

## SYNOPSIS

### TITLE OF PAPER:

## **Conveyor Belt Selection Design of High Speed Conveyors**

The demand for raw materials is growing and so mine operators are calling for ever more cost-effective conveyor systems that can run at high speeds. Designing units of this kind requires precise calculation of all belt tensions during steady and non-steady conditions and of the additional stresses that occur, for instance, in the trough transition zones and on curves.

The belts have to be designed and manufactured to the highest standards so as to ensure trouble-free running.

Detailed calculation work on belt splices followed by their testing on test benches gives an accurate indication of their strength.

References will be made to mining projects that use high-speed belts.

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## 1. Summary

To ensure that a high speed conveyor remains cost-effective and reliable, careful and detailed work is necessary from the planning stage right through to the commissioning phase.

Calculation of the belt tensions and other stresses must keep pace with the changes in conveyor belt technology. Particular attention must be paid to the dynamic processes involved in conveyor operations.

At the manufacturing stage, raw materials have to be used that will withstand the high dynamic stresses to which the belt will be subjected. The belt itself must be manufactured absolutely straight in order to be perfectly straight-running when in operation.

The splices have to be suitable for the particular belt system and should be calculated in detail and be tested exhaustively.

The maximum belt speeds have certainly not yet been reached in the case of many conveyors. However, belt speeds considerably higher than 7.5 m/s will remain an exception.

## 2. Introduction

Mine operators are facing a range of interrelated problems: the demand for raw materials is growing constantly; those materials are available either in low-yielding or geologically difficult deposits. Conveyor systems have to be ever more cost-efficient, and so belts must have ever greater conveying capacities to move both - overburden and materials.

In the lignite mines of W. Germany, conveyor units with capacities of 37,500 tonnes per hour have been in action since the mid-70s.

Conveyors with capacities on this scale have to be equipped with belts that meet very high technological standards, for they run at very high speeds, and have a variety of other stress factors to cope with.

### 3. Classification of belt speeds

The belt speed is a key factor in the design of a conveyor unit. In general, belts run at the following speeds:

* Special operations (process related) e.g. cooling conveyors	< 0.5 m/s
* Sensitive-material conveying e.g. coke bench conveyors	0.5 - 1.5 m/s
* Standard applications e.g. gravel conveyors	1.5 - 3.5 m/s
* Large quantities of materials long conveying distances e.g. overburden conveyance	3.5 - 8.5 m/s
* special conveyors e.g. thrower belts	> 8.5 m/s

It can generally be said that the faster a belt runs, the more economic it is. Obviously, this is especially significant in the case of long-distance conveyors. Such units are always operated at the highest speeds possible.

Naturally, higher speeds make greater conveying capacities possible. This means that with a specific flow of materials, it may be possible to use a narrower belt or one with a simpler trough design. Moreover, lower working tensions may result and the whole unit can quite often be built on a smaller scale and thus overall costs can be reduced.

#### 4. Design factors relating to high-speed belts

##### 4.1 Elongation properties - modulus of elasticity (dynamic ply modulus)

In addition to conveyor unit parameters, such as loading conditions, a number of other important factors have to be considered. Fast belts with long centre distances should have a high modulus of elasticity, i.e. low elongation values - permanent and elastic. (Fig. 1) When belts are started up or braked, additional stresses occur that are difficult to calculate: knowledge of the conveyor parameters and dynamic ply modulus is essential.

Even with continuous conveying, a belt unit can not be said to be in a quasi-stationary condition, as belts are stressed dynamically at the drive pulleys. Moreover, dynamic processes occur when a belt is started up or braked. The lower the modulus of elasticity is, the more difficult it is to prevent uncontrolled vibration.

Only steel cables have this high dynamic ply modulus and are therefore ideal as tension members for long-distance belts. Naturally, the design of the cable tension members is also very important if the belt is to perform efficiently.

##### 4.2 Rubber covers

Higher speeds mean greater wear. If no material acceleration equipment such as acceleration belts or specially designed chutes are available, the material has to be accelerated on the belt itself to the belt speed. Depending on the type of material conveyed, the covers have to be wear, cut and age-resistant in order to prevent damage occurring.

Furthermore, high adhesive values and shear strengths down to the core rubber are essential for transmitting the high drive forces from the pulley to the belt.

Ideally, covers for belts carrying normal bulk materials should have the following characteristics - the minimum values according to the German standards DIN 22102 ( fabric belts ) and DIN 22131 ( steelcable belts - Nov. 88 )

Grade X

- |                            |                           |
|----------------------------|---------------------------|
| * low abrasion             | max. 120 mm <sup>3</sup>  |
| * high tensile strength    | min. 25 N/mm <sup>2</sup> |
| * high breaking elongation | min 450 %                 |

Not only does abrasion play a role in wear. An ideal combination of abrasion resistance, tensile strength, breaking elongation and hardness leads to optimum, wear resistant covers.

If resistance to hot materials, oils, greases or other substances is not required, covers made of a mixture of natural rubber (NR) and synthetic rubber (SBR) are usually the best.

