	1400/5	650	800	900
	1400/6	800	900	1000
	1600/6	800	900	1000
	1600/4 2000/5 2500/6	For details please refer to Dunlop Belting Division		

TABLE A5.12

Minimum belt width for satisfactory troughing

Belt Designation		Pulley Diameter (mm)		
Type	Rating	Drive Pulleys	Terminal tripper and high tension snub pulleys	Low tension bend and snub pulleys
STARFLEX HEAVY DUTY	200/2	300	250	200
	315/2	350	300	250
	315/3	450	400	300
	500/3	450	400	300
	630/3	500	450	350
	630/4	650	500	400
	800/4	800	650	450
STARFLEX EXTRA	400/4	450	400	300
	500/5	650	500	350
	630/4	500	450	350
	800/4	500	450	350
	800/5	650	500	350
	1000/4	650	500	350
	1000/5	650	500	400
	1000/6	800	650	450
	1250/4	650	500	400
	1250/5	800	650	500
	1250/6	900	800	500
	1400/5	900	800	500
	1400/6	900	800	500
	1600/6	1000	900	650
	1600/4 2000/5 2500/6	For Details please refer to Dunlop Belting Division		

TABLE A5.13

Minimum pulley diameters for satisfactory flexing

NOTES:-

1) The above dimensions represent the minimum recommended pulley diamters for belts operating at their maximum

rated tension under normal conditions.

2) For belts operating at less than 75% of the maximum rated tension, all high tension pulleys may be reduced by 10%.

3) For belts operating at less than 50% of the maximum rated tension, all high tension pulleys may be reduced by 20%.

4) For belts operating at temperatures in excess of 120°C for long periods, the pulley diameters used should be 20% greater than for equivalent belts at normal temperatures.

Belt tension at transition as percentage of max. rec. tension with vulc. joints	Troughing Angle (degrees)				
	20	27½	30	35	45
90 - 100	1.37	1.82	2.02	2.47	3.51
80 - 90	1.15	1.53	1.70	2.08	2.95
70 - 80	1.04	1.38	1.53	1.88	2.67
60 - 70	0.93	1.24	1.37	1.68	2.38
40 - 60	0.88	1.17	1.29	1.58	2.25
30 - 40	0.93	1.24	1.37	1.68	2.38
20 - 30	1.04	1.38	1.53	1.88	2.67
10 - 20	1.31	1.75	1.94	2.37	3.38

Belt tension at transition as percentage of max. rec. tension with vulc. joints	Troughing Angle (degrees)				
	20	27½	30	35	45
90 - 100	0.91	1.24	1.37	1.56	2.02
80 - 90	0.76	1.04	1.15	1.31	1.70
70 - 80	0.69	0.94	1.04	1.19	1.53
60 - 70	0.62	0.84	0.93	1.06	1.37
40 - 60	0.59	0.79	0.88	1.00	1.29
30 - 40	0.62	0.84	0.93	1.06	1.37
20 - 30	0.69	0.94	1.04	1.19	1.53
10 - 20	0.88	1.19	1.31	1.50	1.94

TABLE A5.14

Minimum transition distances

NOTES:- The minimum recommended transition distances for all STARFLEX types of belt are shown above expressed as a function of the belt width. The upper table is relevant where the top of the terminal pulley is set in line with the top of the centre roller of the idler; the lower where the top of the terminal pulley is raised by one third of the depth of the troughing idlers. These tables are based on three equal roll idler sets. In the case of long centre-roll idler sets these tables can be used as a guide, by using a belt width equivalent to three times the length of the wing idler, rather than the actual belt width.

Belt Designation		Belt Modulus
Type	Rating	kN/m
STARFLEX HEAVY DUTY	200/2	1330
	315/2	2070
	315/3	2070
	500/3	3330
	630/3	4200
	630/4	4670
	800/4	6000
STARFLEX EXTRA	400/4	2930
	500/5	3670
	630/4	4670
	800/4	6000
	800/5	6000
	1000/4	7330
	1000/5	7330
	1000/6	7330
	1250/4	9330
	1250/5	9330
	1250/6	9330
	1400/5	10330
	1400/6	10330
	1600/6	12000
	1600/4 2000/5 2500/6	For details please refer to Dunlop Belting Division

TABLE A5.15

Belt Modulus (E)

Type of material to be conveyed	Maximum Lump size (mm)	Cover Gauge (mm)		Cover Quality
		Top	Back	
Non-abrasive. Wood chips, pulp, flue dust, ground cement, very fine coal etc.	All sizes	1.0 - 1.5	1.0	STARLIFE
Slightly abrasive. Sand, earth or bituminous coal rock.	75	1.5 - 3.0	1.5	STARLIFE
Abrasive. Anthracite coal, coke, sinter, overburden.	250	3.0 - 4.5	1.5	STARLIFE
Limestone, iron and copper ore	150			
Heavy and Abrasive. Limestone, slag, iron, copper, zinc and lead ore.	250	4.5 - 6.0	1.5	STARLIFE
Heavy, Sharp and Abrasive. Iron, copper, zinc and lead ores	250 +	6.0 -10.0	2.0 - 3.0	STARLIFE
Trap rock, quartz, glass cullet etc.	All sizes			
Hot Material 80 - 120°C	Lumpy	3.0	1.5	BETAHETE
	Fines	5.0	1.5	
Hot Material 120°C +	Lumpy	3.0 - 5.0	1.5	STARHETE
	Fines	4.5 - 6.0	1.5	
Hot Material 150°C +	Lumpy	4.5 - 6.0	1.5 - 3.0	STARHETE
	Fines	6.0 -10.0		

TABLE A5.16

Gauge and quality of belt cover

NOTES:-

The table gives a guide to the minimum cover gauge and quality which should be used for various applications.

1) Where material over 120°C is to be conveyed, in addition to selecting the correct cover quality, allowance should be made in selecting the belt tension rating and pulley diameters as described in the relevant tables.

2) Where material of an oily nature is to be handled, advice should be sought from Dunlop Belting Division, and a special construction recommended to withstand the chemical effects of this duty.

Belt Designation		Carcase Weight	Carcase Thickness
Type	Rating	kg/cm width/m Length	mm
STARFLEX HEAVY DUTY	200/2	0.033	3.0
	315/2	0.035	3.2
	315/3	0.047	4.2
	500/3	0.050	4.5
	630/3	0.054	4.8
	630/4	0.071	6.4
	800/4	0.076	6.8
STARFLEX EXTRA	400/4	0.046	4.2
	500/5	0.059	5.4
	630/4	0.050	4.6
	800/4	0.053	4.9
	800/5	0.064	5.9
	1000/4	0.063	5.8
	1000/5	0.069	6.3
	1000/6	0.078	7.2
	1250/4	0.069	6.4
	1250/5	0.079	7.3
	1250/6	0.084	7.8
	1400/5	0.088	8.1
	1400/6	0.095	8.8
	1600/6	0.106	9.7
	1600/4 2000/5 2500/6	For details please refer to Dunlop Belting Division	

TABLE A5.17

Belt weight and thickness for the Starflex range of belts

NOTE:- The nominal carcase weight and thickness for the various Starflex belt specifications are tabled above. The total belt weight and thickness can be obtained by adding to the basic carcase the weight or gauge of covers in question.

Weight of covers:-

1 mm - 0.012 kg/cm/m

APPENDIX 6

Scope for Belt Service Data Collection

In Chapter 4 the need for a data collection procedure to help relate belt design and selection more to field conditions was stressed and it was mentioned that information from production, sales, control testing, service-related testing, site visits and field trials was required. This appendix lists the information that could be collected from these sources. It should be remembered that no such list can ever really be complete because future belt construction or conveyor structure developments might mean that service factors other than those considered now might become of concern. It is therefore important to collect as much data as possible, even though certain information might, at present, seem to have no bearing on belt service requirements.

a) Information from production

(1) Belt number.

(2) Date of production.

(3) Length and width of production.

The three pieces of information above will have no bearing on belt performance, but are necessary to help to ensure that full records of the total production are kept.

(4) Cover compound, interply compound and fabric used.

(5) Cure details. It has been shown that the amount of cure can influence ply delamination [80]. The reference number of the press used for the production should also be recorded in case it is found that belts from one particular press are inferior to others.

(6) Details of belt shrinkage after cure. (If different amounts of shrinkage occur across the width of the finished belt, a natural curvature might be introduced. This can cause problems when trying to track the belt.)

(7) Number and form of any defects noticed during production.

(8) Number and form of fabric joints occurring during production.

b) <u>Information from Sales Department</u>

(1) Date delivered and, if possible, date of installation.

(2) Belt site, application and installation details. This would include the information required for the belt design procedure which should be recorded on a standard calculation sheet.

(3) Reasons for final belt construction selection.

(4) Length and width of belting supplied.

(5) Any complaints made about the belt.

Much of the information listed above also applies to the site surveys, mentioned in Section 4.1, carried out as a customer service. The information from these should also be included as part of the data collection system.

c) Information from Control Testing

Previous studies of how belt service performance might be affected by the properties measured in control tests are described in Chapter 2. A list is given below of the information that can be collected from test machines available to most belting manufacturers.

(1) Fabric tensile strength and elongation characteristics in both warp and weft directions.

(2) Cover and interply compound tensile strength and elongation details.

(3) Compound abrasion resistance characteristics.

(4) Compound resistance to flexing.

(5) Compound resistance to attack from oxygen and ozone.

(6) Compound co-efficient of friction details.

(7) Whole belt tensile strength and elongation characteristics.

(8) Cover-to-ply and ply-to-ply adhesion properties.

(9) Cover gauge and interply thickness.

(10) Troughability data.

(11) Whole-belt flexibility characteristics in both longitudinal and transverse directions.

(12) Fastener holding capabilities under static conditions.

An experimental programme should also be set up to study the effects of temperature changes on all the properties listed above.

It should be noted that the list above does not include control tests for conveyor belts requiring special properties such as fire resistance or oil resistance. (An excellent example of how a control test can be designed to be service related is given by the National Coal Board Drum Friction test for fire resistant belting - see Section 2.2.)

d) <u>Information from Service</u>

<u>Related Testing</u>

A complete knowledge of a test's relevance to service can only be ascertained after correlation has been found between the test and service data. It is, however, important to test all standard belt constructions on any machine that is thought to be service related as this will, at least, give comparative data on the different types of belting. The information that should be obtained by Dunlop Angus Belting Group from such testing machines includes:-

(1) Flex fatigue characteristics of whole belt samples on small diameter pulleys.

(2) Impact resistance properties with the pendulum impact tester described in Section 2.2.

(3) Splicing and fastener efficiencies under dynamic conditions. (The mechanism of fastener failure should be studied to help in the development of better mechanical jointing techniques.)

(4) Belt elongation properties during a loading cycle representing the tensions occurring in service. (Extension modulus is a parameter appearing in some formulae in the belt selection procedure - see Chapter 3.)

(5) Whole-belt abrasion resistance.

e) <u>Information from Site Visits</u>

It is realised that full information about a belt's performance cannot be obtained from every routine visit made to a conveyor installation by Dunlop Belting Division sales personnel. However, if all available information is collected, a complete picture will eventually be built-up. (In this connection, as mentioned in Section 4.3, the possibility of employing technicians purely to carry out data collection should be considered.)

The information collected could include the following details:-

(1) Estimate of belt running hours.

(2) Any evidence of belt tearing, cover wear or cover cracking.

(3) Any obvious excessive belt sag.

(4) Any fastener damage.

(5) State of maintenance of equipment.

(6) Spillage or load slippage occurring.

(7) Any other apparent form of damage (for example, edge-wear due to belt rubbing against some part of the conveyor machinery).

(8) Any changes to the original conveyor equipment.

If possible, any portion of the belt removed from service should be collected for testing in order to measure the parameters discussed in (c) and (d) above.

f) Information from Field Trials

It is vital to ensure that full information is collected from any field trial belt. Current standard constructions as well as any developments should be included in field trials purely to get information from belts that have proved, perhaps due to overspecification of design properties, to be satisfactory in service. If a field trial is to be carried out as a scientifically controlled investigation then the full co-operation of the purchaser is essential to document the daily happenings on the conveyor. In addition, regular and frequent visits by Technical Department staff are essential to ensure that no changes are made to the load, loading system, tonnage rate or conveyor installation.

The measurements and observations made could include:-

(1) Comprehensive load and installation details including the exact state of conveyor maintenance.

(2) Number of hours of belt service and the tonnage carried.

(3) The situation at transition points.(For example, is belt deforming badly?)

(4) The situation at changes in the conveyor gradient. (For example, is the belt edge buckling at a concave bend?)

(5) Positions where spillage and/or slippage of the load occur.

(6) Power consumption on starting and under load.

(7) Quantitative assessments of cover cracking, cover peeling or any other visible damage.

(8) Longitudinal and transverse belt deformation measurements.

(9) The mechanism of joint failure.

(10) Reasons for removal of any piece of belting and the number of service-hours of such pieces.

Again, any belting removed during the trial should be tested to find the changes in belt properties that occur during service.

The above comments and suggestions are only guidelines to the work that should be carried out. Measurement techniques and data recording sheets should be devised for each section of the data collection service if it is to work efficiently. It must be emphasised, again, that a data collection system is a long term project, but the possible benefits are great. Complete co-operation and co-ordination between several departments and groups of people is essential if such a system is to provide meaningful results.

APPENDIX 7

Programming Details of Work carried out to show Application Areas where Load Support determines the result of the Belt Selection Procedure

In Section 5.1 it was explained how a computer program was used to carry out a large number of belt tension calculations in order to try and show the areas where belt selection could be made by reference to load support data alone. (That is, where the tension calculation was unnecessary.) This appendix gives the programming details of that work.

The Program

```
10  INPUT W, P, B, M, I, Z
15  FIXED 2
20  PRINT "WIDTH (MM) = "; W
21  PRINT "WEIGHT OF MOVING PARTS (KG/M) = "; P
22  PRINT "APPROXIMATE BELT WEIGHT (KG/M) = "; B
23  PRINT "LOAD WEIGHT (KG/M) = "; M
24  PRINT "IDLER SPACING (M) = "; I
25  PRINT "TENSION MAXIMUM FOR LOAD SUPPORT (KN/M) = "; Z
26  PRINT "FRICTION FACTOR = 0.03"
30  FOR K = 2.6 TO 1 STEP - 0.2
35  FOR L = 50 TO 5 STEP - 5
100 PRINT "L", "H", "K", "TE", "TM/W"
```

```
110 FOR H = 20 TO 0 STEP - 1
115 IF H > 2/3 * L THEN 200
120 LET L1 = L + 45
130 LET T1 = (0.03 * L1 * (P + M) + M * H) * 9.8
140 LET T2 = B * H * 9.8
150 LET T3 = 0.4 * 9.8 * 0.03 * L1 * P
160 LET T4 = 6.25 * 9.8 * (B + M) * I
170 LET T5 = K * T1
180 LET T6 = T1 + T2 - T3 + T4
190 IF T5 > T6 THEN 200
191 IF T6/W > (Z + 2) THEN 220
192 PRINT "SAG TENSION CONSIDERED FOR NEXT CASE"
195 LET T5 = T6
200 IF T5/W > (Z + 2) THEN 220
210 PRINT L, H, K, T1, T5/W
215 IF T5/W < Z THEN 230
220 NEXT H
230 NEXT L
240 NEXT K
250 GOTO 10
260 END
```

<u>Explanation of the Programme Elements</u>

A flow diagram is presented in Figure A7.1 to summarise the program.

<u>Line 10</u> defines the information that must be fed into the computer. In this case six figures must be input - the belt width, which for the purpose of the program was designated W, the weight of moving parts,

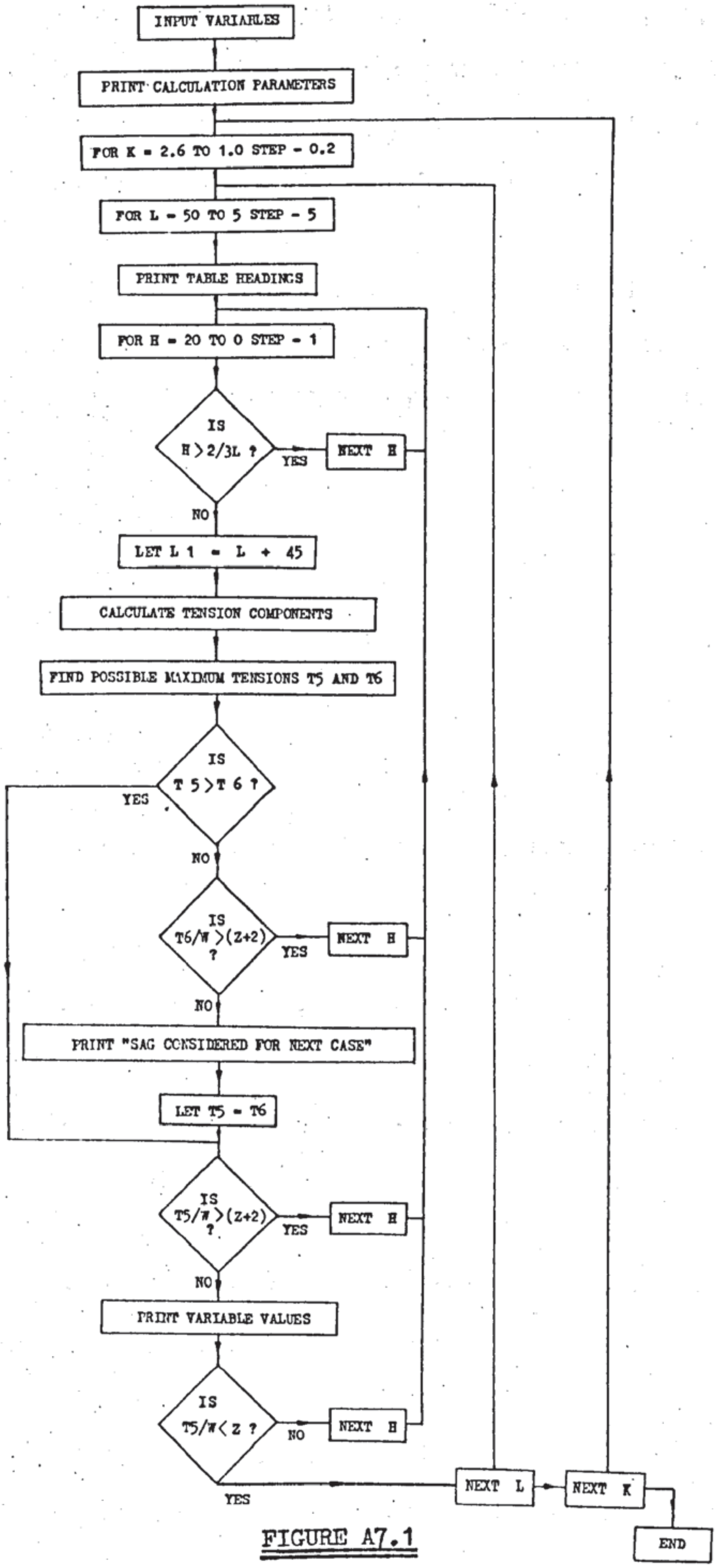

FIGURE A7.1

Flow diagram of program described in Appendix 7

P, the approximate belt weight, B, the load weight, M, the idler spacing, I, and the value of the limiting tension, Z, below which belts can be correctly selected by reference to the load support data only. (For full details see Section 5.2.3.)

All the belt parameter values used in the investigation are given in Figure 5.1.2 but, as an example, for a 500 mm wide belt carrying a load density 1.0 t/m^3 the following figures would be applicable:-

P = 24 kg/m

B = 4.10 kg/m

M = 28.1 kg/m

I = 1.65 m

Z = 20 kN/m

Thus when the program is started a string of six figures would have to be input:-

500, 24, 4.1, 28.1 1.65, 20

As soon as the sixth number has been input the program would continue to the next stages.

<u>Line 15</u> fixes the number of figures after the decimal point as two for all further calculations in the program.

<u>Lines 20 to 26</u> print out the data that has been input as a check for the operator, and also to provide a printed record. An example of the print-out is given later.

Lines 30 and 240 form a loop for the values of K, the drive factor. The program is first of all run with K = 2.6, and then automatically re-runs for K = 2.4. This is continued in equal steps until K = 1.0.

Lines 35 and 230 form a similar loop for the value of the conveyor centres length. In this case the values run from 50 m to 5 m in steps of 5 m; in later programs different ranges including longer lengths were used.

Line 100 prints out a heading to the results table.

Lines 110 and 220 form a loop for the value of the conveyor lift. In this program 'H' decreases from 20 m to zero in steps of 1 m.

Line 115 stops the calculation if the conveyor lift is greater than two thirds of the centres length as this would not represent a practical situation for a normal belt conveyor. If it is found that this situation does exist,Line 115 makes the program go straight to Line 220 and the next value of 'H' is tried. This sequence is followed until 'H' does become less than two thirds of L, in which case the remainder of the program is allowed to continue.

Line 120 adds on the length correction. This correction is a standard part of the belt selection procedure and is intended to allow for frictional effects at the conveyor end pulleys (see Section 3.3 for full details). In all elevating or horizontal conveyors (as is the situation assumed for the present work) the length correction is 45 m.

Lines 130 to 160 work out the various tension components that occur in the tension calculation.

Line 170 determines the working tension that has to be induced in the belt to ensure that movement of the load and belt will occur. This is equal to the drive factor x effective tension ($K \times T_e$) and is called T_5 in the program. (T_e is called T_1 in the program because numerical subscripts have to be used with the BASIC language of the Hewlett Packard model 9830-A programmable calculator used.)

Line 180 works out the tension which must be induced in the belt to limit sag between adjacent idler sets. This is called T_6 in the program.

Line 190 finds which of these two possible maximum belt tensions (T_5 and T_6) is greater. If it is T_6, a statement to this effect (Line 192) is printed out. If T_5 is the larger then the program jumps to Line 200.

Line 191 avoids excessive print-out by ensuring that the maximum tension result, T_M, is not printed if it is more than 2 kN/m above the limiting tension of Z kN/m. In some cases this factor (Z + 2) was changed to (Z + 10) when the tension increments between the fictitous conveyor installations formed by the input data increased. This was intended to ensure that the maximum tension value nearest to,but greater than,the limiting tension would always be printed out.

Line 195. If the program reaches Line 195 then it has already decided that the maximum tension is controlled by sag considerations and is T_6. A notice to this effect has been printed out and so there is no longer any need to differentiate between the two possible maximum tension values, T_5 and T_6, and there is no longer any interest in how the maximum tension figures was arrived at. Line 195 therefore gives T_5 a new value equal to that of T_6. (Remember this only happens if T_6 was greater than the original T_5.) This means that T_5 is now the value of the maximum tension no matter how the actual value was arrived at. This process is done to simplify the rest of the program because it does away with the need to deal with two possibilities for the maximum tension.

Line 200 avoids excessive print-out in a similar way to Line 191. (This line will only function if the original T_5 is the maximum tension. If T_6 is the significant quantity, this process has already been carried out by Line 191.)

Line 210 prints out the value of the conveyor centres length, the conveyor lift, the drive factor, the effective tension, T_E, in Newtons and the maximum working tension in kN/m. (Working tension is usually described in terms of total tension across the belt width divided by the belt width. Hence the unit for this is kN/m.)

Line 215 causes the program go to the next value of conveyor length if the value of the maximum tension rating has gone below the limiting tension Z. This is done to avoid unnecessary calculations when it is already known that the maximum working tension for the next situation will be less than Z.

Line 250 prepares the calculator for the next set of input data, and, if required, another set of six figures can be printed in.

This program will print out the maximum tension per unit width in the range from (Z + 2) to the first value that is less than 'Z' along with the corresponding values of L, H, K and T_E.

Print-Out

If the six figures mentioned above and corresponding to a 500 mm wide belt carrying a load of density 1.0 t/m^3 were input, then the print-out format would be as shown in Figure A7.2. The print-out process as shown would continue as the program gradually worked its way through all the possible combinations of values of the conveyor and belt parameters used in the tension calculations.