#### 4.3 Belt strengths

Belts conveying large quantities of materials over long distances must have a high breaking strength. Belts of St 10000 i.e. a breaking strength of 10000 N/mm are theoretically possible, but the upper limit is around 7000 - 8000 N/mm, for a number of reasons:

- \* Belts over St 7500 require steel cable tension members at least 14 mm thick and thus a drive pulley diameter of over 2000 mm
- \* The belt thickness for cables of this diameter is at least 35 - 40 mm, and so the ratio of dead load/working load rises considerably
- \* The part-lengths can only be about 150 - 250 m long because of the belt thickness, weight and the resulting roll diameter

- \* Splice lengths have to be over 6 m long, and so are time-consuming and costly.
- \* More partlengths mean more splices
- \* The efficiency does not increase in direct relation to the belt strength
- \* The troughability will be reduced in the case of small belts

Even belts up to this strength are difficult to manufacture; they need highly accurate processing. To ensure the steel cables in the belt are parallel and to prevent them moving sideways during vulcanization, the cables have to be pretensioned. Production facilities must be designed to pre-tension cables of these strengths and diameters.

#### 4.4 Belt widths

The conveyors used in the open pit lignite mines in Germany with a conveying capacity of 37500 t/h incorporate steel cable belts of 2800 to 3200 mm in width. The widest belt ever installed - 6400 mm - operates in one of these lignite mines. It, too, has set new standards.

#### 4.5 Straightness of steel cable belts

Steel cable belts that are used in such high-performance conveyors have to be made in production facilities where continuous individual cable pre-tensioning is possible. During manufacture, in order to achieve uniform pretensioning, the cables must under no circumstances be relaxed. Any irregularities in the manufacturing process, such as unequal cable pre-tensioning, can lead, especially in high-strength belts, to mistracking, causing automatic switch offs and leading to faults affecting the whole system.

## 5. Designing conveyors

### 5.1 Designing conveyors with the lowest belt tensions possible

There are various ways of attaining large conveying capacities. The trend is towards belts units operating at high speeds. A higher speed means that while the same capacity is achieved the load per meter is reduced and so a narrower belt can be used.

So the design of a conveyor is crucial. Even if the belt tensions are reduced by only 10 %, great savings can be made in the case of a long-distance belt system.

Various methods can be used to determine the drive power and the belt tensions. German standard DIN 22101, drawn up back in 1942 and completely revised in 1982, has played a vital role here; it provided the basis for Continental's first computerised calculations (CONTI-COM) in this field. However, calculation of the dynamic processes is not covered by this particular standard and so it has not been possible to calculate accurately the belt tensions that occur during start-up and braking. This shortcoming has been compensated for by applying suitably high safety factors.

The safety factor is calculated from a series of reductions, one of which - the basic reduction  $r_b$  - takes into account the belt's dynamic fatigue strength for finite life, among other factors.

At the design stage these days, less emphasis is placed on the supposed highest belt tension  $T_1$  on the drive pulley when the belt is starting up or running. Instead, a comprehensive analysis of the belt tensions at all points along the conveyor is made.

Belts that operate on uneven terrain as well as go up and down gradients often have their greatest belt tensions at various points along the conveyor. (Fig. 2)



Part-length calculation is a must today when a conveyor is designed. This procedure determines both locally occurring maximum belt tensions and critical belt tensions, which, for instance, can occur when the belt is braked and may cause a complete tension relief.

#### 5.1.1 Calculating the required power

All part-lengths with their different lifts have to be recorded in detail. Beside the normal operating conditions, i.e. empty belt or fully loaded belt, all other loading conditions have to be considered, i.e. transport starting, transport ending, part-lengths loaded on uphill or downhill gradients which will lead to the smallest (negative) or largest motional resistances. The installed power must be sufficient to handle such conditions. (Fig.3)

#### 5.1.2 Determining the belt tensions

The belt tensions can be accurately calculated once the motional resistances affecting each part-length have been ascertained. Additionally, the mass moments of inertia of idlers, belt, pulleys and drive units as well as the conveyed material are taken into account.

The belt tensions can be reduced significantly by changing the drive configuration, varying the angle of wrap, modifying the start up characteristics or tensioning systems. An optimum design will thus be obtained.

Of course, doing design work of this sophistication and thus achieving the ideal solution is only possible with support of computer programs. In view of the amount of calculation work involved, only a computer will identify the unfavorable conditions and secure reliable conveyor operations.

Important parameters such as

- \* limitation of slip on driven or braked pulleys
- \* limitation of belt sag along the installation
- \* constant total stretch of belt with fixed take-up pulley
- \* limitation of the stopping time or of the stopping distances

are subject to constant checks by Conti's computer programs (CONTI-COM). (Fig. 4)

## 5.2 Additional belt stresses

When selecting a belt, calculation of the belt tension is only one important factor among several that have to be considered. Computer-aided calculations generally indicate only the mean belt tension; however, this is then regarded as being constant right across the width of the belt.

In the flat-to-trough or trough-to-flat zones, on vertical curves and belt turnovers the belt geometry has to be checked carefully to make sure that no impermissible tension peaks occur.

Detailed research is especially important in the case of fast-running conveyors because changes in belt tension take place very quickly. A good example of this phenomenon is the belt tension characteristic for a trough-to-flat zone which is followed by a drive pulley.

### 5.2.1 Trough-to-flat transition (Fig. 5)

The trough-to-flat transition is the length between the last fully troughed idler set and the pulley. Due to the deformation of the belt in this zone the belt edges will get more stretched than the belt middle.

If the conveyor runs horizontally or ascends, the belt tension at the discharge pulley is considerably higher than at the loading station. In such cases, extra stressing at the belt edges can be expected. But it will not be excessive if the transition zone has been well-designed.

#### 5.2.2 Flat-to-trough transition (Fig. 6)

The belt undergoes the same changes in its geometry at the loading station. Normally, the belt tensions in this area are rather low, and so a pulley elevation is often considered. However, if the transition length is too short, and depending on the degree of belt tension in this area, only the belt edges will be subjected to tension, whereas the belt middle will be compressed. In such cases, there is a great risk of troublesome "omega" folds occurring.

This will not happen if the correct pulley elevation is used. In any case, accurate calculation work is necessary.

#### 5.2.3 Vertical curves

Concave and convex curves must also be carefully analysed in order to minimize tension peaks. Vertical curves and transition zones often overlap causing an accumulation of stress and leading to excessive tension at the belt edges. Thorough calculation work is necessary here, too.

Continental provides computer calculations (CONTI-COM) for dealing with all such cases. The programs ensure that belts operating at low safety factors will be troublefree and exceed their life expectancy.

### 6. Splicing Systems

The drive power of the motors is transmitted to the belt via the drive pulleys using the well known Eytelwein formula. The high belt tensions resulting from this, however, have to be transmitted by both the belt and the splices.

On the assumption that a belt is manufactured without defects, its splices must then be regarded as the weakest links in transmitting the belt tensions. Fast conveyors with their high revolution values and high dynamic stresses require special splicing systems and these will vary depending on the belt in question.

#### 6.1 Static conditions

Even today, many belt units are still being designed without much attention being given to the splice configuration. Static splice strengths are frequently overemphasized. It's an approach that is totally inadequate, especially in the case of high-speed conveyors.

In the above-mentioned standard, DIN 22101, static breaking strength losses of 0 % are specified for one and two-step splices. Safety factors for stationary and non-stationary conditions are then related to the residual breaking strength of the splice. Safety factors between 6.7 and 8 are laid down for stationary conditions if the conveyor is a modern one.

#### 6.2 Dynamic conditions

In DIN 22101, the losses in breaking strength resulting from constant reduction are described as 'favourable', 'normal' or 'unfavourable'.

Key aspects such as

- \* the differences between the various tension members  
( fabric/steel cables )
- \* the differences between high-strength and low-strength belts with the same kind of tension member ( St 1000 - St 6600 )
- \* changes in the splice configuration
- \* the progress achieved and still to be achieved on the dynamic fatigue strength of splices

are not taken into consideration.

With conveyors that run at high speeds and which are subject to high belt tensions, the dynamic effects which occur in the splicing zone have to be taken into account. When the belt runs round the pulley, the belt tension goes from a high level to a low level very rapidly.

Dynamic stress becomes very great, especially in fast-running conveyors, and so most suitable rubber compounds need to be chosen.

The relatively high safety factors in a static design ( $S = 6.7 - 8$ ) can, under certain circumstances, be considerably reduced by taking account of the dynamic conditions and, if necessary, on the basis of static calculation, be reduced to a safety factor of about 5.

If measures of this kind are taken, lower-strength belts can be used or greater centre distances are possible or higher conveying capacities can be achieved - even though the belt tensions remain the same.

### 6.3 Splice configuration

Splices nowadays are made either as recommended by the belt manufacturer or else based on standards in which splice designs are specified, such as DIN 22131. However, research has shown that such splice designs, especially in the high-strength range, where three or four-step splices are necessary, do not always produce ideal results.

It is possible nowadays to design in advance an optimum splice by referring to the rubber characteristic values of the belt and the physical data of the tension member. To do this, it should be remembered that the tensile forces are only transmitted through the core rubber from one cable to the next and thus all in all from one part length into the next. (Fig. 7)

When designing a splice, two objectives are essential:

- \* the breaking strength of the steel cables must be used to the maximum
- \* the shear deformation of the intermediate rubber must be used to the maximum.

During a dynamic splice fatigue test, depending on a splice design, either the breaking strength of the steel cables will be exceeded or the shear stress in the intermediate rubber will lead to splice failure. In the former the splice tears; in the latter, the cables are pulled apart lengthwise.