In the example shown the maximum tension must always have been found from multiplying the drive factor, K, by the effective tension, T_E. If this had not been the case, and the tension required to limit sag had determined the maximum tension, then the marker "SAG TENSION CONSIDERED FOR NEXT CASE" would have appeared. (This was only included in the program for information and to check that the sag tension was, in fact, being considered.)

The print-out results were transferred to tables showing the conveyor situations where belt selection could be made by reference to load support data alone. An example of this is shown in Figure 5.1.3.

WIDTH (MM) = 500.00

WEIGHT OF MOVING PARTS (KG/M) = 24.00

APPROXIMATE BELT WEIGHT (KG/M) = 4.10

LOAD WEIGHT (KG/M) = 28.10

IDLER SPACING (M) = 1.65

TENSION MAXIMUM FOR LOAD SUPPORT (KN/M) = 20

FRICTION FACTOR = 0.03

L	H	K	T_E	TM/W
50.00	9.00	2.60	3931.25	20.44
50.00	8.00	2.60	3656.07	18.01

L	H	K	T_E	TM/W
45.00	10.00	2.60	4129.88	21.48
45.00	9.00	2.60	3854.69	20.04
45.00	8.00	2.60	3579.51	18.61

FIGURE A7.2

Example of print-out format from program used to find situations where load support alone determined belt selection

APPENDIX 8

Programming Details of Tension Calculations for Steel Plant Conveyors

In Section 5.1 a proposal was put forward for programming the belt selection procedure: a demonstration was given to show how efficient a computer could be to carry out various parts of the design procedure. This involved carrying out the tension calculations for a large number of conveyors representing one particular customer's requirements on several sites. The University of Aston's computer was used for this exercise.

In this particular instance it was required to find the effective tension, T_e, and the maximum working tension per unit width of belt for approximately one hundred and seventy conveyors.

The program was written in Fortran and basically ran through the procedure summarised in flow diagram form in Figure 3.3.1,printing out various statements as required.

The actual program was as follows:-

```
      MASTER TENSION
      REAL W, L, H, T, V, I, P, A, K, F, M, C
   10 ICOUNT = 0
      WRITE (2,1000)
 1000 FORMAT (1H1/49H CONVEYOR BELT LENGTH LIFT CAPACITY
     1 60H BELT IDLER WT MOVING APPROX DRIVE EFFECTIVE MAXIMUM/
     A 15X, 16H WIDTH (CENTRES), 17X,
     B 40H SPEED PITCH PARTS BELT WT FACTOR,
     3 21H TENSION   TENSION/16X, 29H(MM)  (M)  (M)  (T/H,
    434H) (M/S)  (M)  (KG/M)  (KG/M), 12X, 18M (N)  (KN/M)//)
   20 READ (1,1001)NAME, W, C, H, T, V, I, P, B, K
 1001 FORMAT(3A4,3X,F5,0,2F10,1,F5,0,F5,2,F5,1,F5,0,F5,1,F5.2)
      IF(W)  900,900,0
      ICOUNT  =  ICOUNT + 1
      F=0.03
      L = C/2
      M = (T/V)*0.278
      L1  =  L  +  45
      T1  =  F  *  L1  *  P  *  9.8
      T2  =  F  *  L1  *  M  *  9.8
      T3  =  M  *  H  *  9.8
      T4  =  0.4  *  T1
      T5  =  6.25  *  9.8  *  (B + M)  *  I
      T6  =  B  *  H  *  9.8
      T7  =  (T1  +  T2  +  T3)
      T8  =  K  *  T7/W
      T9  =  (T7 + T6 - T4 + T5)/W
      IF (T9, GT, T8)  T8 = T9
  100 WRITE (2,1006) NAME, W, L, H, T, V, I, P, B, K, T7, T8
```

```
0030          IF (T9, EQ, T9) WRITE (2,1004)
0031     1004 FORMAT (1H+, 110X, 3H SAG)
0032     1006 FORMAT (/1X, 3A4, F7.0,F9.1,F8.0,F7.2,F7.1,F9.0,F9.1,F9.2,
0033        1      2F11.2)
0034          IF (ICOUNT-25) 20, 10, 10
0035      900 STOP
```

Explanation of Program Elements

Line 1 is the program title.

Line 2 shows that variables designated by the symbols W, L, H, T, V, I, P, B, K, F, M and C will be used in the program. The term REAL shows, in this case, that these variables need not be integers.

Line 3 sets a special counter to zero. The reason for this will become clear later.

Line 4 prints out headings to the results tables at the top of the print-out page.

Lines 5 - 10 define the format of the headings printed by Line 4. An example of the print-out is given after the program details.

Line 11 details the data that must be fed into the computer. This is the NAME of the conveyor (that is, the conveyor user's code for that particular installation), W (used, in this case to signify the belt width), C (the total belt length), H (the conveyor lift), T (the capacity of the conveyor), V (the belt speed), I (the idler pitch), P (the weight of moving parts), B (the approximate belt weight) and K (the drive factor). Normally, conveyor centres length is quoted in belt specifications. It so happened that, in this case, the total belt length was given by the original equipment manufacturer. Rather than manually divide this figure by two for every conveyor and then feed the result into the computer, it was decided that the program should be written to accept the total belt length and then allow for this later in the calculations. A slight acceptable error would be introduced because, strictly speaking, the belt length will be slightly more than twice the centres length due to the extra length of belt needed to go round the end pulleys and other equipment such as gravity take-ups. However, as no information was available about the extra belt length that had been allowed for, it was decided to carry out the work on the basis already explained.

Line 11 shows that there are ten pieces of information to be fed into the computer. This might seem excessive for each conveyor but it is no more than would have to be collected to carry out the selection procedure normally. In fact, if a sophisticated program with data storage facilities was being used then some of the conveyor dependent variables such as P, B and K could be selected automatically by the computer from the store instead of by the operator from the Starflex manual.

Line 12 defines the format that has to be used for the input data. Computer cards were the most suitable facility for data input on the machine available. It was therefore important to type the conveyor information in the correct format on the cards. (One card was used for each conveyor and each card would have ten pieces of information.)

Line 13 is a device to stop the program when all the calculations have been carried out. If the belt width, W, is input as less than or equal to zero then the program stops. Therefore, when all the data cards have been typed an additional card is included with W negative or equal to zero. When this card is reached, the computer will know that the data has finished and so will stop the print-out. If 'W' is greater than zero (as will always be the case for a practical situation) the program continues.

Line 14 adds one to the value stored in the counter of Line 3.

Line 15 defines F, the friction factor to be used in the tension calculations, as 0.03. This is the value, from the manual, for "average conditions" and was applicable to every conveyor.

Line 16 makes L, the conveyor centres length, equal to C/2, (half the total belt length).

Line 17 finds the load weight, M (kg/m), which is equal to the conveyor capacity, T (t/h), divided by the belt velocity, V (m/s), all multiplied by a units factor of 0.278.

Line 18 finds the corrected conveyor centres length, L1, by adding 45 m to the length L.

Line 19 to 24 works out the various tension components occurring in the calculation.

<u>Line 25</u> calculates the effective tension, T_e, in Newtons. In the program T7 represents T_e because only numerical subscripts are available in Fortran.

<u>Line 26</u> calculates T8 which represents the lowest value of maximum working tension that must be induced in the belt to ensure drive. T8 is actually equal to $(K \times T_e)/W$ and so its units are kN/m.

<u>Line 27</u> calculates T9 which is the maximum working tension that must be induced in the belt to prevent excess sag between the idlers.

<u>Line 28</u> finds which is the larger of T8 and T9. If T9 (the "sag consideration" tension) is the larger, the computer gives T8 a new value equal to that of T9.

<u>Line 29</u> is the command for the program to print-out the original input data and results of the calculation.

<u>Line 30</u> prints out a market SAG if the value of T8 equals that of T9. Thus the marker will only be printed if it is sag considerations that decide the maximum working tension in the belt (see Line 28). This was done for information purposes and was used later in an analysis of the factors deciding belt selection for a variety of situations. (See Section 6.2.)

<u>Line 31</u> defines the positioning of the SAG marker.

Lines 32 and 33 give the format in which the print-out from Line 29 will be given.

Line 34 looks at the value of the counter of Lines 3 and 14. If (ICOUNT-25) is less than zero then the program returns for another set of input data. If it is equal to or greater than zero then the program begins again, ICOUNT is put equal to zero, a new page of print-out is started and a new heading is printed before any more data is input. This is done so that each page of computer print-out has a results heading and gives the results for twenty-five conveyors. That is, it is a device to give a neat set of results (in practice (ICOUNT-25) will never become greater than zero).

Line 35 stops the program. This line is only reached if the belt width, W, has been input with a negative or zero value. (See Line 13.)

Example of Data Input and Print-out

Two of the conveyors actually from the scheme analysed had the following characteristics:-

	(a)	(b)
Conveyor Reference:	70007	65A
Belt width (mm):	1000	1000
Total Belt Length (m):	39.4	265
Conveyor Lift (m):	0	33
Capacity (t/h):	616	1000
Belt speed (m/s):	1.25	1.93
Idler pitch (m):	1.0	0.9
Weight of moving parts (kg/m):	60	60

Approximate belt weight (kg/m):	20.5	20.5
Drive factor:	1.68	1.35

One computer card was used to input the data for each conveyor. The actual format of the data on the card is defined by Line 12 of the program but basically consists of a line of ten pieces of information spaced out across the width of the card. Thus, for the two conveyors being considered, two cards would be prepared with the following data:

(a)	70007	1000	39.4	0.00	616	1.25	1.0	60	20.5	1.68
(b)	65A	1000	265.0	33.00	1000	1.93	0.9	60	20.5	1.35

The print-out would then be as shown in Figure A8.1.

The print-out shown demonstrates that conveyor 70007 had its maximum working tension decided by sag considerations whereas that of 65A was decided by drive requirements.

The results of this work were presented to Dunlop's manager dealing with original equipment manufacturers and they were then used as a basis for selection of belts for the scheme under consideration. The most obvious advantage of the computer analysis of this scheme was the fact that the program took only minutes to run whereas it is unlikely that time would have been available at all to do the calculations manually. The reaction to this work from the Dunlop personnel concerned with belt selection is discussed in Section 5.2.

CONVEYOR	BELT WIDTH (MM)	LENGTH CENTRES (M)	LIFT (M)	CAPACITY (T/H)	BELT SPEED (M/S)	IDLER PITCH (M)	WT MOVING PARTS (KG/M)	APPROX BELT WT (KG/M)	DRIVE FACTOR	EFFECTIVE TENSION (N)	MAXIMUM TENSION (KN/M)
70007	1000	19.7	0.00	616	1.25	1.0	60	20.5	1.68	3706.72	12.90 SAG
65A	1000	132.5	33.00	1000	1.93	0.9	60	20.5	1.35	57200.91	77.22

FIGURE A8.1

Example of print-out format from program used to calculate

maximum belt tension

APPENDIX 9

Calibration of Load Washers used in experimental investigation

It was proposed that the normal forces occurring at idler supports should be measured during the experimental work, and it was thought that load cells of the type described as "load washers" would be suitable for the task. These have dimensions similar to those of a normal washer and can be used to measure forces in areas inaccessible to the usual forms of load cell.

The type of load washer chosen for the investigation was manufactured by A. L. Design Limited and the specification details are shown in Table A9.1. It was claimed by the supplier of these devices that shear forces across the surface of the washer would have no noticeable effect on the output signal and that no special loading arrangement would be required to allow for off-centre loading of the washer. Usually, with all types of load cells it is important to ensure that loading is central and a common method of doing this is shown in Figure A9.1.

Preliminary experiments with one load washer showed that the quoted maximum input of ten volts gave a stable output and so a circuit to operate at this voltage was built up to include the four load washers. The design of the circuit was such that an input voltage of ten volts was, at all times, present on all four washers and a switch allowed the output from each to be read, in turn, from a digital voltmeter. Balancing potentiometers were present to allow a zero output to be

Approximate outside diameter	30 mm
Approximate internal diameter	10 mm
Approximate height	15 mm
Load capacity	225 kg
Excitation voltage	10V, A.C or D.C
Input resistance	350 ohms
Output resistance	350 ohms
Rated output	1.84 m V/V at capacity
Safe overload	150% of rated capacity
Ultimate overload	250% of rated capacity
Circuit layout	Four active arm Wheatstone Bridge

TABLE A9.1

Specification details of load washers

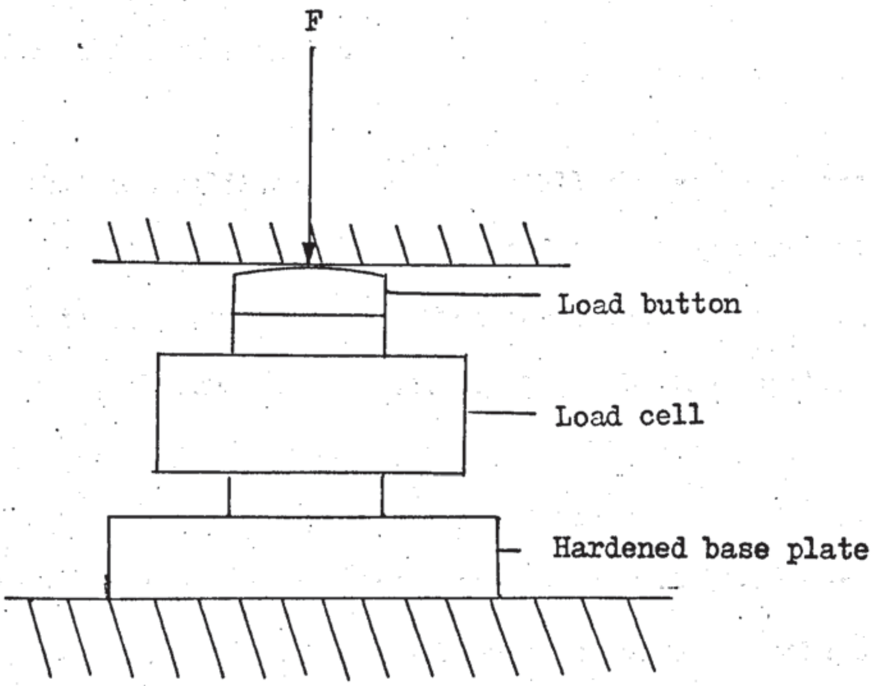

FIGURE A9.1

Common method of installation of
load transducers

obtained under unloaded conditions. A circuit diagram which, for simplicity, shows only two load washers is given in Figure A9.2.

Each load washer was then calibrated by positioning it on the platens of an Amsler tensometer and several load cycles were carried out before any measurements were taken.

It was found that the calibration curves of all the washers were neither linear nor reproducible, and it was suspected that they were not performing within their specification. Therefore, one load washer was returned to the suppliers for evaluation. Tests by the suppliers confirmed that the washers were not meeting the published specifications and it was suggested that the problems could be overcome if provision for non-central loading was included. As the special idler set designed with the load washers in mind had now been constructed and the experimental investigation of transverse load support was about to start, it was decided that the load washers in hand would have to be used. Therefore, an attempt to improve the output of the load transducers was made by including spherical washers on the special idler set. (See Figure A9.3.) These washers were intended to ensure that central loading of the load cell was obtained; spherical washers designed specifically for use with the load washers being used were not commercially available, and it was found to be necessary to use washers having a slightly greater internal radius than the load washers themselves. (See Figure A9.4.)

When the calibration experiments were repeated with the spherical washers present, much improved results were obtained. The calibration curve was still, however, not completely linear or reproducible -

particularly at the lower loadings. A typical result is shown in Figure A9.5. As, during the experiments, the load washers would have a certain amount of preloading due to the weight of the idler rollers and belt, it was decided that the calibration curve would be assumed to be linear. An average calibration factor ignoring loadings below 0.4 kN was calculated to allow easier analysis of results and still provide adequate accuracy. The calibration factors found for the four load washers are given in Table A9.2.

In practice it was found that the load washers did not perform well and only qualitative statements about idler loadings could be made from analysis of their outputs. (See Chapter 8.) However, as this investigation of idler support loads was only a relatively minor part of the whole experimental programme, the poor load transducer performance by no means invalidated the rest of the work.

During the experiments it was noticed that the performance of the load washers improved as the load on the belt increased. It was found also that the output of the load washers was not affected by the various machines working in the same area and, at high loadings, no drift of the output signal was observed. These points suggest that the performance of the load washers could be improved for further work by preloading them to a much greater extent. This would require some form of clamping arrangement to press the idler roller down onto the load transducer. Further improvements would almost certainly be achieved if spherical washers designed specifically for the load washers being used were to be purchased or manufactured. If these modifications still resulted in unsatisfactory performance, but load washers were still to be used because of their very small size, it would be necessary to adapt the idler set to allow washer loading as shown in Figure A9.1.

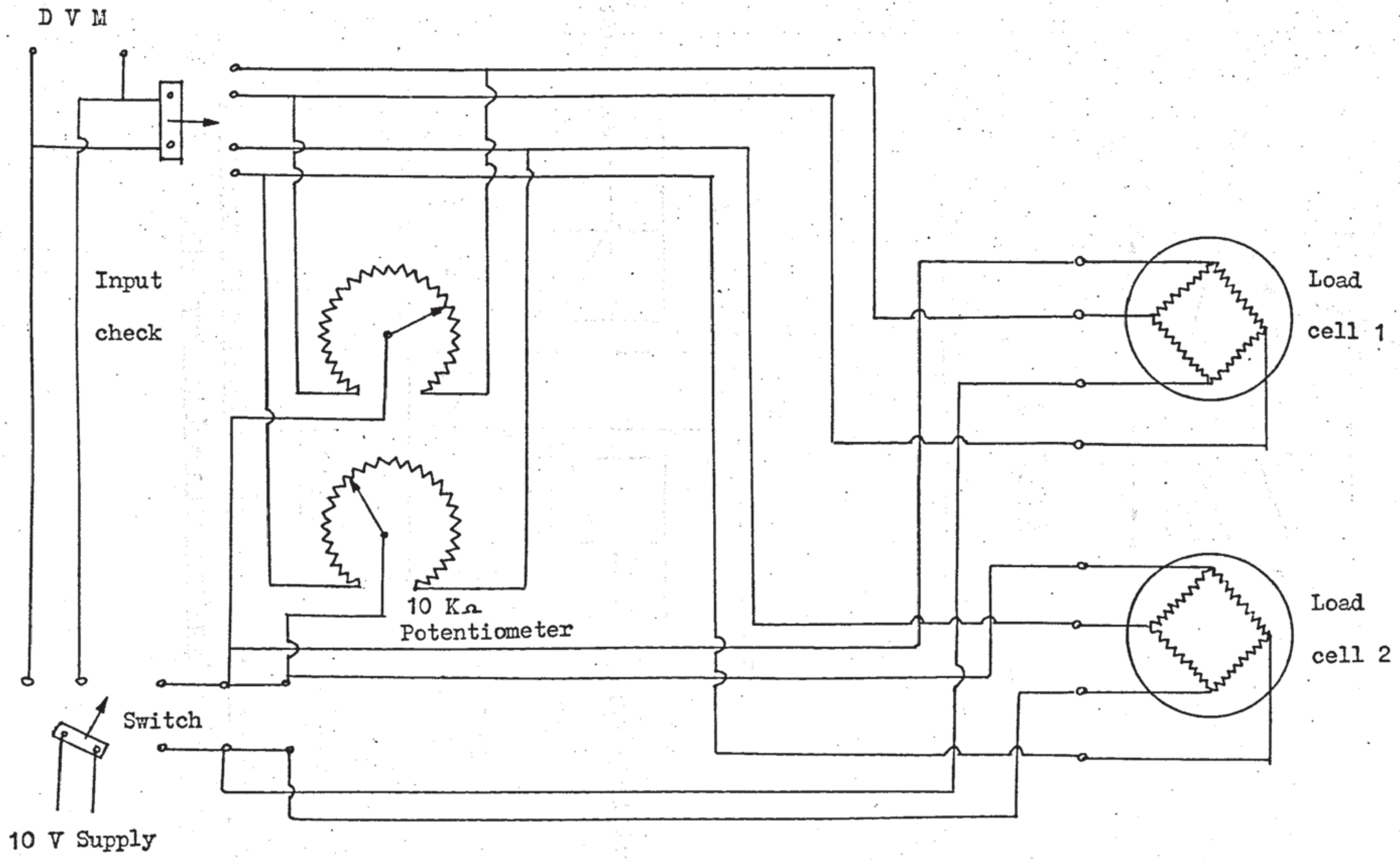

FIGURE A9.2

Diagram of circuit for load washers

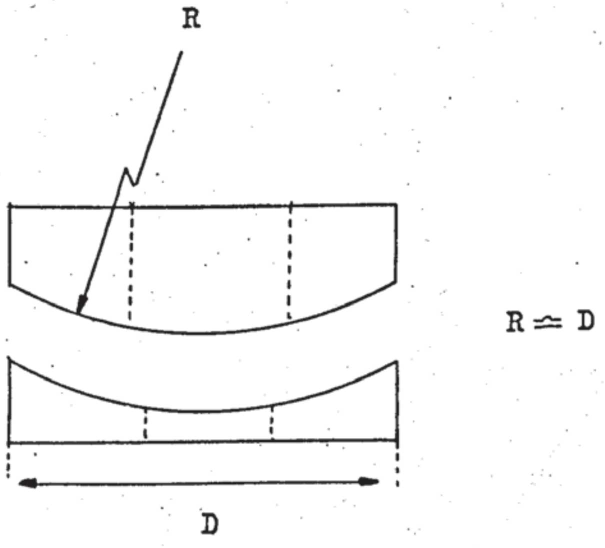

FIGURE A9.3

Cross-section through spherical washers

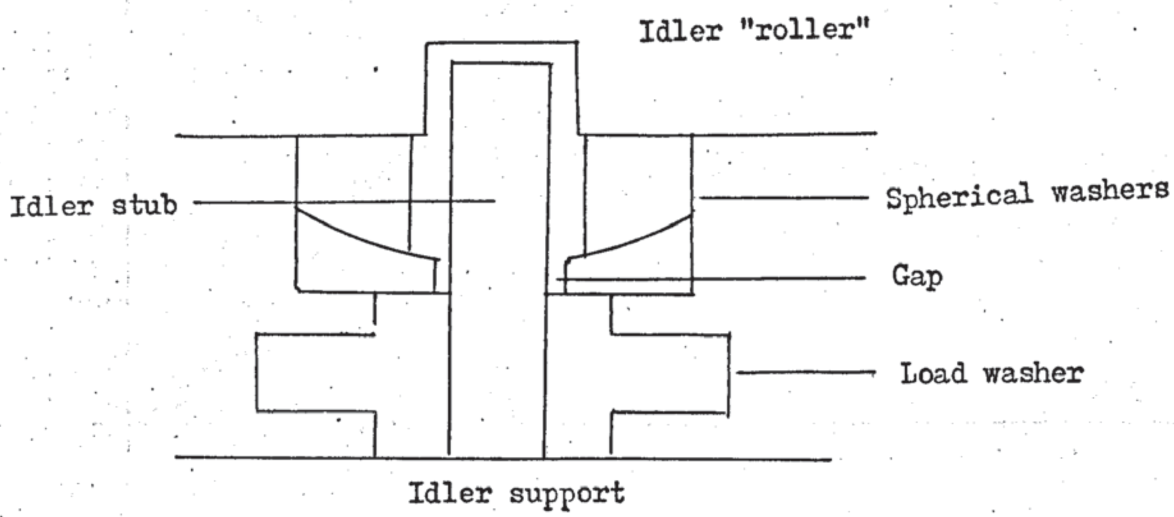

FIGURE A9.4

Loading arrangement of load washers showing gap between idler stub and lower spherical washer