#### 6.4 Testing procedures

Both the Institute for Conveying Technology of the University of Hannover (IFH) and Continental have conveyor belt test stands for checking the dynamic fatigue strength of conveyor belts. (fig. 8)

The revolution test stands are comparable to the conditions that the belts operate in. With a standard conveyor belt, the belt tension increases steadily from the tail up to the head station and then decreases very quickly during the run over the drive pulley.

On the test stands, the increasing belt tension is achieved by slowly moving the pulleys apart. After reaching the peak loads, which are much higher than field values, the pre-tensioning is reduced rapidly by moving the pulley into the opposite direction and a new cycle begins.

If the belt splice exceeds more than 180,000 revolutions it is judged to have reached the "fatigue strength".

The research is based on destructive testing of belt splices.

A fatigue strength curve is derived from a specific number of belts, each with the same kind of splice.

The test starts with the first belt at a very high peak load, and continues until the splice breaks. The second specimen is subjected to a lower peak load etc. A test series involving splices of the same kind generally produces a curve which is similar to that shown in (Fig. 9).

Splice research on the test stands is very time-consuming and costly because each test series includes up to 6 belts.

The aim of the testing is to verify the theoretical values and to increase the dynamic splice strength. Different splice designs, e.g.

- \* the individual step lengths
- \* the number of steps
- \* radically new splicing configurations
- \* varying intermediate rubber thicknesses

lead to different fatigue strength curves.

Fatigue strength has also been significantly increased due to improvements in the core rubber compounds.

The Institute of Conveying Technology (IFH) recommends that the local momentary highest tension on a belt, e.g. at the belt edges in the trough-to-flat transition, should be not higher than 50 % of the achieved fatigue strength, i.e. that a safety factor of at least 2 in relation to the dynamic fatigue strength can be permitted.

## 6.5 Results of splice investigations

Continuous research and development together with comprehensive testing procedures have enabled Continental to attain values of approx. 50 % of the nominal belt strength, even with high strength steel cable belts. These are values which until recently were only achieved by one or two-step splices in the low-strength range. (Fig.10)

The St 6600 that was installed at Palabora attained, with 50 % of the nominal belt strength or 3300 N/mm, the highest fatigue strength in absolute terms ever recorded at the Institute of Conveying Technology.

## 7. Examples of high-speed conveyors

Depending on the type of material being transported, it is possible to classify belts that run at more than 4 m/s as 'high-speed conveyors'.

### 7.1 Palabora Mining Company, RSA - St 6600

Surface-mined copper ore is transported out of the pit via a tunnel on a Continental steel cable belt, St 6600, 1800 mm wide, with a capacity of around 6500 t/h. The centre distance is 1100 m and the lift is about 300 m. Belt speed is slightly over 4 m/s. The belt has been in operation since 1988.

### 7.2 Prosper Haniel - underground mine, W. Germany - St 7500

Coal is transported on a steel cable belt, St 7500, 3700 m long with a lift of appr. 800 m. A single pulley head drive with an installed power of 2 x 3100 kW was chosen to drive the conveyor. The belt is 1400 mm wide and conveys 1800 t/h of coal to the surface, and on the return side 1000 t/h of overburden into the pit at a belt speed of 5.5 m/s. The belt was installed 1986.

### 7.3 Rheinische Braunkohle Werke - lignite mines, W. Germany - St 4500

Steel cable belts of St 4500 and 2800 mm width are used to move overburden from the Hambach open pit to the Fortuna open pit. The longest conveyor has a centre distance of appr. 5600 m. 12,000 kW of power is necessary to convey 37,500 t/h. The belts were installed between 1983 and 1986. Continental and other W. German belts are in use here. The speed of the conveyors is 7.5 m/s.



#### 7.4 Selby Coalfield, England - St 6950

In Selby, two parallel conveyors have been installed. One is fitted with a Cable-belt system, while the other, the South drift conveyor is equipped with a steel cable belt, St 6950. With a centre distance of 12,200 m, and a lift of appr. 800 m, a maximum of 2400 t/h of coal can be transported on a belt 1300 mm wide. With 10,100 kW of power, the belt is run at speeds up to 8.4 m/s. The unit was installed in 1982/83.

#### References:

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- [3] Beckmann, D. and Alles, R.: Calculation of transition lengths for  
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- [4] Flebbe, H.: Dynamic splice strength - design criteria for  
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Fig. 1 Elongation of various tension members, as exemplified by a belt with a strength of 1000 N/mm

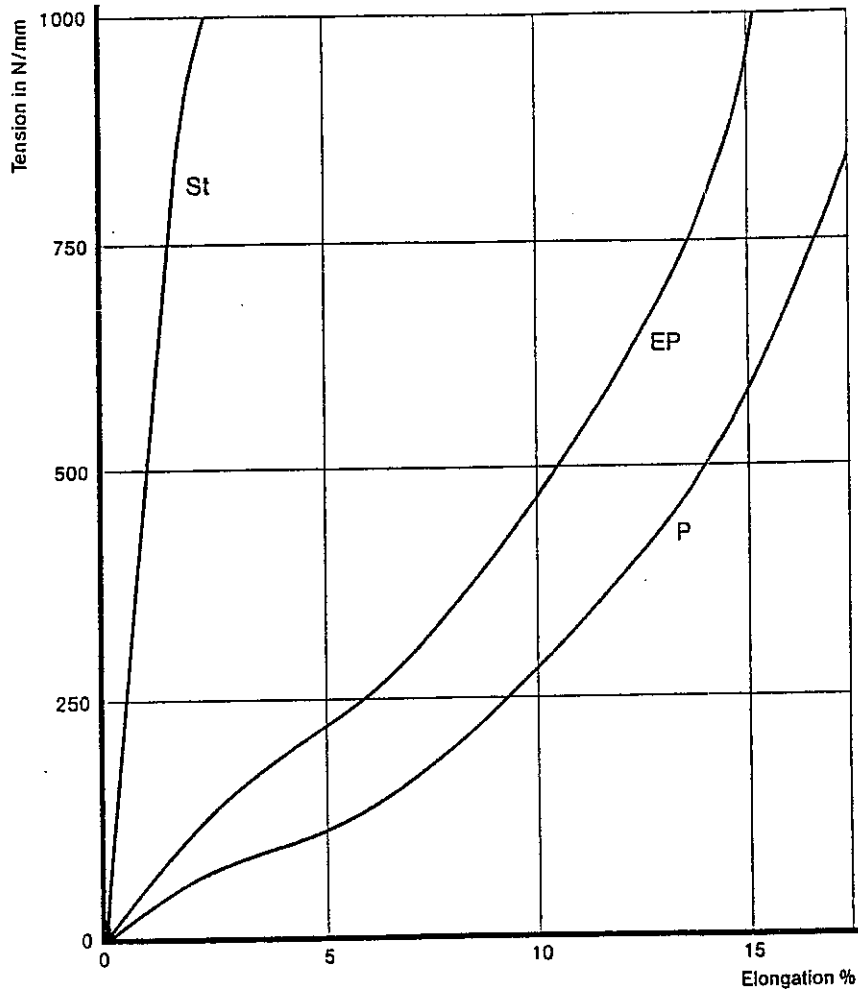


Fig. 2 Loading situations

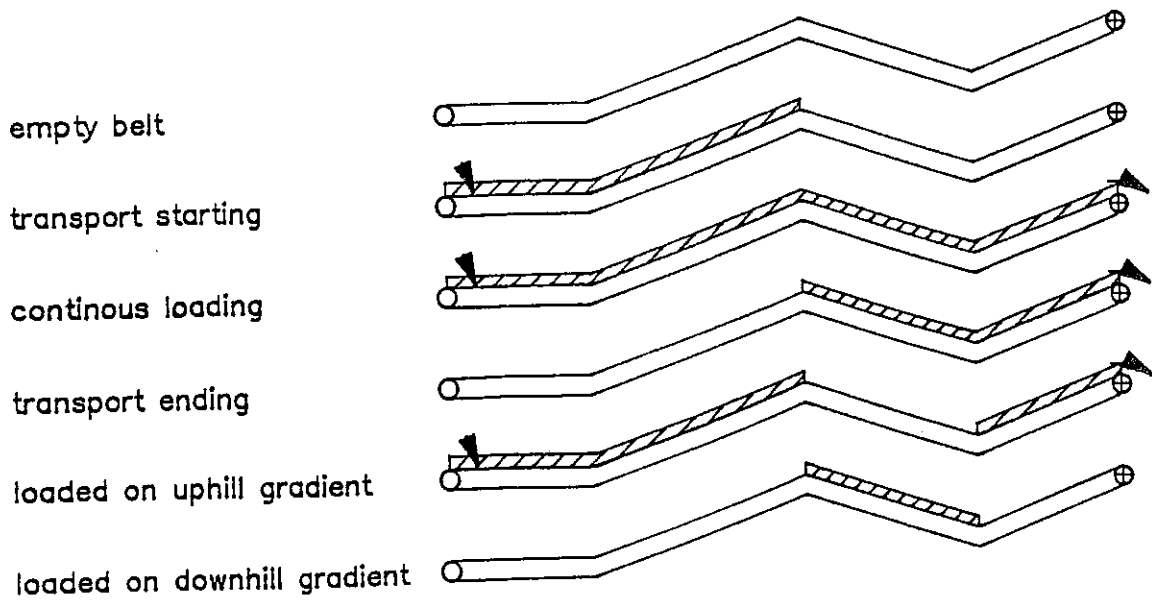


Fig. 3 Conveyor belt calculation

Continental - Application Engineering - Conveyor Belting			
Conveyor Belt Calculation			
Inquiry International Materials Handling Conference		Beltcon 5	
Feeding Data			
Capacity in m3/h .....	2500	Belt weight in kg/m.....	40.75
Bulk weight in t/m3 .....	1.60	Weight of Carr.Idlers in kg/m	14.86
Belt speed in m/s .....	5.50	Weight of Ret. Idlers in kg/m	6.83
Conveying length in m .....	1500.00	Friction Factor	
Section 1 length in m .....	1000.00	Carrying Side .....	0.020
Section 2 length in m .....	500.00	Return Side .....	0.020
Lift in m .....	50.00	Coefficient .....	1.060
Section 1 Lift in m .....	-50.00		
Section 2 Lift in m .....	100.00	Efficiency .....	0.90
Belt width in mm.....	1200	Idler Spacing in m .....	1.50
Results			
Masses to be moved			
	Section 1	Section 2	
Carrying Side - empty Belt .....	55618 kg	27809 kg	
Carrying Side - loaded Belt ....	257639 kg	128819 kg	
Return Side .....	47585 kg	23792 kg	
Kinetic Resistance			
	Section 1	Section 2	
Carrying Side - empty Belt .....	-9.0 kN	45.4 kN	
Carrying Side - loaded Belt ....	-68.5 kN	263.4 kN	
Return Side .....	29.3 kN	-35.3 kN	
Secondary Resistance .....	5.3 kN		
Driving Force and Power Input			
Belt loading	Driving Force	Power Pulley	Input Motor
Empty Belt .....	30.3 kN	167.0 kW	185.6 kW
Section 1 loaded .....	-23.6 kN	-130.3 KW	-117.2 kW
Loaded Belt .....	194.3 kN	1068.6 KW	1187.4 kW
Section 2 loaded .....	248.3 kN	1366.0 KW	1517.8 kW
Worked out by Ullmann		Phone (49)511-2141529	Date 8.8.89

Fig. 4 Calculation of belt tensions

Calculation of Belt Tensions										
Feeding Data										
			Power installed in kW	Power inserted in kW	Braking Force in kN	Arc of Wrap in degr.	Reduced Masses in t			
Head - Pulley	1	.....	800.0	800.0	-69.2	160	30.4			
	2	.....	400.0	400.0	-34.6	190	15.2			
Tail - Pulley	3	.....	400.0	400.0	-34.6	180	15.2			