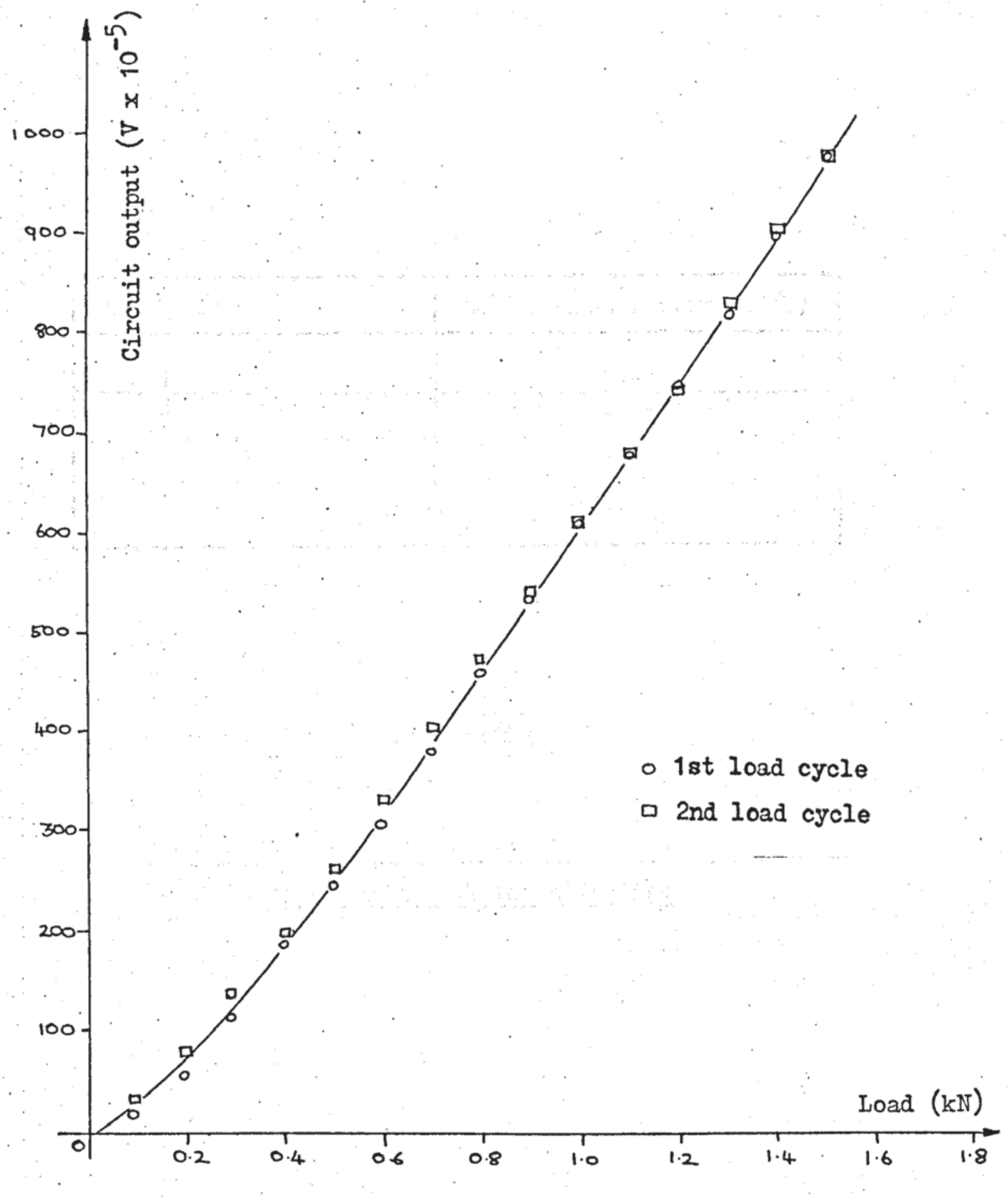

FIGURE A9.5

Calibration curve for load washer 1

Load Washer	Calibration factor (kN/V)
1	1.43
2	1.45
3	1.60
4	1.34

TABLE A9.2

Force calibration factor for load washers used in experimental investigation

APPENDIX 10

Derivation of correction factor for forces occuring at idler roller supports

During the experimental work described in Chapters 7 and 8 a special idler set was used. The length of the centre roller of this idler set could be varied so as to be compatible with any standard belt width being tested. However, in order to minimise construction costs, only one length of idler side roller was used and this corresponded to a belt width of 1400 mm. This meant that for belts less than 1400 mm wide the side rollers would be longer than those commonly used in service, and so a correction had to be made to any force measurements taken at the supports of these rollers. A derivation of this correction is given below.

In Figure A10.1 the situation where the side roller is longer than necessary is shown. Forces F_1 and F_2 act normal to the idler surface at the two roller supports. In Figure A10.2 the standard length of side roller for the belt width is shown and in this case, under the same loading conditions as in Figure A10.1, the two forces at the ends of the roller are F_1' and F_2' . It is required to find the relationship between F_1, F_2, F_1' and F_2' . During the experiments F_1 and F_2 would be measured.

Considering Figure A10.1, if the load force in the y direction due to the mass of the roller itself is given by w per unit length and the load function on the roller due to belt and material is given by

$F(x)$ then, for equilibrium in the y direction,

$$F_1 + F_2 - w\ell - \int_0^{\ell - a} F(x)\,dx = 0 \qquad (A10.1)$$

Taking moments about point 'B' give :-

$$F_1\ell - \frac{w\ell^2}{2} - \int_0^{\ell - a} F(x)\,x\,dx = 0 \qquad (A10.2)$$

Similarly, for the conditions shown in Figure A10.2,

$$F_1' + F_2' - w(\ell - a) - \int_0^{\ell - a} F(x)\,dx = 0 \qquad (A10.3)$$

and

$$F_1'(\ell - a) - \frac{w}{2}(\ell - a)^2 - \int_0^{\ell - a} F(x)x\,dx = 0 \qquad (A10.4)$$

Subtraction of equation A10.3 from A10.1 gives,

$$(F_1 - F_1') + (F_2 - F_2') = w\,a \qquad (A10.5)$$

Subtraction of equation A10.2 from A10.4 gives,

$$F_1' = \frac{F_1\ell - \frac{w}{2}(2a\ell - a^2)}{(\ell - a)} \qquad (A10.6)$$

Substituting this into equation A10.5 gives

$$F_2' = F_1\ell - F_1 + F_2 - wa - \frac{(F_1\ell - \frac{w}{2}(2a\ell - a^2))}{(\ell - a)} \qquad (A10.7)$$

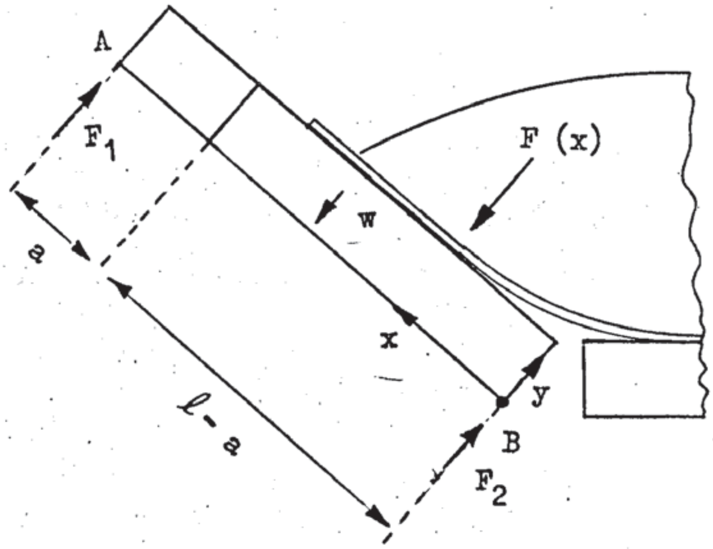

FIGURE A10.1

Forces on side roller of idler set with greater than standard side roller length

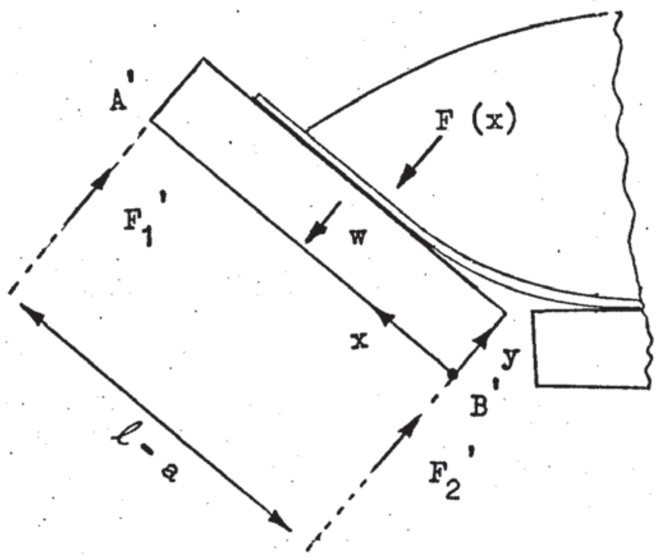

FIGURE A10.2

Forces on side roller of idler set with standard side roller length

Thus, F_1' and F_2' can be found from measured quantities, using equation A10.6 and A10.7.

In practice it was found that the outputs from the load washers used on the test rig were not sufficiently accurate to warrant full analysis. However, if a more suitable measurement arrangement is used in the future as suggested in Chapter 8, the formulae derived in this appendix can be used.

APPENDIX 11

Development of a strain transducer suitable for use on rubber/fabric conveyor belts

An experimental investigation, as described in Section 7.6, demonstrated that direct measurement of the deformation of a grid drawn on a belt was not suitable for the more elaborate strain measurement work proposed for the investigation of transverse load support. As, at the time, it was anticipated that belt load support capability could be related to the transverse and longitudinal strains occuring, it was necessary to develop another method by which strains in rubber/fabric belts could be measured. A review of other belting researchers' work was carried out to see if any method developed previously would prove to be suitable. Some of the work has been mentioned in the scientific literature review of Chapter 2 and it was found that the methods developed could be divided into two categories - those used on small belt samples and those concerned with large scale work. More complete details of the investigations carried out are given below according to these two categories.

Strain Measurement in Small Belt Samples

When small tensile test belt samples are being studied it is relatively simple to attach straightforward mechanical extensometers for direct strain measurement. These devices have been used by Frenzel and Rothe [27] and by Vierling and Scheele [121] . Compressive strains are more difficult to measure because of the bending and distortion that will occur in most belt samples when compressed. Vierling and Scheele tried to overcome this problem by making a "block" of belting by bonding together several layers of one type of belt. A mechanical extensometer was then fitted to the side of the block to measure compressive strains. It is, however, unlikely that this would overcome the problem completely and the cross-section of the block would probably become "barrel-shaped" when compressed so that results would be difficult to interpret.

Rothe [122] has discussed some of the disadvantages of drawing a lattice on a sample and has tried to develop a technique to overcome the problems. The improved method was based on the "brittle lacquer" technique which is in common use for measurement of strains in metals. (In this method the sample is coated with a thin layer of lacquer which cracks when the sample is strained. The amount of crazing gives a measure of the strain occuring.) Rothe was not successful when using actual lacquers because of the haphazard crazing that occurred, but he went on to develop the "paper-screen" technique in which a piece of thin photographic paper was stuck to the surface of the belt sample. Cuts were then made through the paper and into the rubber belt covers, thus effectively forming a number of small pieces of paper covering almost the entire sample surface. If tension was applied to the belt, black lines appeared between the pieces of white paper and the width of the black area at any point gave a measure of the strain occurring there. This technique could not be used in compression. Rothe gave three conditions which must be met by any strain measurement method for belting if a technique based on the brittle lacquer method is to be used:-

(a) The cracks must not be allowed to appear haphazardly.

(b) The "lacquer" (formed by the paper in the work described above) must be stiff enough to ensure that localised length changes will be visible. (The rubber surface directly under the lacquer in the method above is effectively made inextensible by the paper covering it, and so extension only occurs in the cracks.)

(c) In unextended sample conditions the black portion should be almost zero.

The last two conditions are intended to ensure that the strain measured approaches the strain at a point and not the average strain measured over a relatively large distance by most mechanical extensometers.

All the work described above was carried out in simple situations where the properties of small belt samples were being studied. The measurement methods are not suitable for more complex work, particularly where bending and compression can occur.

Strain Measurement in Large Scale Work

Vierling and Scheele [121] extended their studies to carry out some work on belts bending around a drum. They used inductance transducers fitted between the plies of the belts but did not give very detailed information about their calibration techniques. Far more information is available from the National Coal Board's published research [55]. In this work, foil strain gauges were placed inside small pockets left between the plies of the belt sample during manufacture. The gauges used were manufactured by Huggenberger Ltd., and "Evostik", an epoxy adhesive, was used for bonding. The length of the gauges used was not stated in the report but the pockets in which they were fitted were 75 mm long, and so it is probable that the gauges were in the order of forty millimetres long. Details of the method of obtaining the calibration curve have also been given. Strains occurring in each ply were measured as the belt passed around a pulley representing the drive drum of a conveyor. Similar work has been carried out by Dunlop [123] using paper backed strain gauges fifty millimetres long.

A different sort of strain gauge specially developed for use in rubber has been used more recently [124]. These gauges were made by winding wire around a narrow rubber cylinder. The gauges were then bonded to the top and bottom plies of a belt sample to be tested for its impact resistance. A strain limit of 30% was claimed for the gauges, but the gauge sensitivity was found to change with rate of loading and so great care was required to ensure that the gauges were calibrated at the correct loading rate.

The strain measurement techniques described above for large scale test work all required specially prepared belt samples - a situation that was not desirable for the proposed test work on transverse load support. Therefore, none of the methods used previously for conveyor belting proved to be suitable for the task envisaged and so it was necessary to consider other techniques of measuring strain distribution.

Selection of Testing Method

Ideally, a strain sensing device to be used on any large scale conveyor belt investigation should fulfil all the following conditions:-

(i) It should be capable of measuring at least 20% extension and compression.

(ii) It should be flexible enough to take up the radii of curvature that are likely to occur.

(iii) It should be simple to apply without having to prepare special belt samples.

(iv) It should measure the strain over a small length (at a point preferably) so that average strains over a large area are not recorded.

(v) It should be accurate to within acceptable limits, which for the present work are $\pm$ 0.1% strain, and the output should be reproducible over several strain cycles.

(vi) It should be reasonably cheap, and, if possible, re-usable.

(vii) It should not affect the strain actually occurring in the sample.

(viii) It should be capable of withstanding any adverse physical conditions that might occur during the experiments.

It was unlikely that a perfect device would be found, but it was hoped that one meeting most of the requirements could be developed.

One of the simplest and cheapest forms of strain transducer is the electrical resistance gauge commonly used for strain measurement on metal structures where strains are usually less than one to two percent. However, in recent years there has been a great deal of improvement in gauge manufacture and it is now possible to purchase high-elongation foil gauges capable of extending to 20% strain. As such gauges fulfil most of the conditions set out above (provided a protective coating is applied to the gauge), some preliminary experiments were carried out to see if they were indeed suitable for the large scale work envisaged here. It was expected that the gauge stiffness might affect the strain in the sample (see conditions (vii)), but it was hoped that the discrepancy between the recorded gauge strain and the strain that would occur without a gauge would be negligible compared to the large strains that would actually be present. Work was therefore started to see if the gauges could be calibrated for use in large scale experimental work and the discussion below describes the method and results of this investigation.

Evaluation of High Elongation Foil

Gauges bonded directly to belt surface

It was decided that the foil gauges to be used in the investigation would be those manufactured by Micro-Measurements Limited coded

EP-08-250BG-120. The EP designation meant that the foil was formed from special annealed constantan and the backing was flexible high-elongation polyimide. The overall gauge length including the backing was 12.5 mm. A strain limit of $\pm$ 20% was claimed for these gauges and their nominal resistance was 120 ohms. Similar gauges manufactured by the same company had been evaluated by Krempl on metals under conditions of cyclic plastic strains [125]. The results showed that there was some zero shift at the strain levels investigated (up to about 3%) and that the fatigue life of the gauge was severely affected at these levels. The gauges used in Krempl's work had a much smaller gauge length than those used in the present work and such high strain levels as $\pm$ 20% had not been claimed for them. Also, there had been several advances in strain gauge technology since Krempl's work and so it was hoped that the proposed experiments would give better results.

The adhesive recommended by the manufacturer for the gauges to give optimum performance required a high temperature cure. It was not, therefore, thought suitable for this work, as it would be very difficult to apply the gauges on large belt samples and there would also be the possibility that the high temperature required to cure the adhesive might affect the local physical properties of the belt. It was decided that the most suitable cold-cure adhesive, "Micro-Measurement M-Bond 200" (a methyl-2-cyanoacrylate adhesive), should be used. A fresh application of this adhesive was claimed to have an elongation capability greater than 15% [126].

Initial experiments were carried out using this adhesive to demonstrate whether or not the foil gauges were suitable for the task proposed [127]. The method of applying the gauge to the specimen was as advised by the manufacturers [128]. A "Wheatstone Bridge Circuit" was used for the work in which two arms of the bridge were strain gauges; one of which was the active gauge on the belt sample tested and the other was a dummy gauge on an unstrained belt sample to provide compensation for changes in ambient temperature. All output signals were recorded on a digital voltmeter.

A calibration technique was devised which involved extending the gauged sample in a Hounsfield tensometer. A travelling microscope was positioned over the gauge area so that the extension of grid lines could be measured. (See Figure A11.1 and A11.2.) Great care was taken to ensure that grid measurements and circuit output readings were taken, as near to simultaneously as possible to minimise the effects of any time dependent stress/strain behaviour [127]. In any case, it was found that if, after increasing the sample strain, two minutes were allowed to elapse before taking measurement, no significant changes occurred in the circuit output.

Typical results obtained from this work are shown in Figure A11.3. The lines drawn on this figure are only estimates of the "best-fit" curves through the measured points; they are not mathematically derived curves.

The sample to which Figure A11.3 refers was taken to a fairly high measured strain level (12%) and it can be seen that the strain circuit output at a given strain on the curve where tension was being increased was not the same as that obtained at the same strain on the curve of decreasing tension. It also appeared that there was some zero-shift occurring with the circuit output but this was not certain because of the residual strain that was left in the belt sample. (It is not

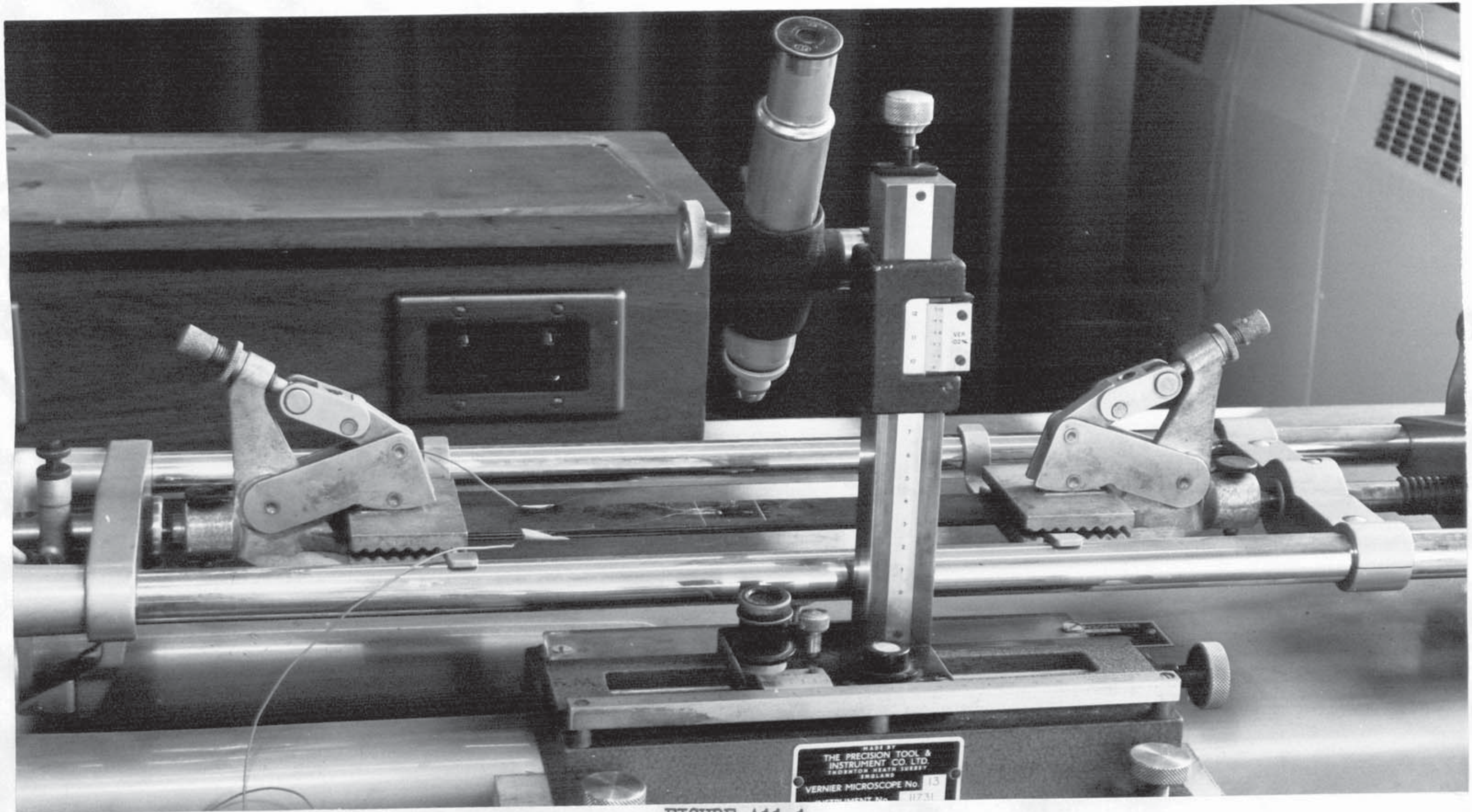

FIGURE A11.1

Apparatus for calibration of high elongation strain gauges bonded to a conveyor belt sample