	4	.....	0.0	0.0	0.0	0	0.0			
Startup Factor	..... 1.40		Tension/ensioning in kN ...				0.0			
Coefficient Start./Brak.	0.35		Limit of Belt Sag in %				1.0			
Working	... 0.30		Belt Sag empty/loaded in kN				7/ 44			
Manual Makeup										
RESULTS: BELT TENSIONS (KN)										
<div> <div> <div>Tail</div> <div>Head</div> </div> <div> <div>Carr. Side T 6</div> <div>Z 11</div> <div>T 1</div> </div> </div>										
<div> <div>Ret. Side T 4</div> <div>Z 12</div> <div>T 3</div> </div>										
<div> <div>Startup</div> <div>Working</div> <div>Braking</div> </div>										
<div> <div>Tail</div> <div>Head</div> <div>Tail</div> <div>Head</div> <div>Tail</div> <div>Head</div> </div>										
empty										
Belt	126	204	293	161	152	198	179	126	150	
	126		157	161		183	179		195	
ts = 3 s	194	91	89	169	140	175	156	164	218	
tb = 7 s										
<div> <div>Tail</div> <div>Head</div> <div>Tail</div> <div>Head</div> <div>Tail</div> <div>Head</div> </div>										
1. Section										
loaded	79	256	328	186	123	168	218	84	122	
	79		173	186		180	218		182	
ts = 5 s	156	82	95	180	151	186	187	171	213	
tb = 20 s										
<div> <div>Tail</div> <div>Head</div> <div>Tail</div> <div>Head</div> <div>Tail</div> <div>Head</div> </div>										
loaded										
Belt	96	118	424	141	78	341	228	0	180	
	96		251	141		244	228		230	
ts = 16 s	182	137	164	189	160	195	203	204	255	
tb = 8 s										
<div> <div>Tail</div> <div>Head</div> <div>Tail</div> <div>Head</div> <div>Tail</div> <div>Head</div> </div>										
2. Section										
loaded	100	111	423	116	107	370	169	92	198	
	100		251	116		246	169		230	
ts = 14 s	186	139	165	178	149	184	153	182	246	
tb = 4 s										
Calculation of Belt Safety Factors										
Belt Typ: St 2000 10/6 mm										
Nominal Strength at Splice: 2000 N/mm										
Safety Factor on start-up SS = 5.6										
Safety Factor during operation S = 6.4										
Safety Factor on braking SB = 9.3										