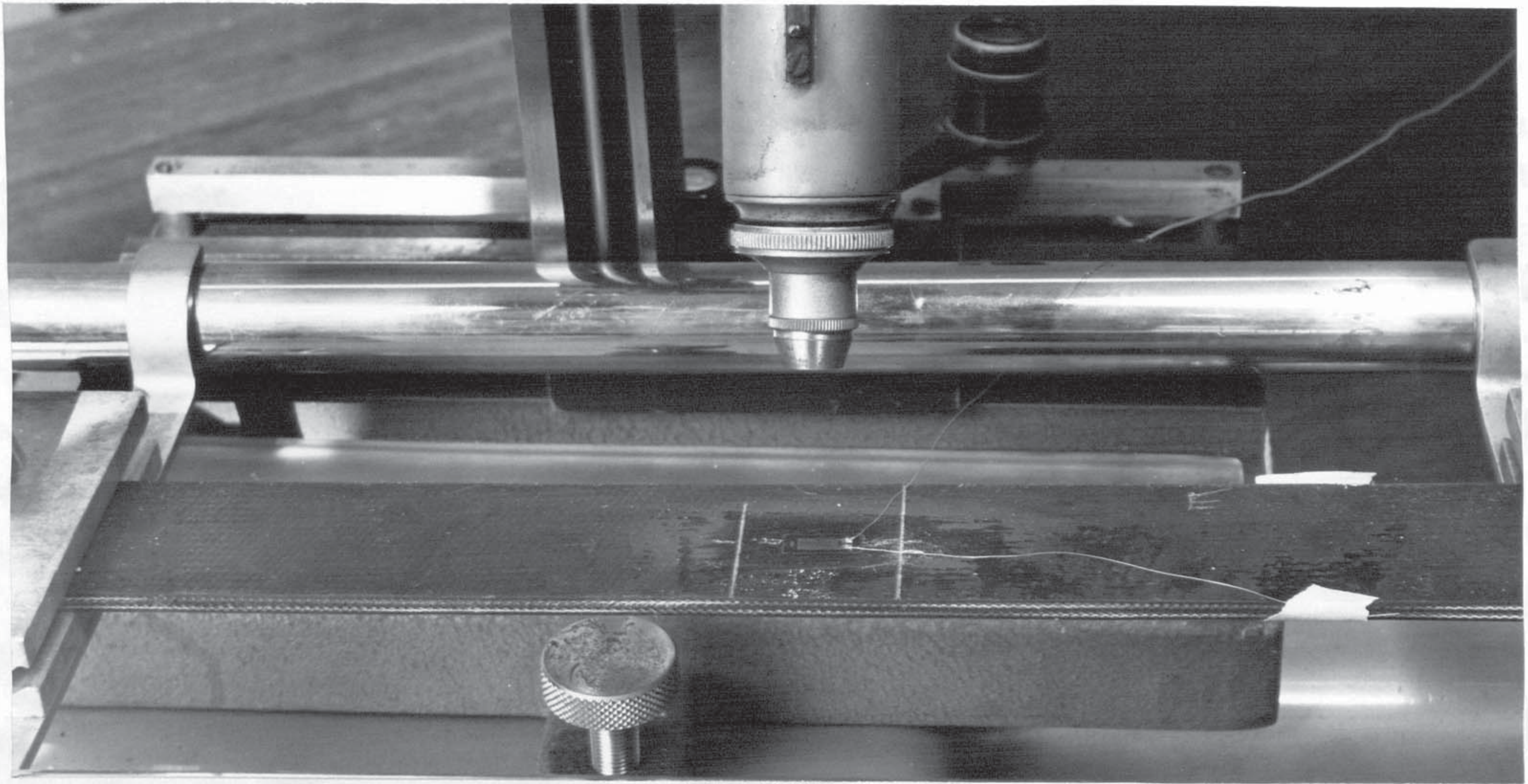

FIGURE A11.2

Close-up of high elongation strain gauge bonded to conveyor belt sample

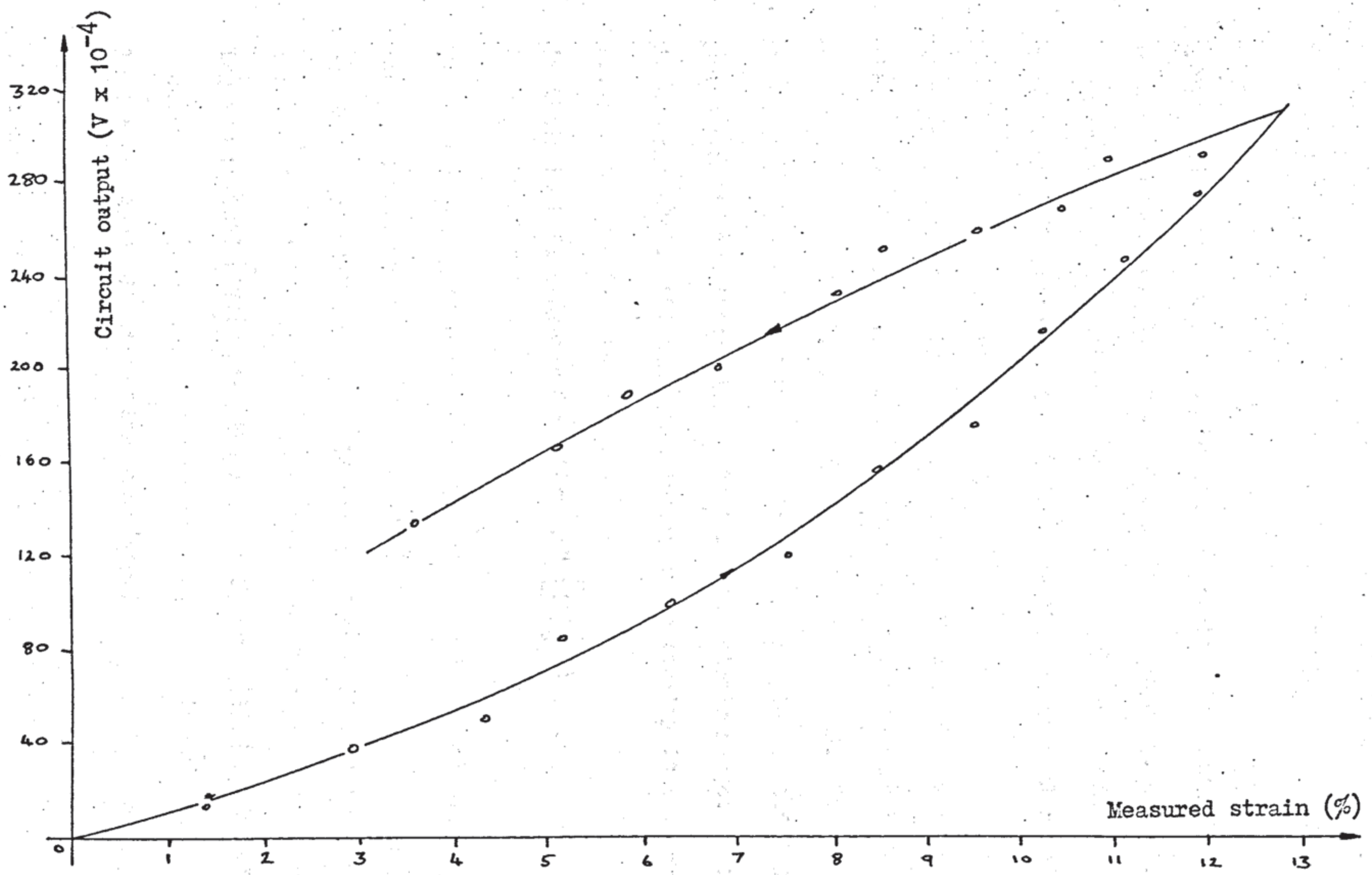

FIGURE A11.3

Typical calibration curve obtained from high elongation strain gauges bonded to belt surface usually possible to make a belt sample return to a zero strain position after having been under tension and so the initial position could not be repeated during the test work.) It was thought that these observed phenomena might occur because of the high strain level and so another gauged belt sample was taken to only 5% strain. However, the same sort of result occurred. It was found that the strain gauge circuit output was more reproducible over a second strain cycle and so it was thought that it might be possible to take a gauge through several strain cycles and then achieve an acceptable reproducible calibration curve. However, this was not the case because it was found that if this was done and the sample was then allowed to recover over several hours before re-testing, the original pattern of results was obtained. It was also found that the output of the circuit for the samples tested did not correspond to the theoretical output for the strain gauges at the strain levels measured. (Strain gauge theory and circuitry is reviewed in several text books. For example, Perry and Lissner cover this subject in their book "The Strain Gauge Primer" [129].)

It was thought that, because of the points mentioned above, the adhesive being used was possibly not providing a good bond between the strain gauge and the rubber belt cover, and so different types of adhesive were tested and an improved method of surface preparation for bonding was devised [127] using the work of Sikorra [130] as a basis for development. (The initial strain gauge experiments had, at least, lead to a standardised testing procedure.)

The results obtained from this further work using different adhesives (Bostik 2402, a two part cold curing isocyanate adhesive and Bostik 3206, a polyurethane adhesive) exhibited better reproducibility over more than one strain cycle. (An example is shown in Figure A11.4 for a gauge bonded with Bostik 3206.) However, in all cases, the circuit outputs did not approach the theoretical values that should have been obtained if the gauges were being strained to the levels recorded by grid measurement. In fact, they were sometimes an order of magnitude less than those that, from theory, should have occurred.

In order to check that the circuit apart from the active strain gauge was obeying the theoretical formula, a variable resistance box was placed in the circuit in place of the gauge and adjusted to the strain gauge resistance, 120 ohms. The circuit was balanced so that the output was zero and then the resistance was changed by a known amount corresponding to a certain resistance change that should occur in a strain gauge when extended to a given amount. The output value confirmed that, provided the active gauge was within the manufacturers specifications, the strain gauges were not giving out the signal corresponding to the strain measured on the sample. The results obtained from placing the known resistance in the circuit are shown in Figure A11.5 and this shows that the output from the circuit with the gauge is about one tenth of the value obtained at an "equivalent strain" with the resistance box. Therefore, the gauge could not have been at the strain measured on the sample, and so the gauge must have been causing a local stiffening effect on the sample surface.

Thus, the following conclusions had to be drawn:-

A straightforward application of high elongation foil gauges to the rubber cover of a belt is not suitable for investigating the surface strain because of the large stiffening effect that they cause. A

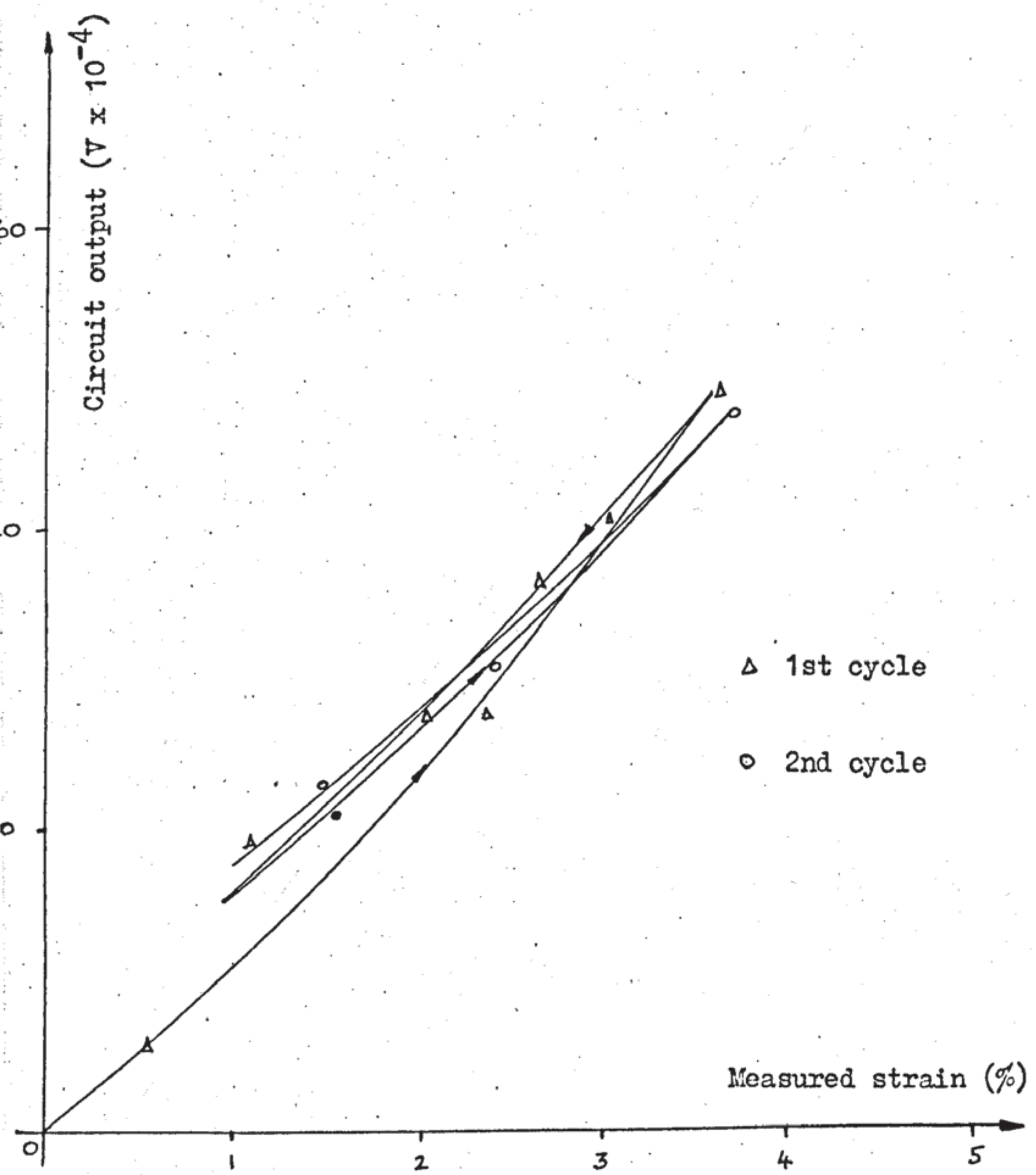

FIGURE A11.4

Calibration curve over two cycles for strain gauge bonded with "Bostik 3206"

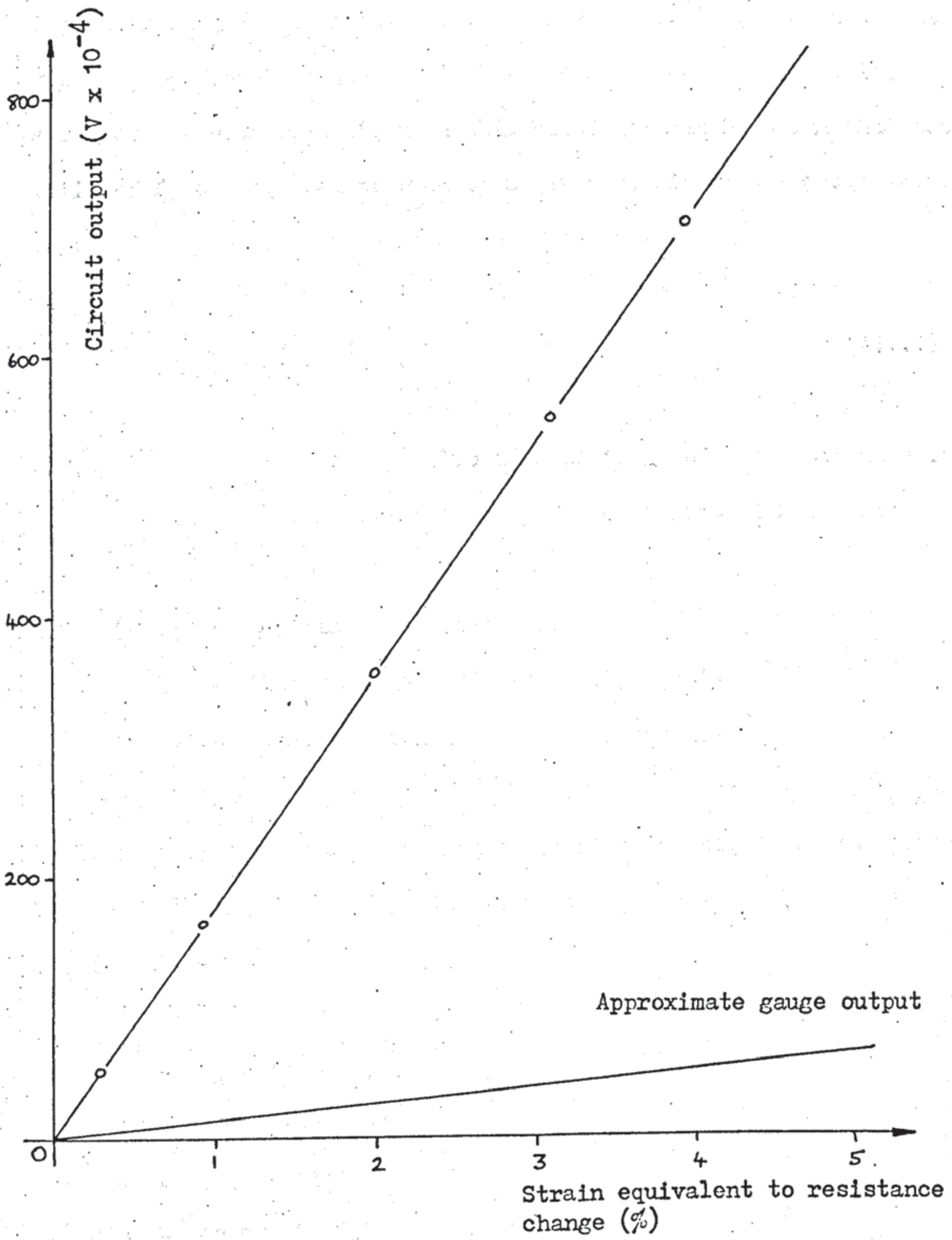

FIGURE A11.5

Observed circuit output versus strain equivalent to resistance change giving that output

fairly simple analysis technique used by Dove [131] demonstrates the order of magnitude of the reinforcing effect that can occur due to bonding a strain gauge on a low modulus surface. If it is assumed that the strain is constant throughout the sample cross-section (that is, gauge + rubber cover cross-section in this case), it can be shown that the ratio of force required to produce a given strain with a gauge in place to the force required when no gauge is present is

$$R = \frac{(E\,A)\,g}{E\,s\,A\,S} + 1 \qquad \text{(A11.1)}$$

Where E = Effective modulus of elasticity of the whole gauge.

A = Cross-sectional area of the equivalent homogenous gauge.

so (EA)g = Stiffness of the gauge.

Es = Modulus of elasticity of the sample.

As = Cross-sectional area of the sample.

R is a measure of the stiffening or reinforcing effect of the gauge. If the following values, applicable to the present work, are put into the formula:-

$$E = 35 \times 10^2 \ \mathrm{N/mm^2}$$

$$A = 1.2 \times 10^{-1} \ \mathrm{mm^2}$$

$$Es = 17.5 \times 10^{-1} \ \mathrm{N/mm^2}$$

$$As = 52.5 \ \mathrm{mm^2}$$

Then $R = \dfrac{35 \times 1.2}{17.5 \times 0.525} + 1$

that is,

$$R \simeq 6$$

The situation assumed for Dove's Analysis is shown in Figure A11.6 and it can be seen that ΔL, the increase in sample length is assumed to be constant throughout the sample cross-section. The situation actually occurring is more likely to be as shown in Figure A11.7.

Thus, the actual situation is more complicated than that assumed by Dove; nevertheless, using his formula shows that there is a large reinforcing effect from the gauge because of the large difference in moduli (both elastic and shear) between rubber and gauge. (Young's modulus of rubber at low extension is approximately 1.8 N/mm^2 whereas that of the gauge is in the order of 0.35 N/mm^2.) Dove suggests that a new gauge factor could be experimentally defined for work with gauges on plastics; that is, a gauge calibration could be carried out in order to find the effective gauge factor for the gauge on the plastic. This would probably prove adequate provided the application is simple (for example, in straightforward tensile tests) and the experimentally defined gauge factor is not very much different from the manufacturers quoted value. However, neither of these conditions was met in the present work and so it was concluded that high elongation foil gauges bonded directly to the rubber belt cover were not suitable for measurement of the complicated strain field occurring in the large belt samples being tested. They are not suitable because the large stiffening membrane effect that they have on the rubber will distort the strain distribution.

Some final, more controlled, experiments were carried out to:-

(a) See if reducing the cover thickness had any effect on the circuit output.

(b) Verify that the gauge was causing a local stiffening effect on the surface of the rubber.

For (a), a two ply belt sample was made with virtually no rubber cover. (Previously, 1.5 mm covers had been used.) A gauge was bonded to the sample surface using the application technique devised for the earlier work described above. It was hoped that, as the gauge was much nearer the fabric load carrying members of the belt, it would take up a much greater strain than the previous gauges. However, it was found that the circuit output values were still in the same order as in the earlier experiments.

For (b), two exactly similar belt samples were prepared. A reject gauge* was bonded to one and three "boxes" were drawn on each as shown in Figure A11.8. Two experimental runs were carried out with the samples and the strain in each "box" was measured. The average measured strain in the outer boxes was then plotted agaist the strain in the centre box, and this curve should have been the same for both samples provided that the gauge was not affecting the strain in the piece of belt that it was bonded to. The results are shown in Figure A11.9. This method of analysis was adequate to show the order of magnitude of the stiffening effect of the gauge. The strain in the centre box with the gauge was so small that it was in the order of the measurement errors.

*Such gauges, rejected for minor faults, cannot be used for strain measurement, but they demonstrate all the usual mechanical characteristics required here.

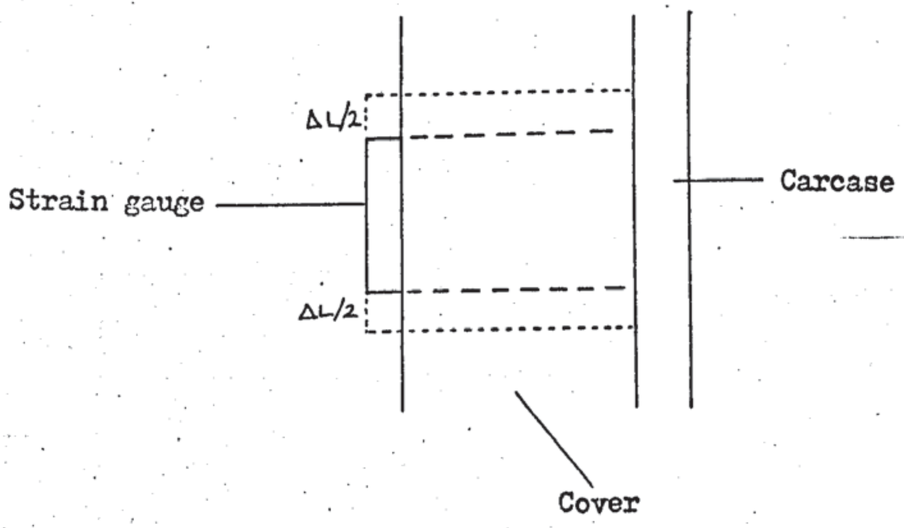

FIGURE A11.6

Strain distribution in strain gauge and belt cover assumed in analysis by Dove

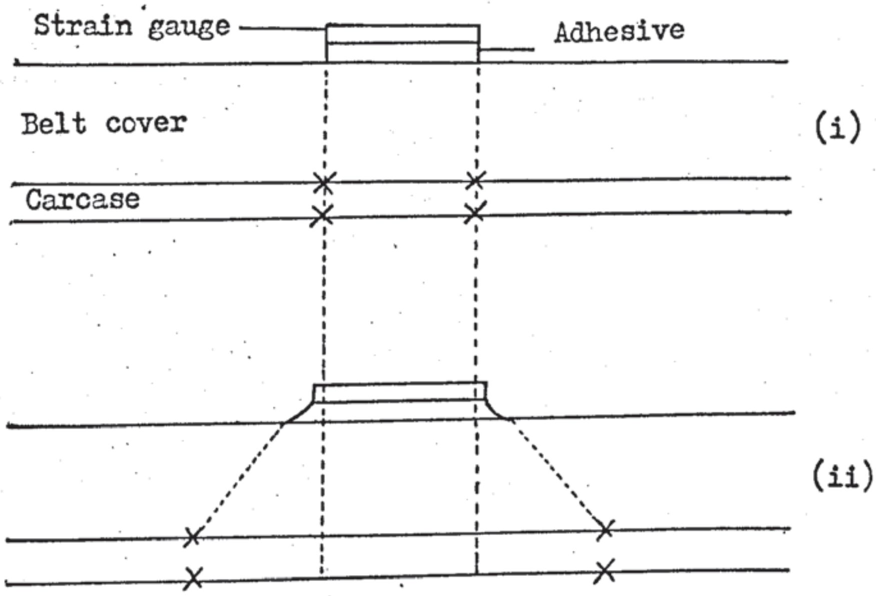

FIGURE A11.7

Probable strain distribution actually occurring with a strain gauge bonded to belt surface

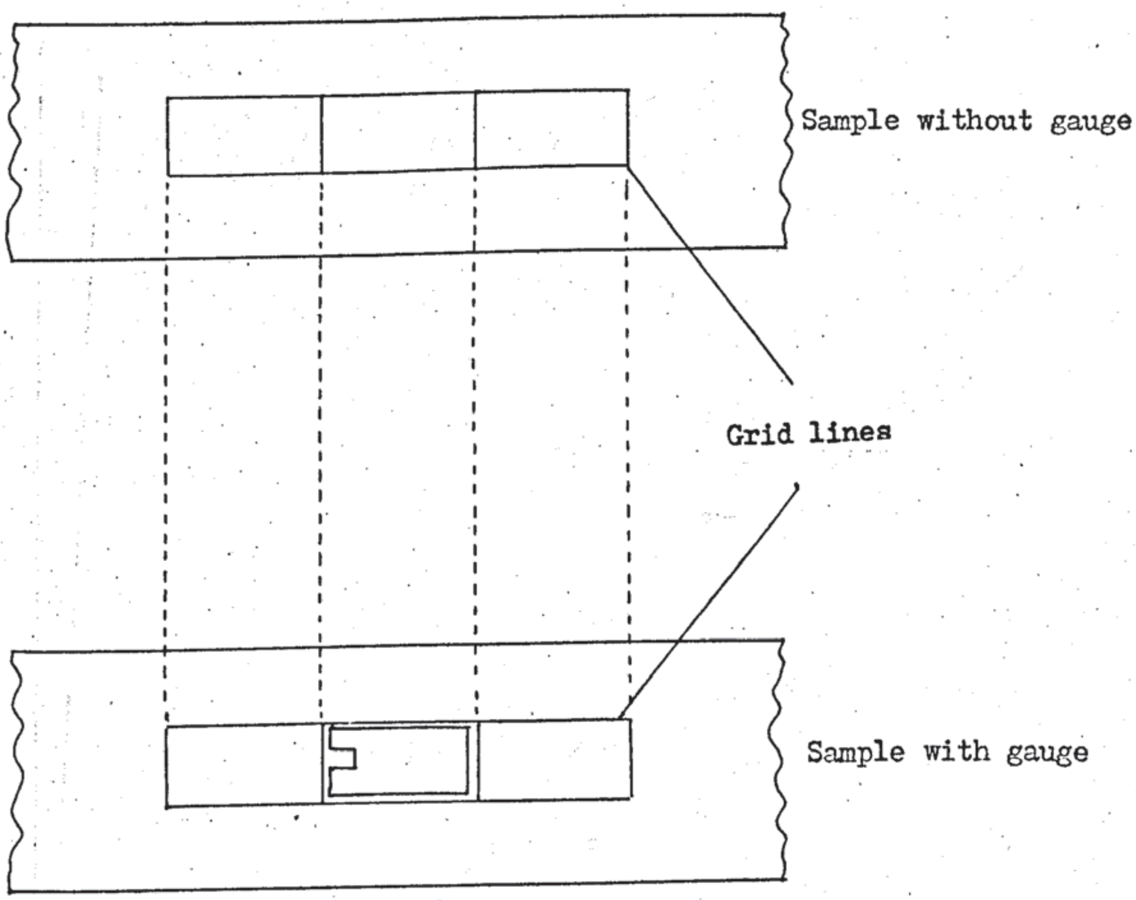

FIGURE A11.8

Grid lines drawn on belt samples to establish stiffening effect caused by strain gauge

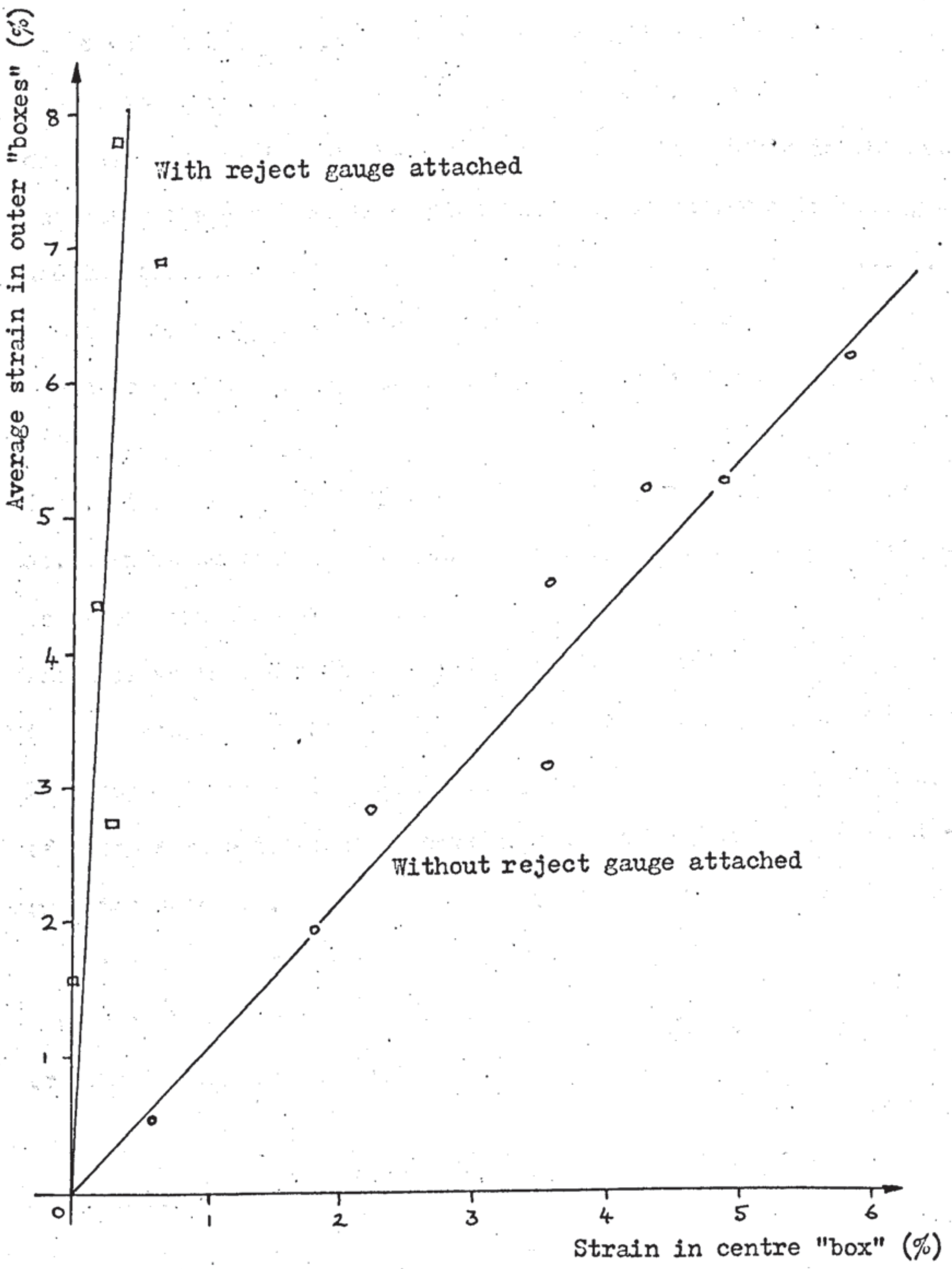

FIGURE A11.9

Results of study to demonstrate stiffening effect of
a strain gauge bonded to the belt surface

Further evidence that the presence of the strain gauge was affecting the strain in the belt sample was obtained from the shape of the strain gauge output curve; as rubber is extended, the modulus increases (this is why it is usual to note the strain range when an elastic modulus is quoted), and so it would be expected that as the strain in the rubber increases the gauge would begin to have a decreasing reinforcing effect and the circuit output would be shaped as shown in Figure A11.10. The recorded curves (with one exception in which the adhesive bond was shown to be faulty) suggested that this shape was being followed.

Thus, it was found that directly bonded foil strain gauges were unsuitable for measurement of strains in a complex strain distribution and so other strain sensing devices had to be considered. It has been shown above that techniques used previously with conveyor belts were thought not to be suitable for the proposed investigation of transverse load support and so it was necessary to investigate the possibilities of using strain transducers developed in other fabric reinforced elastomer technologies. The tyre industry was found to be a useful source of information because it is a relatively high technology dealing with a product basically similar to belting. A review of the strain measurement techniques used in this industry was therefore carried out.

Strain Transducers used on Tyres

In recent years there have been some highly sophisticated methods devised for measuring strains in tyres; these are not the sort of devices that are required, at present, for belt studies, so this review only deals with the more simple techniques that have been used. Kern [132] provides a useful survey of some types of strain transducers including:-

(a) Rubber strips with electrically conductive parts.

A small rubber tube is filled with conductive material such as mercury. Electrodes are fitted to the ends of the tube and resistance of the gauge changes with strain. These are basically the same as liquid metal gauges used by Sikorra [130]. With these gauges the change of resistance is not directly proportional to strain and so they must be calibrated carefully: Kern states that reproducibility and calibration are difficult. A diagram of this sort of gauge is shown in Figure A11.11.

(b) Electric inductance gauges.

Two densely wound wire coils are embedded in the rubber. When the rubber is deformed the coupling between the coils changes. These gauges (the principle of which, was used for Vierling and Scheele's work mentioned at the beginning of this appendix) are relatively insensitive and are complicated to design for optimum results. If such gauges were to be used in any test work, it would probably be necessary to consider the effects from any surrounding machinery or metal structures.

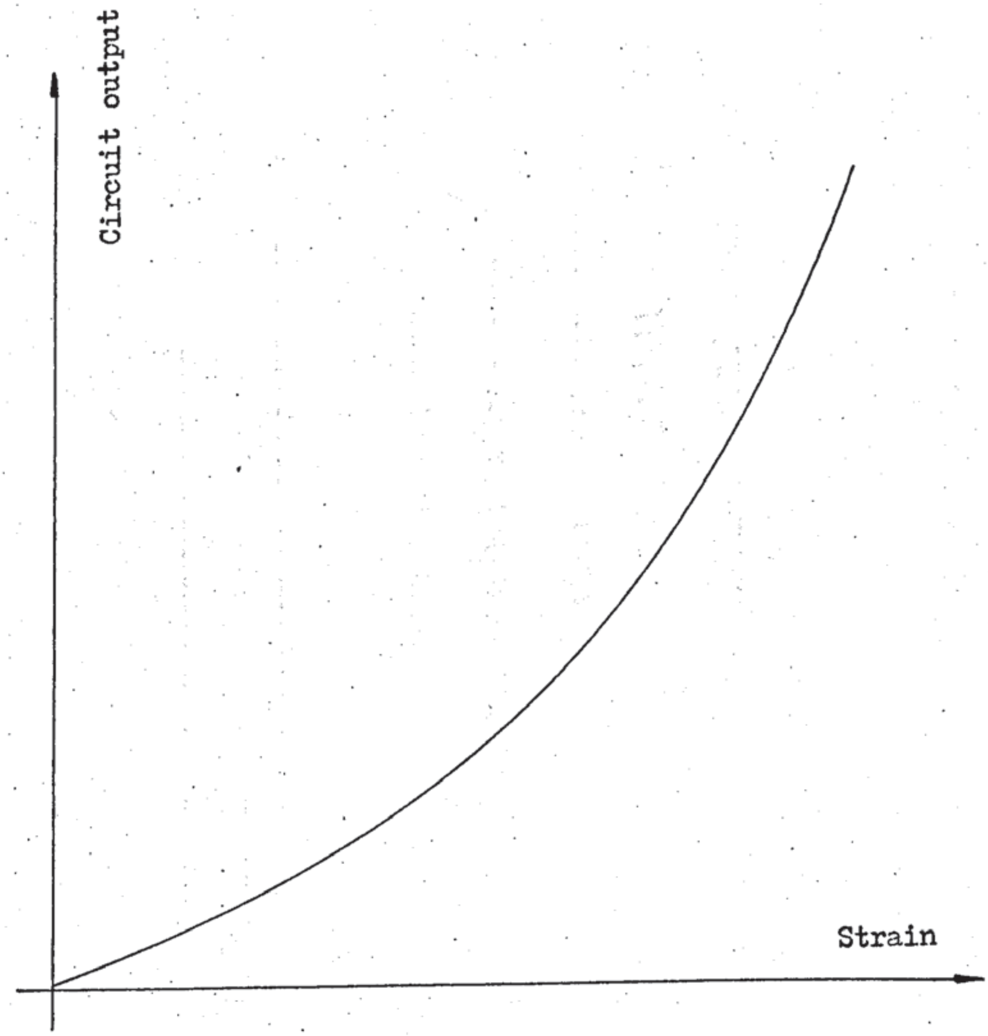

FIGURE A11.10

Expected shape of a plot of strain gauge circuit output versus strain level in belt sample

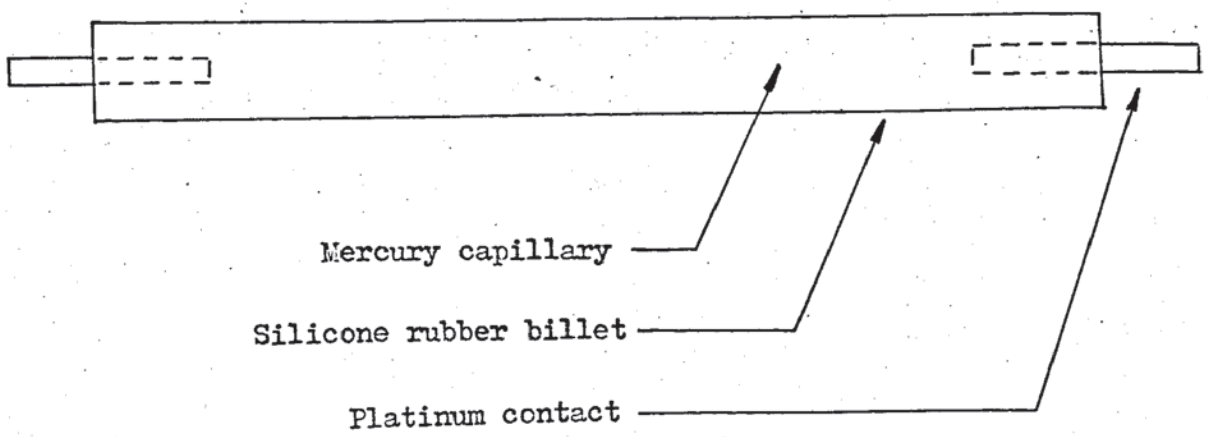

FIGURE A11.11

Liquid metal strain gauge

(c) Clip Gauges.

For these devices, normal strain gauges are bonded to both sides of a small shaped piece of thin steel. The two ends of the steel are connected to pins which are pushed into the rubber. These transducers then have a low effective modulus when extended or compressed and the strains acting in the actual gauges are within their measuring capabilities. A diagram of this sort of gauge is shown in Figure A11.12.

Advantages of these include good reproducibility and calibration; they are insensitive to temperature because two strain gauges are bonded to the steel and connected into a compensating circuit. However, they are sensitive to pressure and no loadings can be placed directly on top of them. Grosch [133] has devised a gauge using the principle of the clip gauge which he has used for strain measurement up to 14% extension in tyres. A diagram of the developed clip gauge used by Grosch is shown in Figure A11.13. This design of clip has a reasonably short effective gauge length.

(d) Rubber/Wire Gauges.

A more recent review of testing methods [134] describes one other form of strain gauge that might be used for any conveyor belt work; this is a rubber-wire gauge consisting of a small diamter pre-stressed rubber thread onto which a coil of fine constantan wire has been wound helically. A diagram of such a device is shown in Figure A11.14. It is almost certainly this sort of gauge that was used by Volotkovskii and Nokhrin [124] in the work described at the beginning of this appendix. One disadvantage, that of gauge sensitivity variation, was mentioned then, but others are that temperature compensation is difficult and there are different calibration characteristics in tension and compression. These gauges are not available commercially.

(e) Photographic Technique.

An ingenious strain distribution measurement technique has been used by a major tyre manufacturer: a "Letraset" transfer consisting of small, evenly spaced dots is fixed to the tyre surface; a photograph is taken of the undistorted sample and then strains are imposed in the tyre cross-section; another photograph is then taken and comparison of the lengths between the dots in two photographs give the strains occurring in the sample. This technique could possibly be adapted for use in conveyor belt investigations but sophisticated camera equipment and facilities for accurate analysis of the photographs are ideally required, and these are not usually readily available.

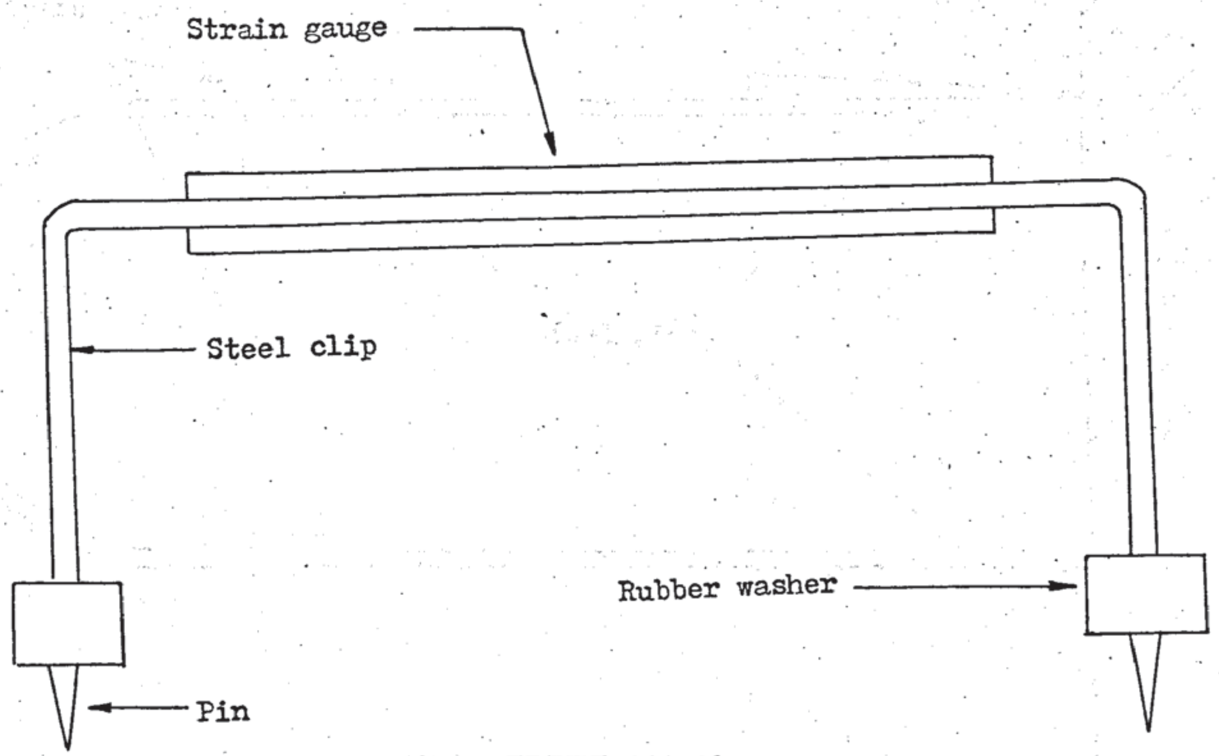

FIGURE A11.12

Clip gauge for measurement of large strains

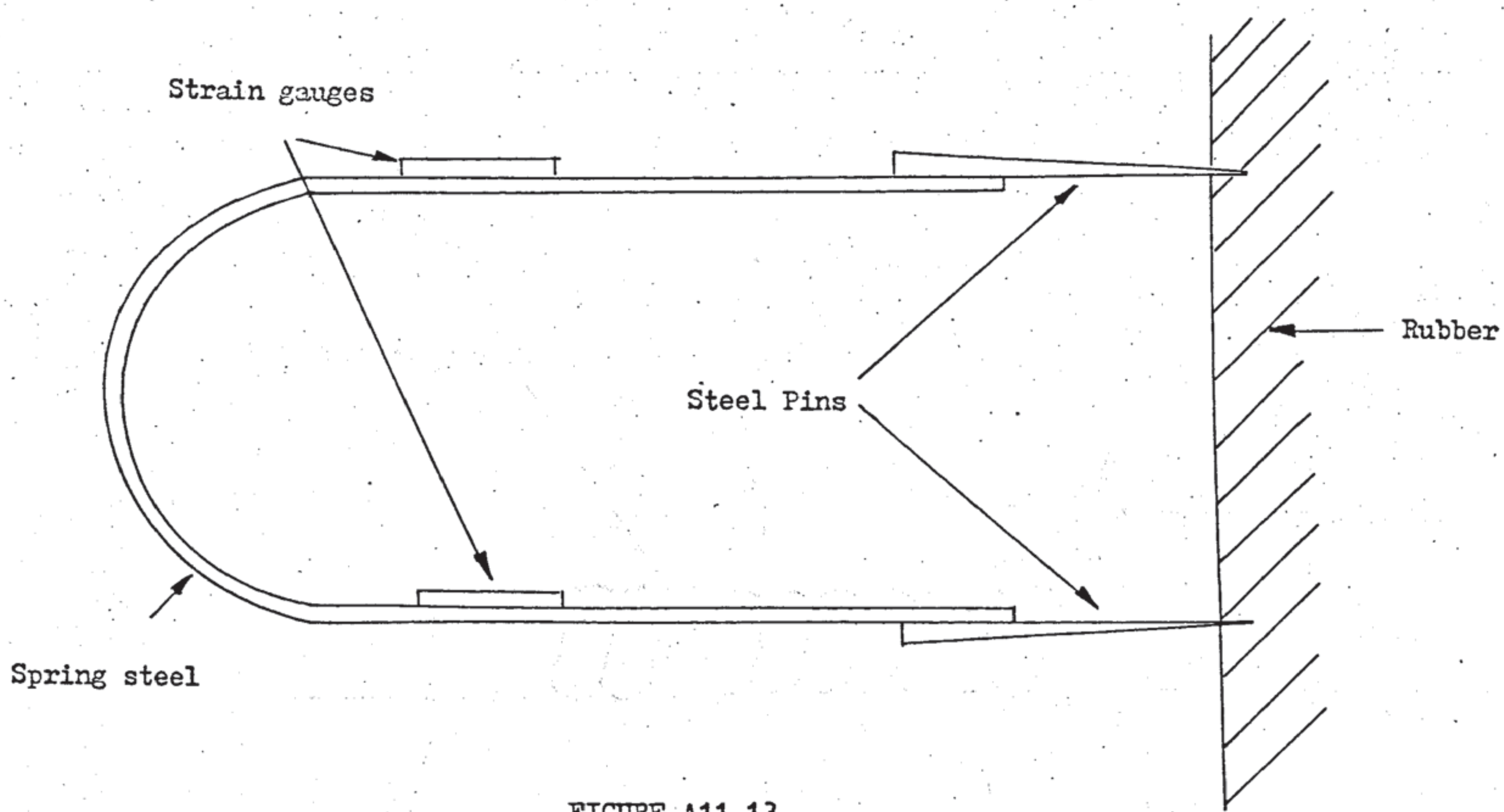

FIGURE A11.13

Clip gauge as used by Grosch for measurement of tyre surface strains

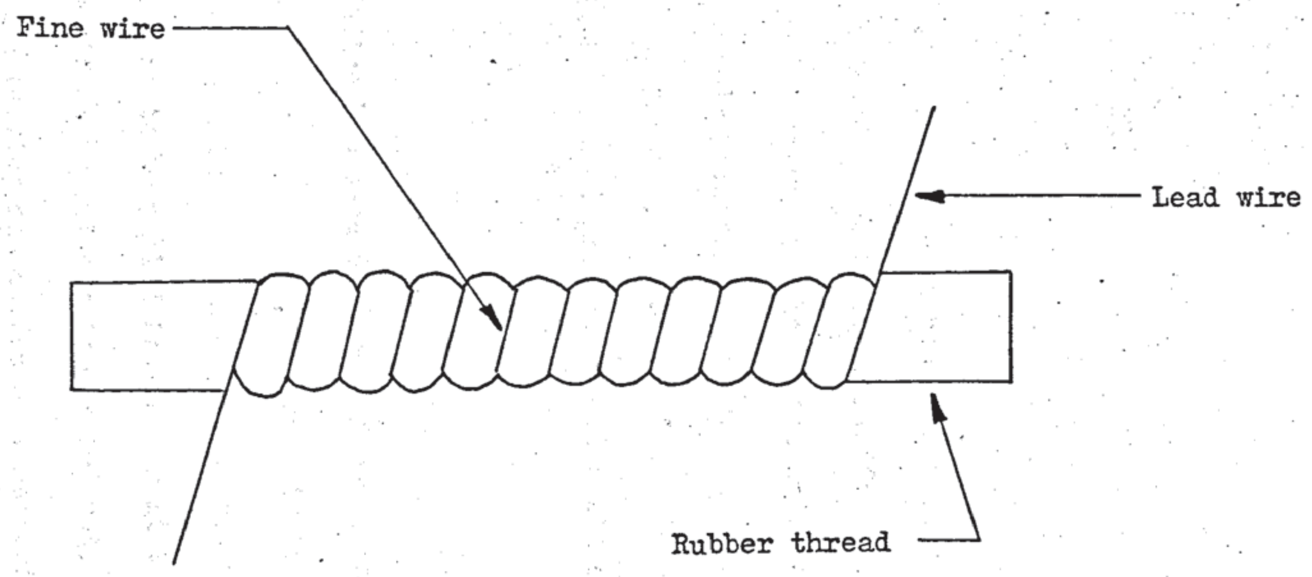

FIGURE A11.14

Rubber/Wire gauge for measuring large deformations

The above methods of strain measurement have all been designed specifically for tyre work. There are also some techniques used in other technologies that would perhaps be suitable for use on belting and these are described below.