Fig. 5 Trough-to-flat transition

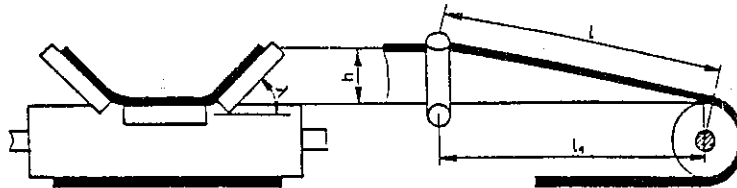


Fig. 6 Flat-to-trough transition

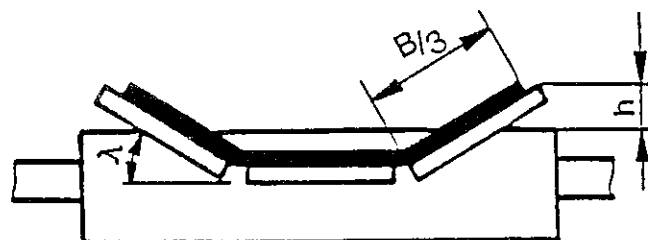
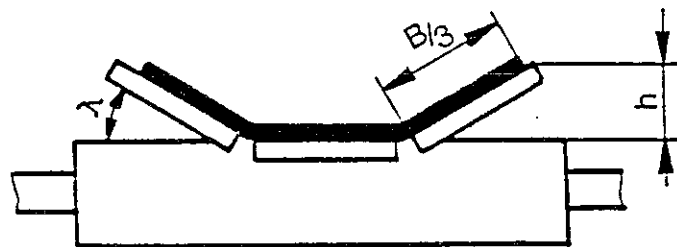


Fig. 7 Different 3-step splices

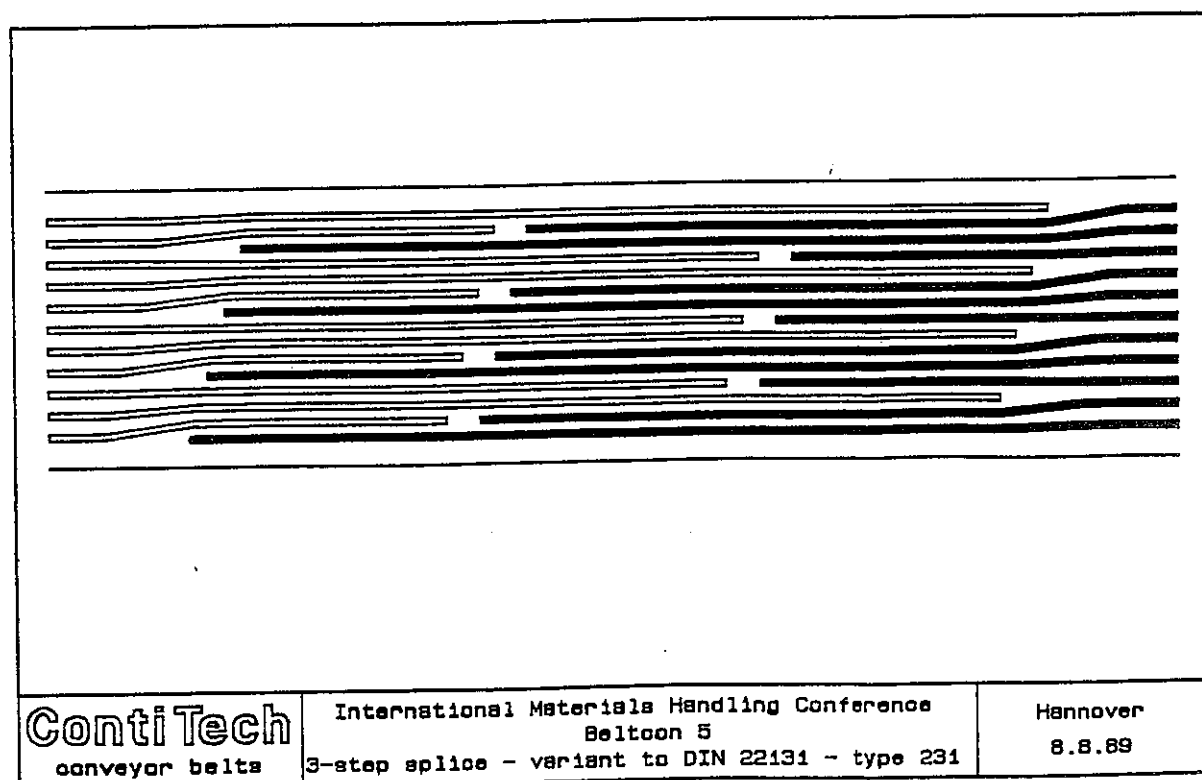
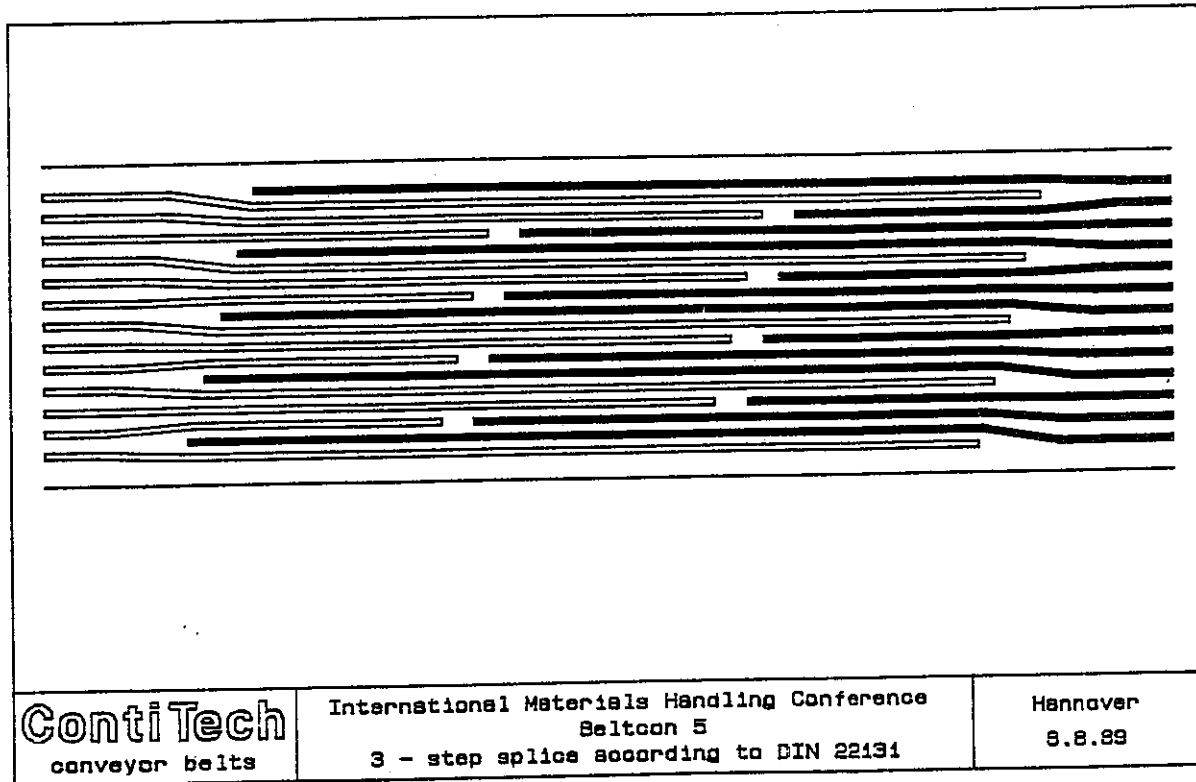