Transducers developed in other technologies

A device similar to the clip gauge has been investigated [135] : perspex was used instead of steel to form the "frame" of the gauge as shown in Figure A11.15. The framework "legs" are stiff compared with the beam and terminate in needle points. This device was originally designed for measuring strains on plaster, but if the dimensions of the legs and beam were changed from those used previously, it could easily be used on rubber.

Another simple device has been developed for extension measurements on hovercraft skirt rubbers. In this, a very soft fuse wire is fixed to the rubber at one end and its other end is placed under a rubber "patch" which allows the wire to be pulled from under it but does not allow it to return. Thus, if an experiment is carried out, the maximum extension reached during testing can be found by measuring the new length of fuse wire visible. This method only records the maximum extension, and so is probably not of much general use in conveyor belt studies but could, perhaps, be used for maximum strain measurement at points inaccessible to other devices. For example, the maximum strain occurring exactly at the idler gap could be measured.

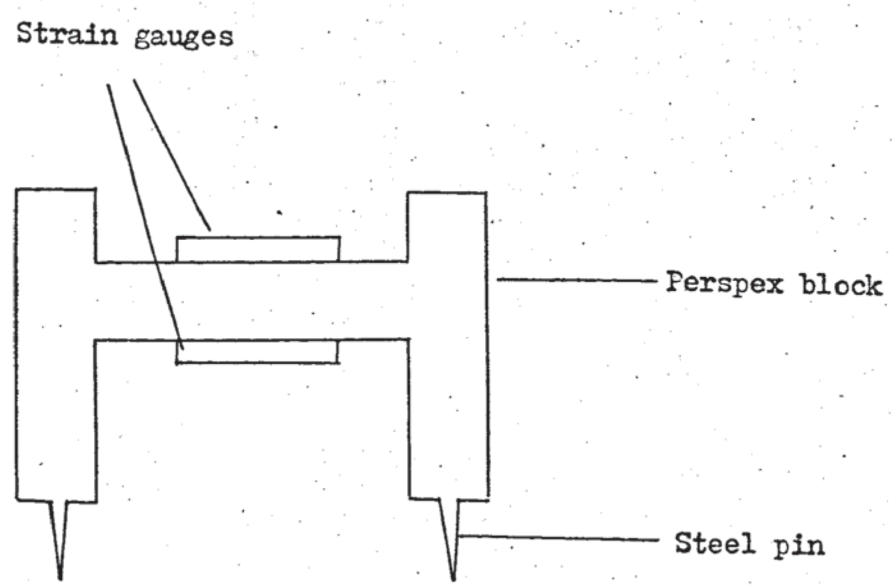

FIGURE A11.15

Portal gauge used for measurement of large strains in concrete

The review of testing techniques described above showed that there were several strain measurement devices that could possibly be developed for use on conveyor belts and it was thought the most suitable of these was the type of clip gauge devised by Grosch. (That is, the U-shaped clip gauge.) These fulfilled all but one of the conditions for a suitable strain transducer as set out near the beginning of this appendix. The one requirement not met was that they would not be able to stand up to any direct loading, and so great care would be required during any experiments to ensure that no load material could touch the transducer.

It was therefore proposed that an effort should be made to produce a clip gauge suitable for use in any conveyor belt strain distribution studies. The work carried out in this connection is described below.

Development of Clip Gauges

Further details of the clip gauges used by Grosch were obtained [136] and a device having the same dimensions as them was made using hardened and tempered spring steel of approximately 6 mm width and 0.2 mm thickness. The method of manufacture of the clip gauge was as follows [137]:-

(a) A strip of the spring steel was cut to the length required for the clip.

(b) The centre portion of the short strip obtained was heated to a dull red colour. (This treatment was required in order to be able to bend the steel into a 'U' shape later in the procedure.)

(c) The strip was cleaned thoroughly according to the instructions given by the strain gauge suppliers.

(d) One strain gauge and a terminal connector was carefully positioned and bonded onto one end of the strip with the recommended adhesive (Micro Measurements M-Bond 200) (see Figure A11.16 (i)). The strain gauges used were high elongation gauges as used in the work described previously; these gauges had no special properties making them unsuitable for this work but would, however, be working greatly below their maximum strain capability.

(e) A protective coating of polyurethane lacquer was applied. A lacquer which did not affect subsequent soldering of lead wires was used.

(f) When the lacquer had dried the steel strip was turned over and another strain gauge was bonded on at the other end (see Figure A11.16 (ii)).

(g) Shouldered pins were fixed in position with an epoxy adhesive. (See Figure A11.16 (iii)). The pins used were cut to size from points intended to be used for pairs of compasses.

(h) The strain gauge lead wires were attached using a link-up system that would prevent any damage to the gauge itself if the wire was accidentally pulled. Screened lead wires were used at all times. (See Figure A11.16 (iv)).

(i) Another protective coating of lacquer was applied.

(j) Finally the clip was bent into the U-shape over a former of the required radius. A photograph of the clip gauge thus produced is shown later in Figure A11.21.

The two strain gauges on the clip were connected into a Wheatstone bridge circuit in such a way that signal magnification was obtained and the strain transducer calibration would be unaffected by temperature variation. The clip gauge and circuit were then tested to see if (a) the transducer caused an unacceptable stiffening effect on the belt surface and (b) an output was obtained such that variations in strain of $\pm$ 0.1% (1000 microstrain) could be detected.

The method used to determine any stiffening effect was that used previously with high elongation foil gauges. It demonstrated that no significant stiffening effect on the rubber was caused by the presence of the clip. (See Figure A11.17.) However, using the method of calibration as before, it was found that, even with using the input voltage giving the highest possible stable circuit output, the output signal was not large enough to measure strains to the required accuracy. An amplifier was connected into the circuit but the calibration curve obtained would still only allow changes of $\pm$ 1% strain to be detected. (See Figure A11.18.) The output from the transducer at a given strain had therefore to be increased. As mentioned previously, the optimum input voltage for the gauges was already being used and if this was

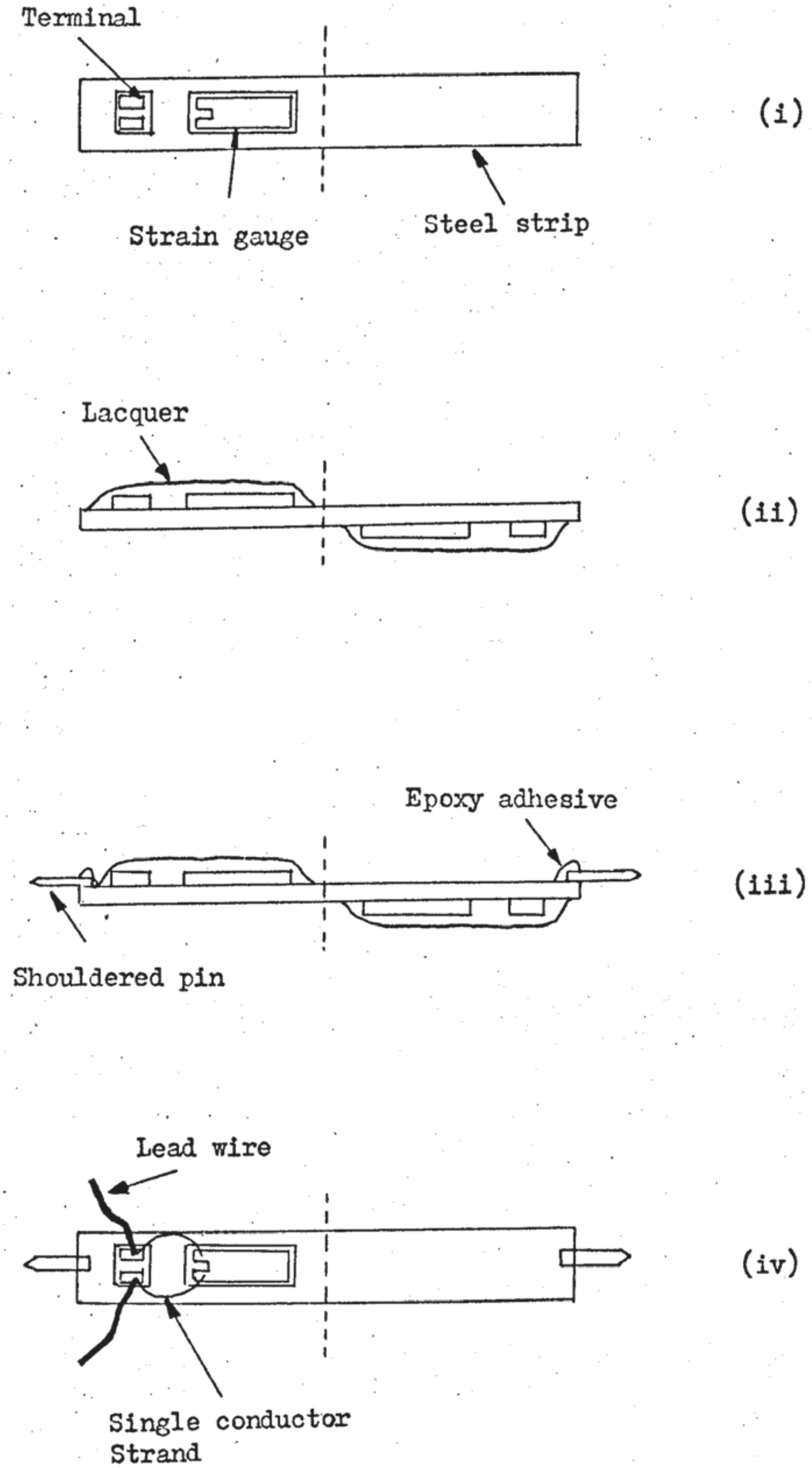

FIGURE A11.16

Manufacturing sequence for clip gauges

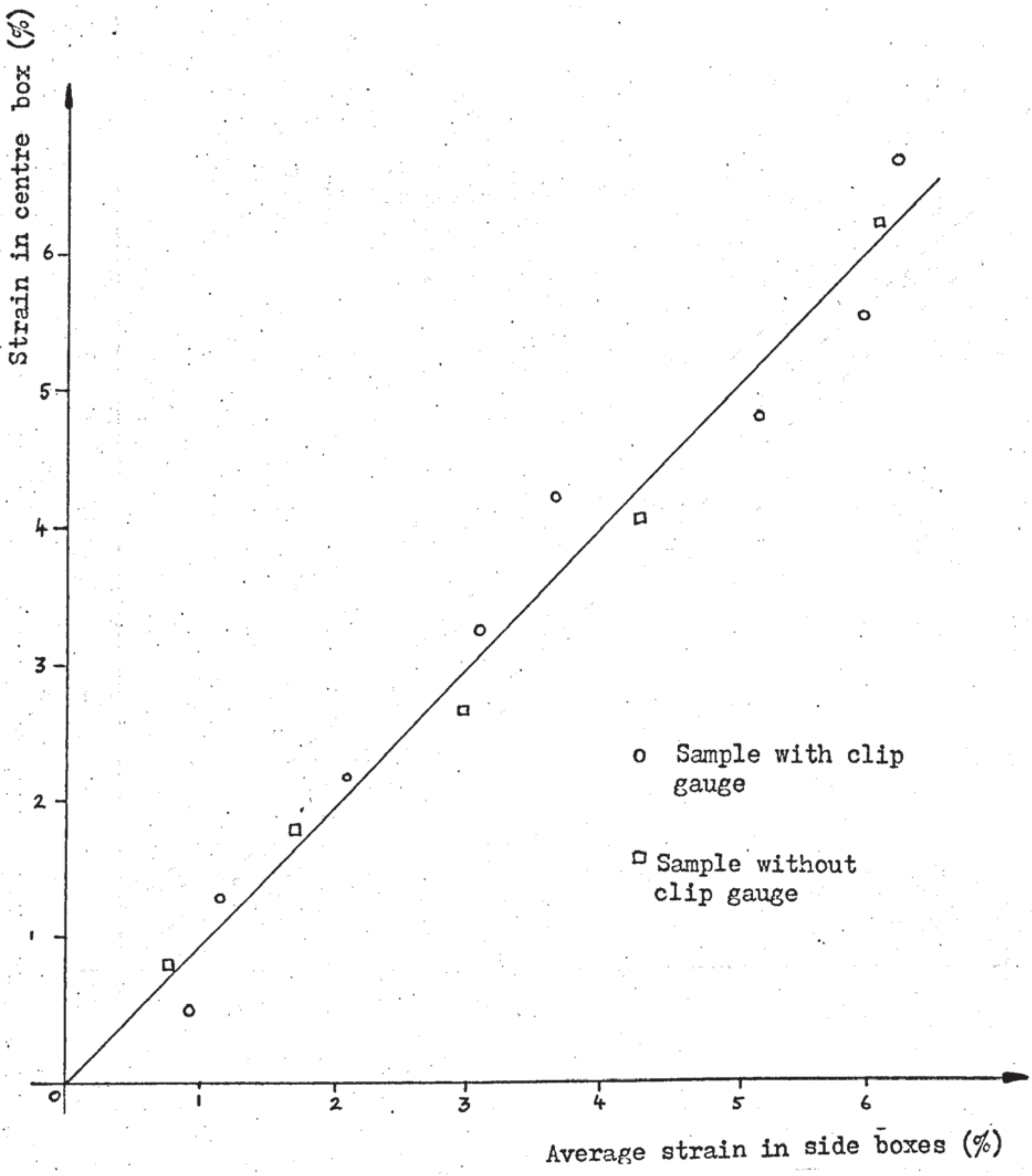

FIGURE A11.17

Demonstration of the negligible stiffening effect of the clip gauge produced

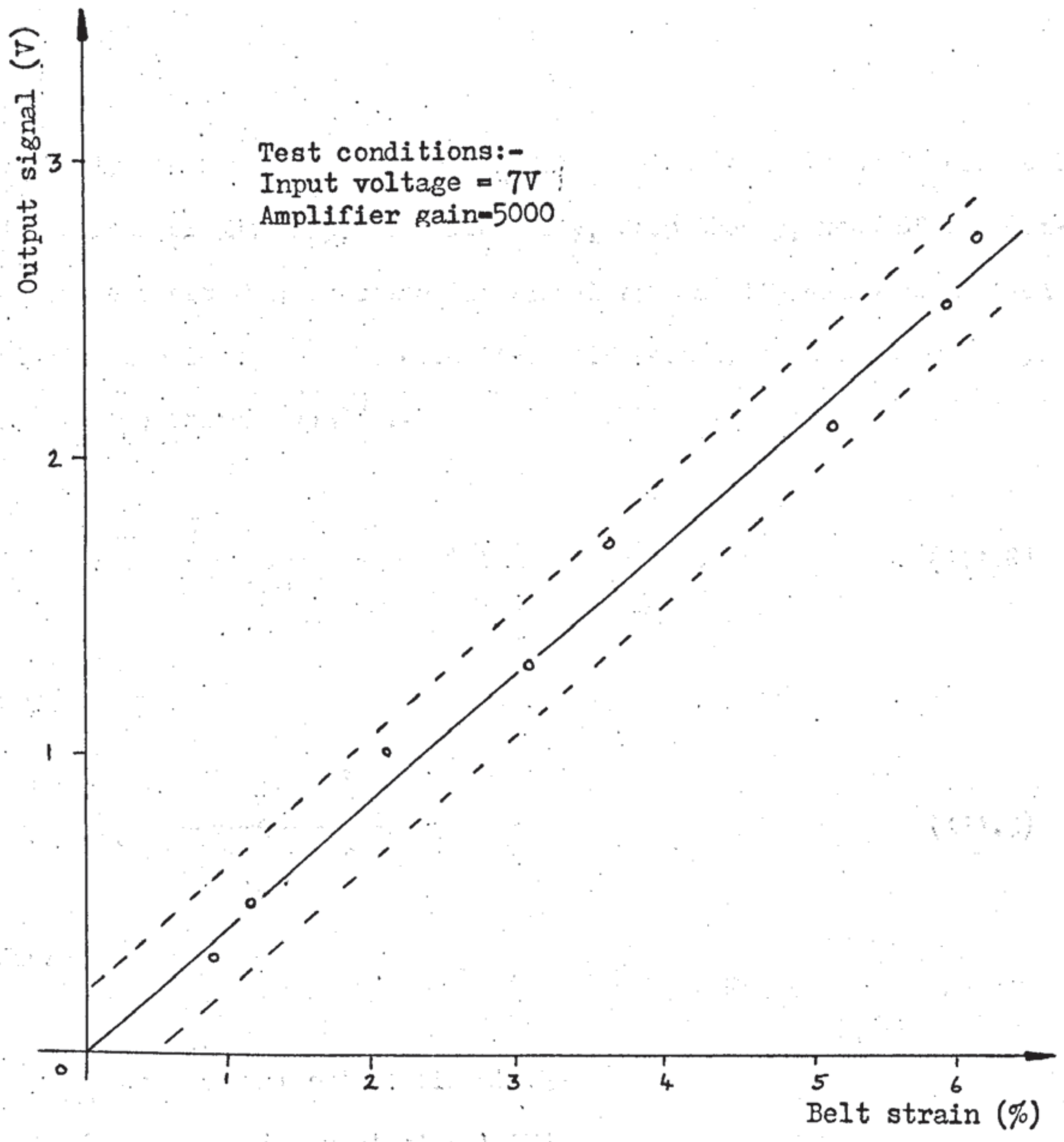

FIGURE A11.18

Calibration curve for first clip gauge produced

Broken lines represent experimental error involved and demonstrate the strain range corresponding to a given output signal.

to be increased, different strain gauges having a higher grid resistance were required.

The only other method of increasing the output signal was to change the dimensions of the clip. Unfortunately, from theory, most of the dimension changes that increase the signal from a clip gauge at a given strain also increase the effective stiffness of the transducer according to the formula [137] :-

$$V_o = \frac{3\ Kw\ \ a\ L_s\ \mathcal{E}_R}{8\ \ L_T{}^3}\ V_I \qquad \text{(A11.2)}$$

and

$$E_c = \frac{E\ b\ a^3\ w}{8\ A\ L_T{}^3} \qquad \text{(A11.3)}$$

Where:-

V_o = Circuit output signal (V)
V_I = Circuit input signal (V)
K = Gauge factor for strain gauges
w = Separation of clip legs (mm)
a = Thickness of steel clip (mm)
L_s = Distance of strain gauges from clip points (mm)
$\mathcal{E}_R$ = Strain occurring in rubber surface
L_T = Effective length of clip legs (mm)
E_c = Effective modulus of clip (N/mm^2)
E = Modulus of spring steel (N/mm^2)
b = Width of steel clip (mm)

From these formulae it can be seen that, for a given strain in the rubber surface, the strain occurring in a gauge (and hence the output signal) is proportional to w, a, L_s and $1/L_T^3$. However, the effective modulus of the gauge is also dependent on most of these parameters. As L_s, obtained from the strain gauge position, is fairly difficult to control and b, the clip steel width is constrained by other considerations such as gauge width, the most easily varied parameters are w, a and L_T. Therefore, an investigation was carried out to find the optimum values of these parameters which would give a high output signal but would not cause a stiffening effect on the rubber surface. To do this some "mock" clip gauges having no strain gauges attached were made up with the dimensions shown in Table A11.1 and tested for stiffening effect as described previously. It was decided that this technique was not sensitive enough and a new method [137] was devised involving the comparison of the stress/strain behaviour of belt samples with and without clips attached. This showed that those clips made of thicker steel had a very slight stiffening effect, as could be expected from theory because, from equation A11.3, it can be seen that doubling the steel thickness increases the clip modulus by a factor of 2^3. It was therefore decided to make up a clip gauge having the same dimensions as "mock clip 1". A calibration curve was obtained from this and reproducibility experiments were carried out to see if the gauge could be moved from one sample to another without any significant change in calibration.

Experiments with "Optimum" Clip

A calibration curve was obtained from the "optimum" clip gauges using the method described previously. The legs of the clip were carefully

	Approximate steel thickness a (mm)	Gap between clip legs W (mm)	Effective clip length L_T (mm)
Original clip gauge	0.2	3	30
Mock gauge 1	0.2	8	28
Mock gauge 2	0.4	7	20
Mock gauge 3	0.4	5	30

TABLE A11.1

Details of "mock" clips used in determination of optimum clip dimensions positioned and then pushed into the belt surface. A small amount of polyurethane based adhesive was used to ensure that the clip could not come out of the belt accidentally. This adhesive had extremely low modulus and very high elongation capability. The strain occurring on the belt surface was measured over a grid length of 50 mm and the circuit output was initially set to zero using a balancing potentiometer.

A plot of the clip gauge output versus strain is shown in Figure A11.19. The gauge output appeared to be fairly linear, showed no noticeable hysteresis and allow measurement of changes of $\pm$ 0.2% strain. When positioning the transducer it was difficult to ensure that the clip was pushed into the belt at "zero strain" (that is, that the clip legs remained at the original separation). It was therefore very important that the output should be linear as, if this was the case, the initial separation of the legs would have no effect on the subsequent calibration curve.

The reproducibility of the clip gauge output was tested by removing it from the first sample and positioning it carefully on another similar sample. The calibration curve thus obtained is also shown in Figure A11.19. The results obtained did not demonstrate the degree of accuracy required and so improvements were made to the design of the clip gauge. These were:-

(a) The clip legs were made of two separate pieces of spring steel. Once strain gauges were bonded to these they were joined at one end by a small steel block using an epoxy adhesive. This meant that heat treatment of the steel clip was no longer necessary and the positioning and bonding of strain gauges became much easier.

(b) As gauge bonding was easier, a gauge adhesive more suitable for long term work could be used; that chosen was "Micro Measurements M-Bond 600" which required an elevated temperature cure.

(c) Two strain gauges were attached to each clip so that four gauges were present on the final clip. This meant that a "four active arm" Wheatstone Bridge circuit as shown in Figure A11.20, could be used which made the wiring of the clip more complicated, but meant that a higher output signal would be obtained for a given separation of the clip legs.

(d) The strain gauges used were smaller than those used previously so that they could be positioned further away from the clip ends and so would be more likely to be within an area of even strain distribution.

Photographs of the clip gauges produced are given in Figures A11.21 and A11.22.

An improved method of calibration was used: for this, a pair of vernier calipers having thin probes was positioned so that its probes touched the clip legs just below the shouldered area. (That is, at the position where the rubber belt surface would be.) The distance between the caliper probes was then varied and a calibration curve was

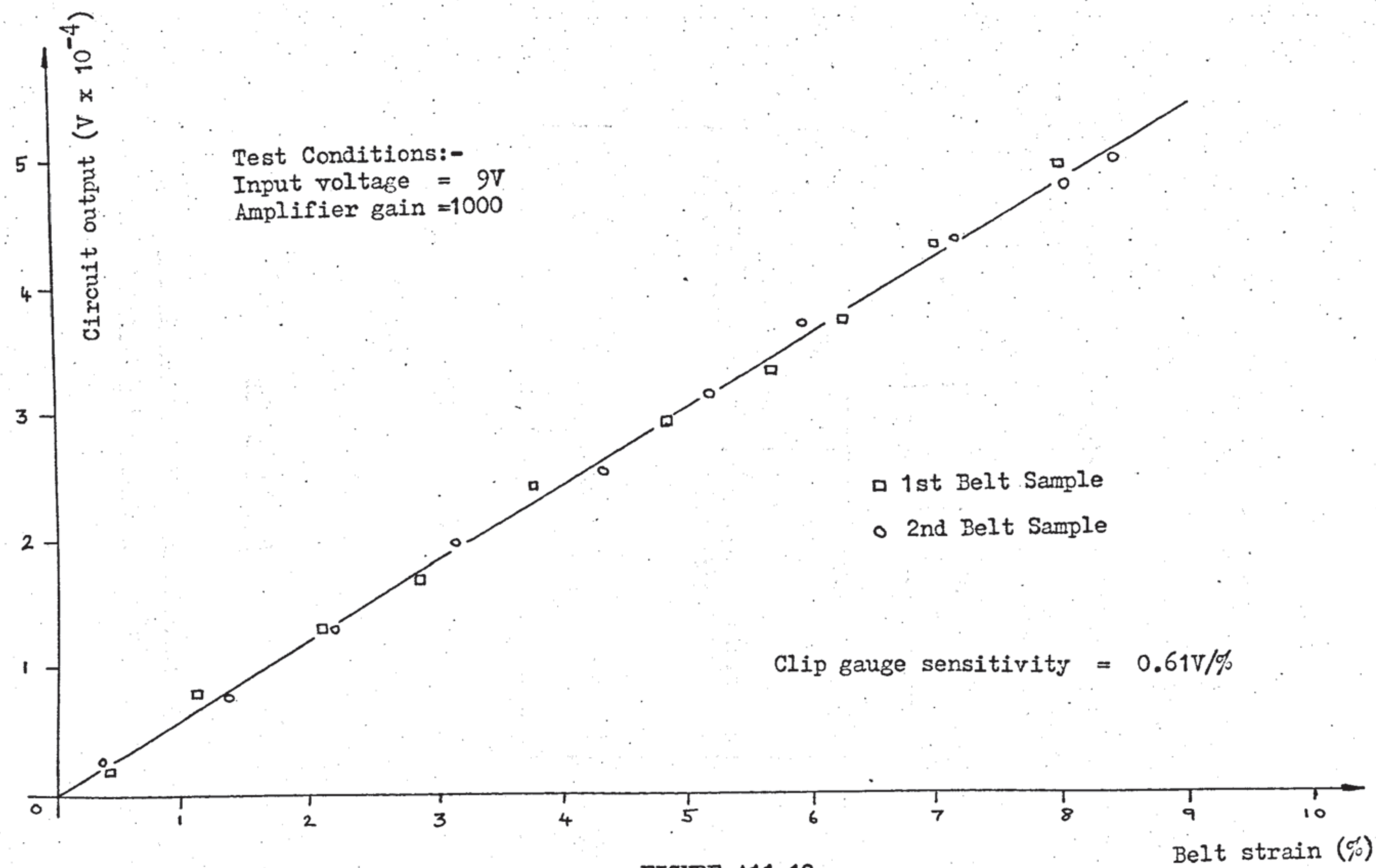

FIGURE A11.19

Calibration curve for "optimum" clip gauge produced

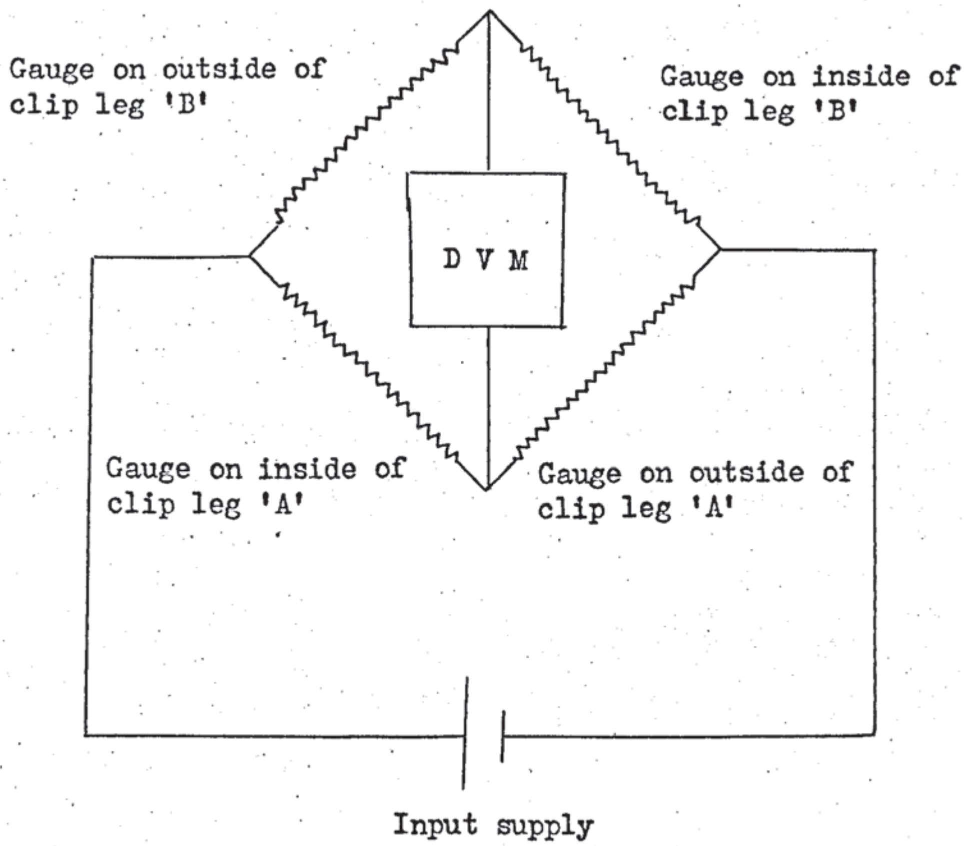

FIGURE A11.20

Wheatstone bridge circuit for clip gauge with four strain gauges

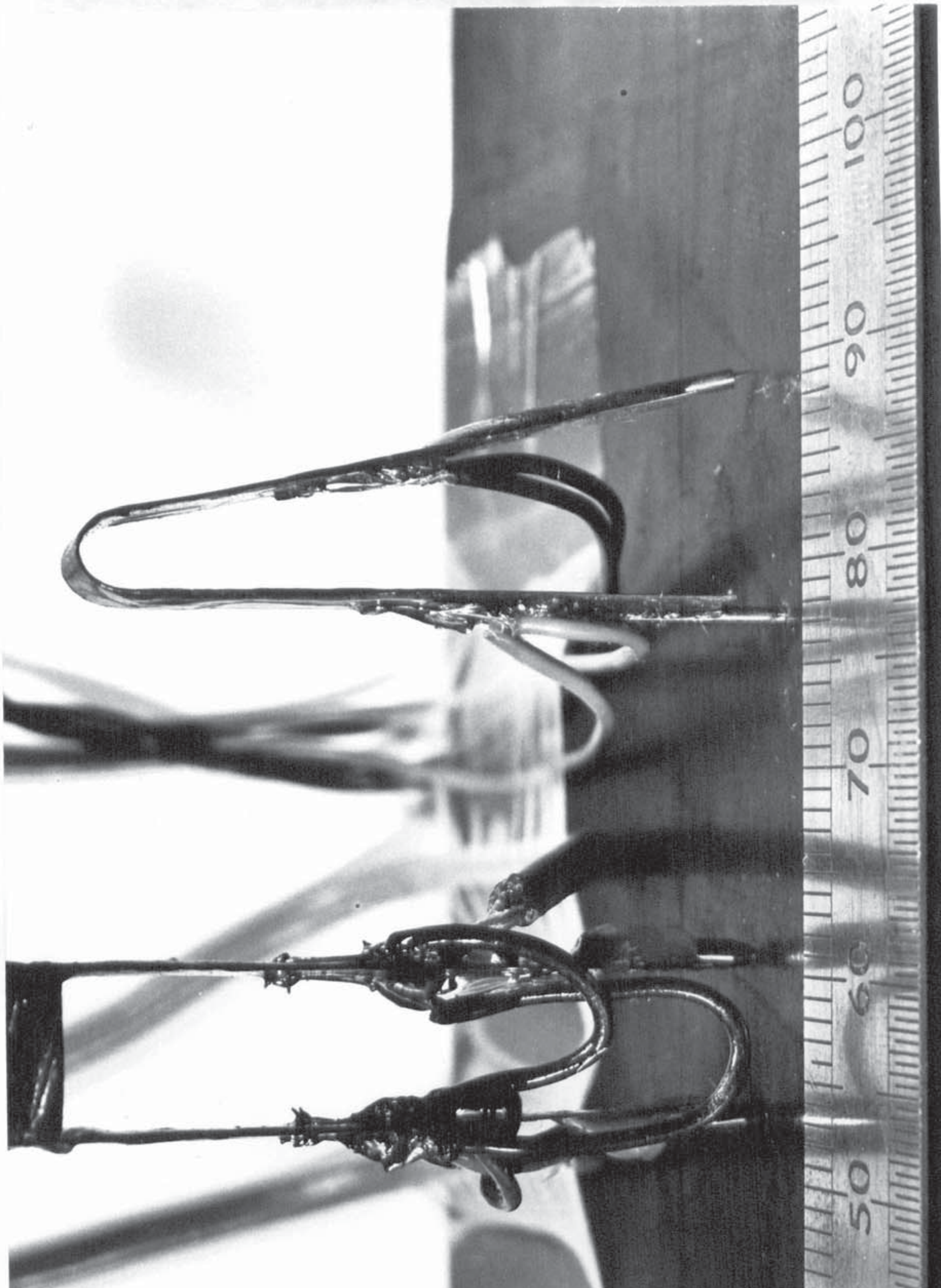

FIGURE A11.21 Two types of clip gauge produced (final design on left)

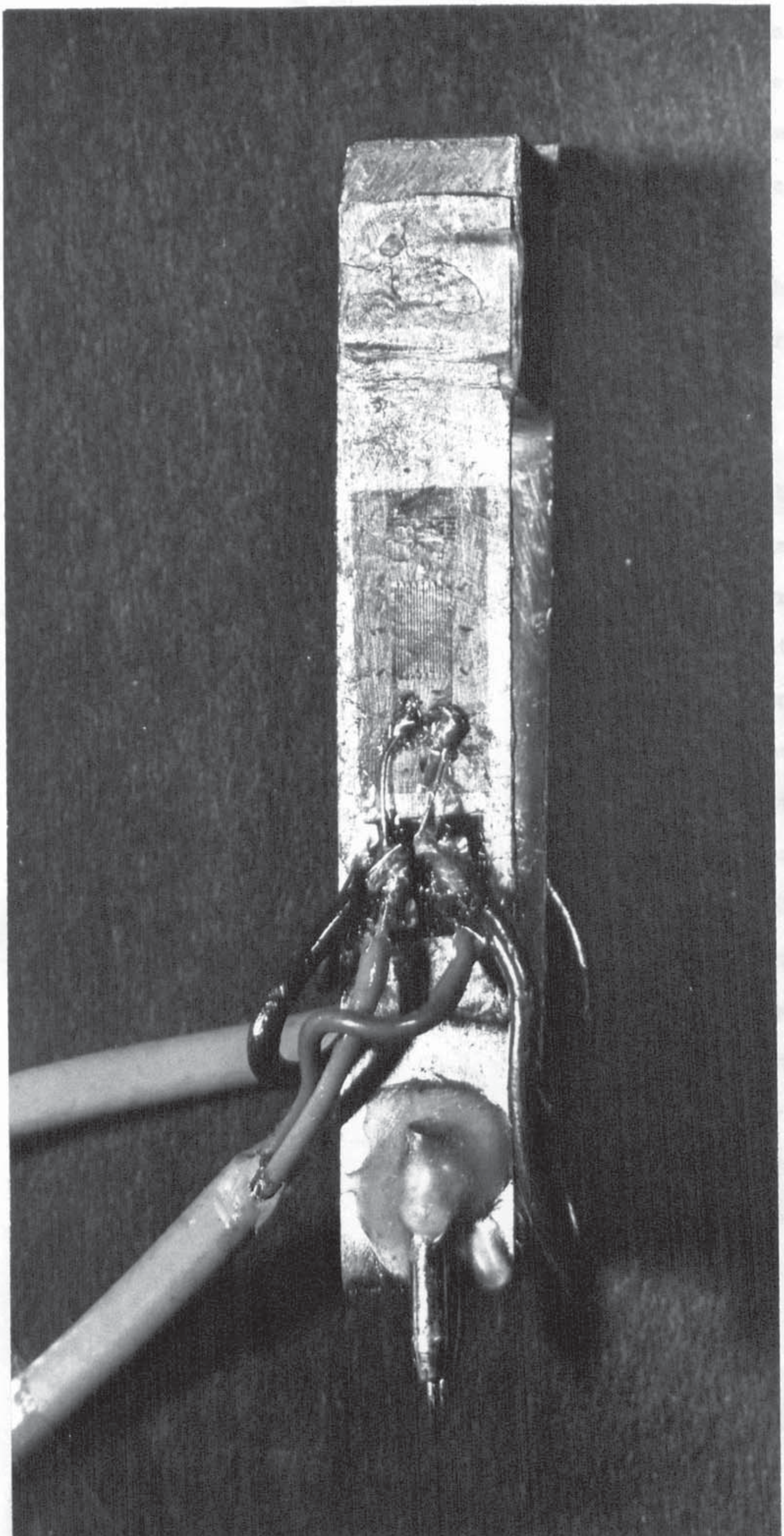

FIGURE A11.22

Close-up of final clip gauge developed

obtained from the clip gauge in both extension and compression. The input voltage used was seven volts as this was found to be the optimum value giving a high output signal without any instability. The results obtained are shown in Figure A11.23. It can be seen that the clip output was linear and represented a great improvement on that of the previous type. Strain variations of 0.1% could be measured without the use of a signal amplifier and the output was reproducible over different strain cycles.

Thus, a strain measuring device suitable for use in conveyor belt surface strain studies was developed. The experimental investigation described in Chapter 8 suggested that the strain levels occurring near the idler gap are not critical for modern plied belt constructions. Nevertheless, a follow-up to the load support programme was envisaged (see Appendix 12) which would include surface strain measurements as such strains, although not necessarily detrimental to the belt performance, could still influence load support. A greater understanding of the strain levels occurring would be useful in the development of (a) plied belt constructions with "optimum" inherent load support and (b) completely new belt constructions such as those using "sheets" of synthetic fibre for which strain levels might, in fact, be critical. In addition, the devices developed could be used in any other experimental study of conveyor belt behaviour and so provide an increased knowledge of the overall mechanism of belt conveying. The devices do still have some disadvantages; no load can be placed directly on top of the clip and each clip has to be individually calibrated. This last point means that if, say, twenty clip gauges were to be produced, twenty calibration "gauge factors" would be required, thus making analysis of results a lengthy process. If a very large number of clip gauges were to be produced, it would be worthwhile to develop a controlled manufacturing technique that would ensure that the gauge factor was, within acceptable limits, the same in every case. Other improvements could also be made to the clips; for example, a more elaborate method of joining the clip legs could be employed to ensure that no slippage could occur between them because of breakdown of the bonding agent. However, it is unlikely that such a degree of sophistication is required, at present, for belt studies and the clips described above should prove adequate for the present state of the technology. It is also likely that such clip gauges could be of use in other relatively low technology industries dealing with textile reinforced rubber products. For example, strain measurements in hose could be one other application for the devices developed.

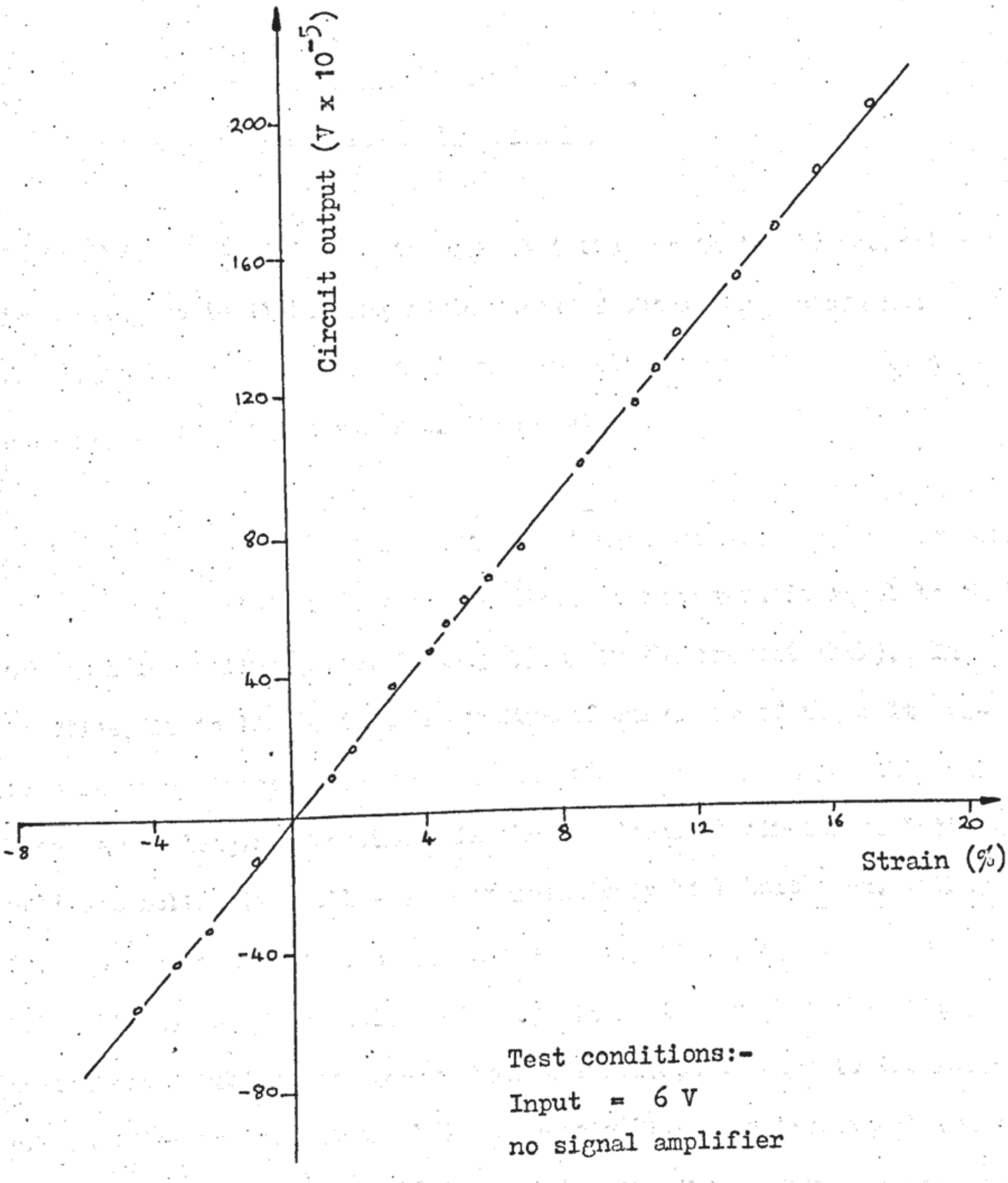

FIGURE A11.23

Calibration curve of final design of clip gauge

NOTE:- The circuit output values were adjusted so that zero output corresponded to zero strain.

APPENDIX 12

Strain levels occurring in a belt at the idler gap

It has been claimed in Section 8.2 that the strain levels occurring in a belt due to it bending in the area of the idler gap are not likely to bring about failure in modern belt constructions. This appendix describes the basis of this claim.

At the idler gap the belt is bent both longitudinally and transversely. The minimum possible radius of longitudinal curvature is equal to that of the idler roller (approximately 75 mm in the present work). In practice, it is likely that the radius of curvature of the belt carcase in this direction is much greater than the minimum possible value because the length of contact, in the longitudinal direction, between belt and roller is small - even at relatively high belt sags. (That is, the belt does not wrap around the pulley to any great extent - but enough to cause anticlastic bending.) It is likely that the belt bottom cover will deform over the small contact length to the radius of the idler roller but the belt carcase will have a "smoother" transition over the idler and will be bent to a radius of curvature greater than the minimum possible. It is therefore likely that longitudinal strains of approximately three to four percent will occur in the outer warp fibres of the belt carcase. Such strain levels will not cause failure.

The strains occurring due to transverse bending are likely to be much more severe, as can be shown from some simple theory:-

If the belt curvature in the area of the idler gap is assumed to be circular - and the experimental results, as demonstrated in Figure 8.2.16, have suggested that this is not an unreasonable assumption - the belt deformation when the bottom belt cover just touches the end of the idler centre roller is as shown in Figure A12.1

In Figure A12.1, A B = A C because tangents to a circle from a point are equal in length. Therefore, in triangles A C O and A B O,

O A is common

O B = O C because they are equal radii

and A B = A C because they are equal tangents

Thus, the triangles are conguvent and so:-

Angle C O A = Angle B O A

Angle C O B is equal to β because it is equal to the external angle of the quadrilateral A B O C. Therefore Angle A O C is equal to $\beta/2$ and the radius of the circle is given by:-

$$R = \frac{A\,B}{\text{Tan } \beta/2} \qquad \text{(A12.1)}$$

A B represents the idler gap and the angle β is the troughing angle. Therefore, in practice, the minimum ("highest strain") value that R can have before the belt starts to press into the idler gap is given, for 45^{o} troughing idlers, by:-

$$R = \frac{10}{\tan 22\frac{1}{2}} \text{ mm}$$

$$\text{or } R \simeq 24 \text{ mm}$$

If the carcase thickness of the bent belt is x, the bottom cover thickness is t_b, and the neutral axis is assumed to be at the mid point of the carcase, the strain in the outer carcase fibres is given by:-

$$S = \frac{x/2}{R - t_b - x/2} \times 100\% \qquad (A12.2)$$

If "worst case" values, which for the present work correspond to the 630/4HD belt construction tested, are put into equation A12.1, the following results:-

$$S = \frac{5.5/2}{24 - 1.5 - 5.5/2} \times 100\% \simeq 14\%$$

Weft samples of the belt constructions tested were placed in a tensometer and their elongation at break was recorded. It was found that, in each case, the strain at break was approximately 30% and the force required to extend the sample to approximately 25% was much lower than that required to increase the length from this value to break. This figure corresponded to the high levels of crimp present in the weft samples (see Figure 7.7.6) and so up to 25% belt extension the carcase cords were not being stretched but, instead, crimp was being removed from them. This would not bring about their failure. Thus, the maximum transverse strain levels occurring during service were unlikely

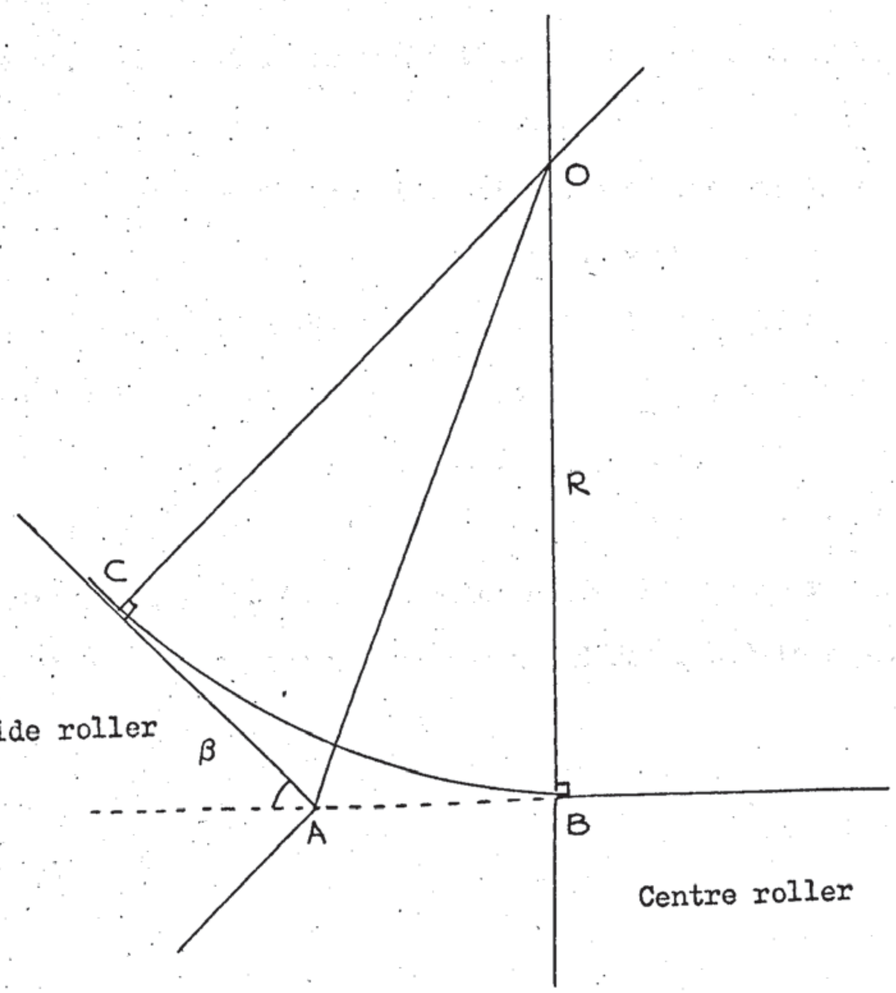

FIGURE A12.1

Transverse belt deformation in area of idler gap to damage the belt. (It should be noted that in practice the belt should not actually touch the end of the centre roller, and if the acceptable minimum non-contact length of 10 mm suggested in Section 8.3 is used, the maximum strains occurring reduce to only 8%. The maximum possible longitudinal strains were much less than the transverse, and so it is probable that the combination of bending in both directions is not detrimental to modern plied belt constructions. However, an experimental investigation of the strain distribution in the idler gap area would show definitely whether or not this is the case. The results of such an investigation would also be useful in the development of new carcase constructions, particularly if they had considerable transverse stiffness and were liable to fracture before they reached the "failure point" for present constructions of starting to press into the idler gap. The strain measuring devices developed during the present work and described in Appendix 11 would be suitable for this type of investigation.