Fig. 8 Test stand at the Institute for Conveying Technology, University Hannover (IFH)

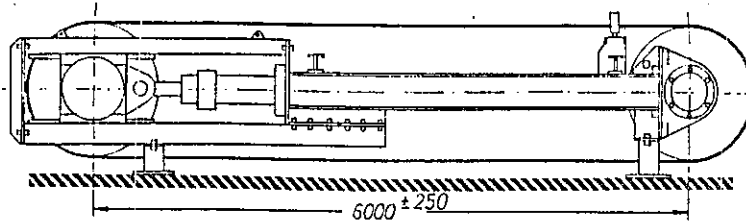


Fig. 9 Determining the fatigue strength of a splice with four samples of the same kind

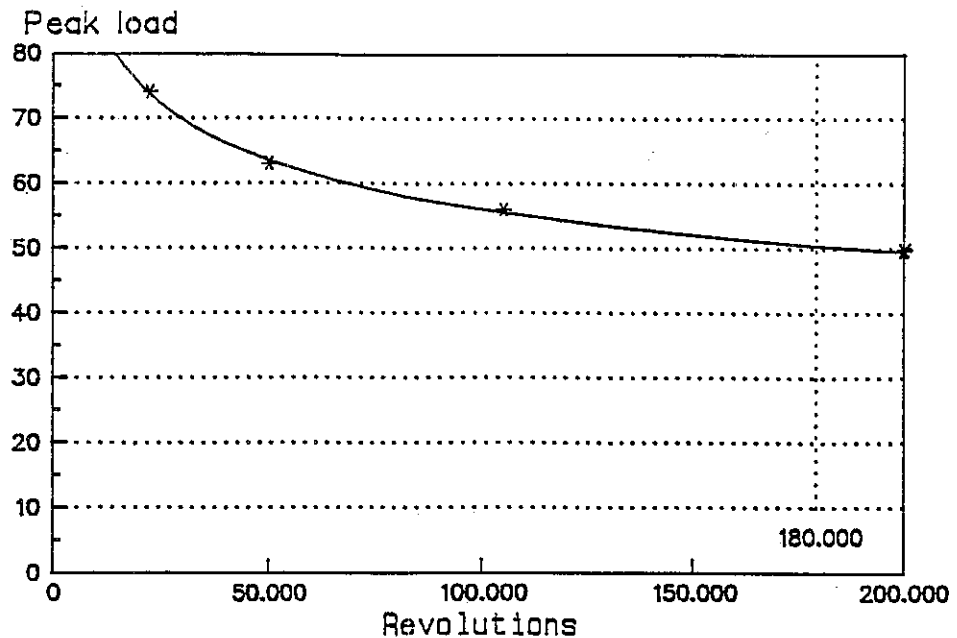


Fig. 10 Dynamic fatigue strength of belt splices, absolute values