APPENDIX 13

Review of methods of testing the Stiffness of Conveyor Belting

The experimental results described in Section 8.2 demonstrated the strong dependence of belt transverse load support on belt stiffness. It was therefore realised that a small-scale stiffness test that reproduced the degree of deformation encountered by the belt in service should be adopted as a standard technique of testing for belt inherent transverse load support capability. This review of methods of testing the stiffness of conveyor belting was carried out in order to help limit the choice for such a test.

It has been stated [138] that "there are three distinct uses for a mechanical test. It may be used (a) to obtain design data, (b) for quality control and (c) for specification purposes". Ideally, a belt flexibility test to assess load support would cover all three categories as, at present, there is no accepted test for any of them. The test chosen should therefore be quick, simple, service-related and reproducible regardless of operator or laboratory. For design data assessment it would also be useful to have a theory relating the results of the test to the physical properties of the sample. Simple elastic theory is usually used to derive these properties from present common flexural tests, but this theory is only valid for light, thin, narrow, linearly elastic (Hookean), homogeneous, isotropic beams bent only a small amount. These conditions are not all satisfied by conveyor belting, but it should still be possible to define terms for design data such as flexural rigidity and flexural strength analogous to those used in the simple theory.

In the sections below the flexibility test methods used previously for conveyor belting are described and these are followed by some techniques which have been used for other highly deformable materials. The methods described are not discussed in any great detail but some of the advantages and disadvantages of each from the point of view of load support assessment are mentioned. (References are quoted from which further information can be obtained if required.) There are some problems common to all of the techniques and these are mentioned in the conclusions.

Testing techniques used previously
for conveyor belting

The test techniques described here have all been used by various organisations for the assessment of belt stiffness or flexibility, but only two of them have been accepted as standard methods and several have been rejected by the organisations that devised them. Some of the tests mentioned were designed for purposes other than that of assessing transverse load support, but this does not necessarily mean that they are unsuitable for this use.

It was found by the National Coal Board [54] that the warpwise flexural rigidity could affect power transmission efficiency and that in the weft direction flexural rigidity must be such as to enable the belt to take up a troughed form readily without being too flexible and giving rise to spillage. The National Coal Board developed three tests because of these factors: one measured hysteresis power losses during cyclic deformation of a belt and so is not relevant to this review; the other two measured flexural rigidity and these are described in the following passages:-

(a) Pantograph or Stechert Test

In this test a belt sample is bent into a 'U' shape in an apparatus consisting of two surfaces maintained parallel to each other through a pantograph linkage (see Figure A13.1). The force, F, required to deform the belt so that the pantograph plates are distance 'd' apart is recorded. The flexural rigidity, EJ, is then given by [109] :-

$$EJ = 0.34830 \quad F \quad (d - t)^2 \qquad (A13.1)$$

where t is the belt thickness.

The National Coal Board used a weight of eight pounds to provide the force and measured the distance 'd' fifteen seconds after the load was first applied. This test method was used during the experimental investigation described in Chapters 7 and 8 and has been used to assess the change in belt flexibility during service [87].

Advantages:

(i) The method is quick and the apparatus simple.

(ii) The test could probably be applied to a part of a continuous length of product without having to cut the length. This is particularly true if measuring longitudinal stiffness, but might be impracticable for transverse measurements.

Disadvantages:

(i) The belt deformation caused during the test is much more severe than that encountered during service (see Figure 8.2.16). If the dimensions of the apparatus were arranged so that this was not the case, the equipment would probably be too cumbersome.

(ii) There is a maximum measurable value for EJ dependent on the size of the pantograph, the load and the thickness of the belt.

(iii) Belt samples must be deformed to a fairly high degree before being put into this apparatus. This can lead to errors occurring due to differing previous loading history of the samples.

(b) Cantilever or Beam Deflection Test

In this test a belt sample is clamped to a flat surface at one end, loaded at the other and the resulting deflection, d, is measured (see Figure A13.2). The test used by Bridgestone Limited for measurement of troughability is basically the Cantilever test. (See Section 2.2.) The National Coal Board used a belt sample of length, X, equal to ten inches and a weight, F, of 0.33 pounds. The measurements were taken ten seconds after releasing the load. If an analogy is made with simple elastic theory the flexural rigidity of the sample is given by:-

$$EJ = \frac{F\,X^3}{3\,d} \qquad \text{(A13.2)}$$

Advantages:

(i) The apparatus is simple.

(ii) The "operator element" in sample loading can be easily eliminated by a trap-door arrangement [54].

Disadvantages:

(i) The result reading with this test takes much longer than with the pantograph.

(ii) There is a lower limit to measurable flexural rigidity owing to the limit on the size of the deflection imposed by the physical dimensions of the apparatus.

(iii) The type of loading and deformation occurring in service is not reproduced.

The National Coal Board concluded that the two methods of testing described above were not satisfactory. Therefore, another test method was devised; this was the "Four-point bend test" and, although it has not yet been included as part of the National Coal Board published Standards, it is still carried out on all conveyor belting submitted for approval to the board's Materials Testing Laboratories:-

(c) Four-point Bend Test

This test consists of bending a sample by applying forces at four points as shown in Figure A13.3. If an analogy is made with simple elastic bending theory, the flexural rigidity of the sample is given by:-

$$EJ = \frac{F\,A\,X^2}{16\,d} \qquad \text{(A13.3)}$$

where F, A, X and d are as shown in Figure A13.3

The sample dimensions used by the National Coal Board are approximately 140 mm by 100 mm and a load is applied so that the inner bars of the apparatus move 6 mm. Five load cycles are carried out and it is found that a stable situation is reached after the first. The

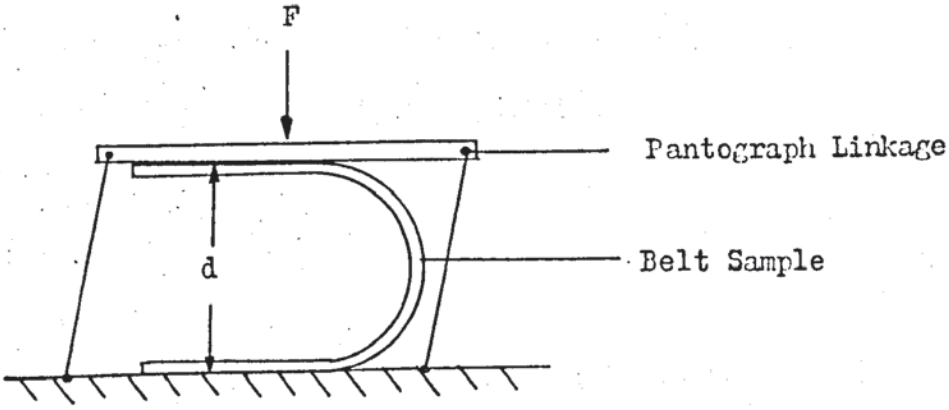

FIGURE A13.1

Pantograph Test

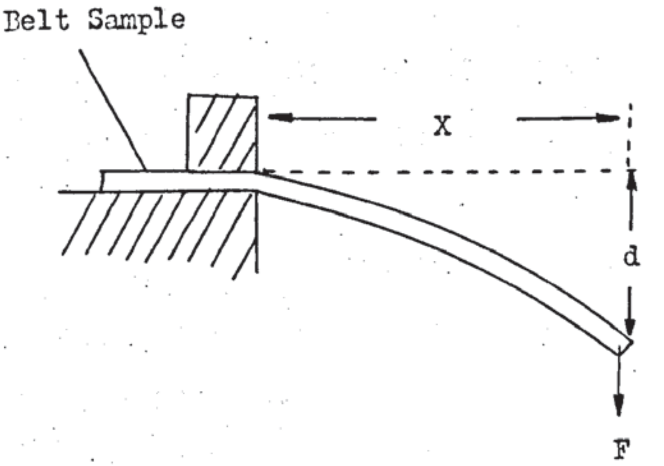

FIGURE A13.2

Cantilever Test

test is repeated with the sample facing in the opposite direction to the original position.

Advantages:

(i) A considerable length of the sample (the central zone) is subjected to a uniform stress.

(ii) Radii of curvature comparable to those occurring in service can easily be obtained with suitable design of apparatus.

(iii) As the sample is placed in the apparatus when flat, the "operator element" is reduced to a minimum and a high degree of reproducibility is obtained.

Disadvantages:

(i) The test is slightly more difficult to devise and carry out than some other bending tests.

It has already been mentioned that two tests that provide a measure of belt flexibility have been included in published specifications. These are discussed below:-

(d) The French Pendulum Test

This test forms part of the French Standard M-81-651 and is used to find the longitudinal flexibility of belting. The apparatus required consists of a pendulum as shown in Figure A13.4. In the specification the pendulum rod length is one metre and the load weight is two kilogrammes.

The belt specimen is clamped at one end and is attached to the pendulum rod at the other. The pendulum is displaced from the equilibrium position to an angle of 45° and then allowed to complete a number of oscillations. The flexibility of the belt is expressed by the angle of deflection, θ, at the tenth oscillation.

Then the radius of curvature, R, of the sample is found from:-

$$R = \frac{L}{\theta} \qquad (A13.4)$$

where L is the free length of the sample.

This method does not result in a simple expression for the flexural rigidity. The equation quoted can only give a comparative indication of flexibility.

<u>Advantages:</u>

(i) The test apparatus is simple.

<u>Disadvantages:</u>

(i) The test does not reproduce the transverse bending that occurs in service.

(ii) The measurement of flexibility from the angle of displacement at the tenth oscillation would appear to be arbitrary.

(iii) There is some difficulty in measuring the angle Θ accurately.

Some improvements to the specifications for the sample length and other parameters have been made [139] , but it would still appear that the basic test method is not suitable for load support capability assessment.

(e) The F/L Test

This test, again originally introduced in France, has been accepted by the International Standards Organisation and forms part of British Standard 490 for conveyor belting. The test method is described in Section 2.2 and a diagram of the apparatus used is given in Figure 2.2.6.

Advantages:

(i) The test is simple, and has been found to be very reproducible.

(ii) As the test sample is, in effect, a beam hanging under its own weight, a term analogous to flexural rigidity can easily be defined.

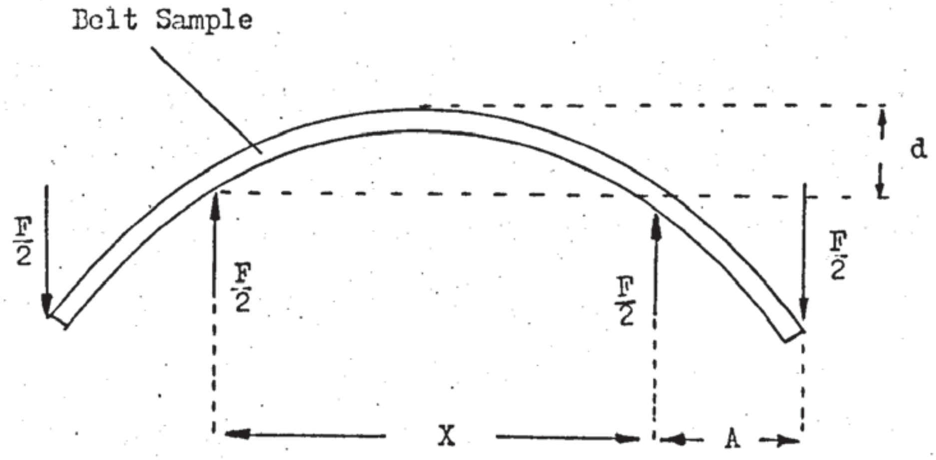

FIGURE A13.3

Four-point Bending Test

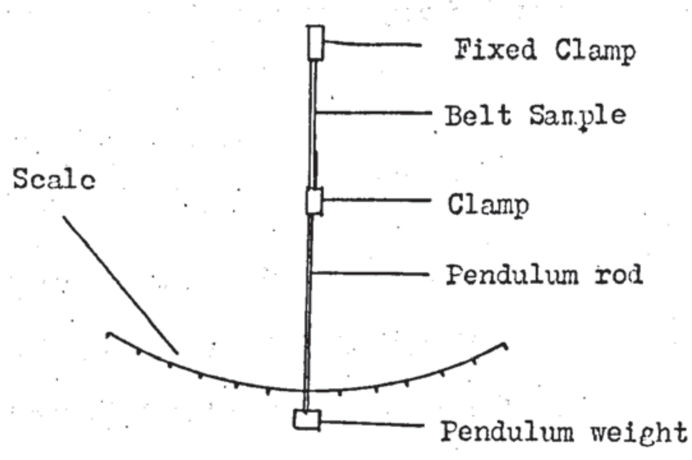

FIGURE A13.4

Pendulum Test

Disadvantages:

(i) The test is only intended for empty belt troughability data and does not give load support information. (Attempts to specify maximum permissible F/L values have not been successful.)

(ii) The test result expressed does not vary much with changes in belt construction (for modern plied belting). This is particularly true for wide belt samples.

Apart from the National Coal Board and the various Standards Institutions, the only organisations to have published details of any belt stiffness test work are belting manufacturers. For obvious reasons the information on any such work is fairly limited, and no manufacturer has felt sufficiently strongly about a test method to put it forward for consideration as a standard. Details of two tests devised by manufacturers concerned with conveyor belting are given below.

(f) Enka Glanzstoff method

The following test method was devised by the yarn manufacturing company Enka Glanzstoff to test the buckling behaviour of fabrics in conveyor belts.

For this, a belt sample is made up with a brass plate of 0.25 mm thickness included on the surface (see Figure A13.5). The test specimen is then bent on a four point bending apparatus so that compression appears in the fabric and rubber. The brass plate acts as a neutral fibre and the bending force required to compress the sample can be found by subtracting the results obtained when using only the brass plate from those which include the effect of the belt sample.

This test would obviously not be suitable for a standard test to assess the load support capability of production belts as it requires a special belt sample with an attached brass plate. It is, however, useful to note the method used to ensure that only compression takes place in the belt sample, and this could prove to be useful in any studies of the behaviour of belting under compression.

(g) Goodyear Test

The test described here cannot be regarded strictly as a bending test, but it was devised specifically to compare the load support properties of different belt constructions [23]. The test is carried out under dynamic conditions by running an endless belt sample on a special two end-pulley machine at various speeds and tensions. Some three-roll idler sets are mounted between the end-pulleys. (See Figure A13.6.) As a bulk material load cannot be put on such a machine, the idler sets are arranged so that a vertical component of belt tension tends to force the belt into the idler gap, thus causing the type of failure that can result from inadequate transverse load support. Initially, tests on belts known to be successful with respect to load support in the field were carried out to provide control data.

Advantages:

(i) This is a dynamic test attempting to reproduce the conditions encountered in service.

(ii) The test reproduces the type of damage that occurs when load support is inadequate.

Disadvantages:

(i) The equipment required to carry out this test is relatively large scale.

(ii) Inherent properties of the belt construction (such as belt stiffness) are not compared. Instead, only the times to failure can be compared, thus making it difficult to relate the test to any elasticity theory.

(iii) The time to obtain results is long compared to the other tests described.

It is thought that this test makes a serious attempt to reproduce service failure and a rig of this type could be used to correlate inherent belt properties with load

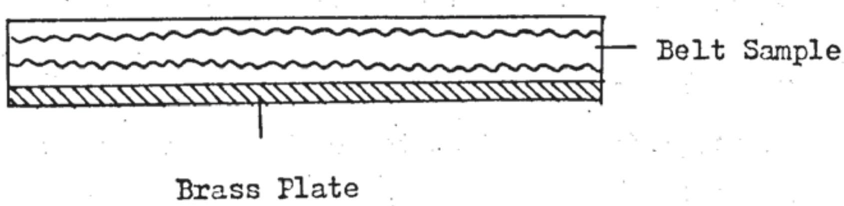

FIGURE A 13.5

Test Sample used in Enka Glanzstoff Test

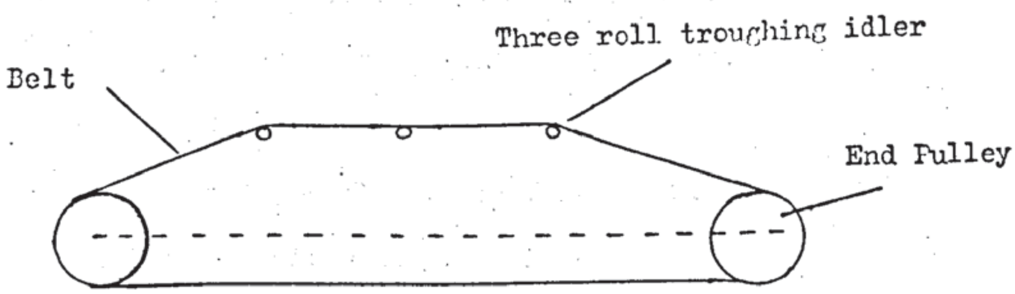

FIGURE A13.6

Layout of equipment used in Goodyear Testing Rig support failure. That is, it could be used to carry out work along the lines of that described in Chapters 7 and 8 but under dynamic conditions. Unfortunately no further details of the work by Goodyear have been published and no standard method of describing transverse load support characteristic has resulted.

Other Test Techniques that could be used for Conveyor Belting

The following two test techniques have not been used for any published work with conveyor belting, but they are both suitable for highly deformable materials.

(h) Three-point bend test

The three point loading system (see Figure A13.7) is similar to the four point system. It is used extensively for standard testing of plastics.

The parameter analogous to flexural rigidity can be easily found from the formula:-

$$EJ = \frac{3 F X^3}{D} \qquad (A13.5)$$

where F, X, Y and D are as shown in Figure A13.7

Advantages:

(i) The automatic recording of the central deflection is much easier than with four-point bending.

(ii) There is a great deal of published work on large deformation in three point bending [141] and limitations of simple elastic theory have been discussed [142].

Disadvantages:

(i) Unlike the four-point bend test, there is not an area of uniform stress along the sample.

(i) The Loop Test

This is an extremely simple test which involves finding the "bending length", C, of a flexible strip of material. The test is carried out by laying the strip of material on a horizontal surface in the form of a loop (see Figure A13.8). The bending length is then given simply by 1.103 times the loop height [143]. The full theory of the test is given in the reference quoted. The flexural rigidity, EJ, can easily be found from the relation:-

$$EJ = Z\,C^3 \qquad (A13.6)$$

where 'Z' is the sample weight per unit length.

Advantages:

(i) This test is obviously very quick, inexpensive and simple.

(ii) The quantity "flexural rigidity" can be easily calculated.

Disadvantages:

(i) No loading whatsoever is applied to the sample, and so the test does not represent service conditions.

(ii) There are differing views on the validity of the theory of the test. [138, 144].

Conclusions from the review of bending test procedures

None of the tests described above has a proven correlation with service conditions, but now that the importance of inherent belt transverse stiffness has been quantified and the belt shape near the idler gap has been studied in the present work, it should be possible to select a small scale test which reproduces, as nearly as possible, the stresses and deformation occurring in a working belt. In practice, both longitudinal and transverse curvatures occur simultaneously, but the small-scale tests described do not cause this type of bending. However, as the strain levels occurring are unlikely to cause fracture in modern belt constructions (see Appendix 12) and load support capability therefore is dependent more on stiffness than on strength, it is probable that a bending test causing only transverse curvature would serve the purpose. If this is the case, it would appear from this review that the Four-point Bend Test would be the test method most

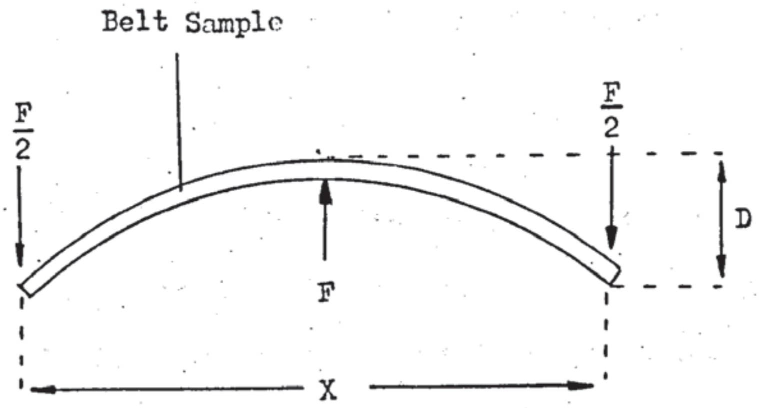

FIGURE A13.7

Three-point Bend Test

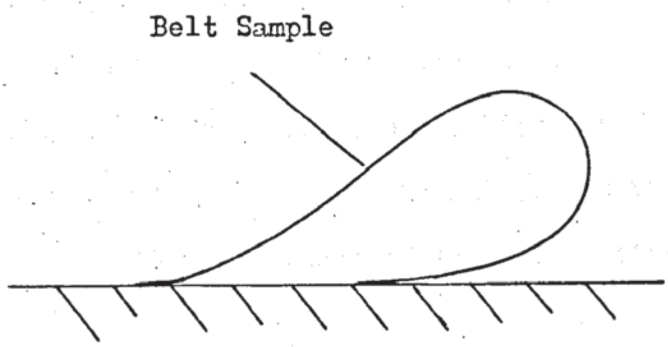

FIGURE A13.8

Loop Test

likely to fulfil all the requirements of a standard test method for transverse load support capability.

The major advantage it has is that, theoretically, it produces a region of constant stress in the belt sample, and it is likely that the belt area near the idler gap approaches this in practice. Another advantage, not mentioned previously, is that tensometers already required by any belting laboratory to carry out certain standard tests can be adapted to be used for the Four-point Bend Test. It is therefore considered that work should now be carried out to design a Four-point Bend Test suitable for assessment of inherent belt load support capability of modern plied carcase constructions.

However, a more useful objective would be to devise a completely new test causing curvature in two directions and so representing service conditions more closely. This could then be used to assess load support according to two criteria for <u>any</u> type of belt constructions:-

(a) Strength - The effective strain caused by bending in two directions should not be sufficient to cause the possibility of the belt carcase breaking. (That is, the maximum strains would only be allowed to reach a certain acceptable percentage of those occurring at break.)

(b) Stiffness - Sufficient stiffness must be present to prevent the belt pressing into the idler gap during service.

Before such a service-related test can be devised, it is necessary to gain a better knowledge of the strain distribution occurring near the idler gap, and so an investigation using the strain measuring devices described in Appendix 11 would be useful. Two other features of belt behaviour should be taken into account when selecting the final test method:-

(i) Time dependent deformation - an attempt should be made to choose a test method which results in a stable situation within an acceptable time rather than involve arbitrary time limits.

(ii) Deformation dependent on previous loading history - as the degree of deformation of a belt sample under a certain load can be affected by the previous stresses imposed on it, it is advisable to select a test method in which the belt is under no initial strain. That is, the belt sample should be flat and unloaded when the test is started.

It was thus concluded from the review of bending test procedures that a new test method should be devised to cause curvature in two directions to the degree encountered in service. Meanwhile, the four-point bend test would appear to be the most suitable test for the assessment of load support capability of the plied belting at present in common use.

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