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Gabriel Lodewijks

Current Developments in Bulk Solids Handling

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Gabriel Lodewijks (ed.)
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Vogel Buchverlag

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CONTENTS

Preface...7

A Belt Conveying...9

A.1 Design Considerations to Reduce the Costs of Conveyor Systems, *C.A. Wheeler*................11
A.2 Determination of Rolling Resistance of Belt Conveyors using Rubber Data: Fact or Fiction?,*G.Lodewijks*..23
A.3 Indentation Rolling Resistance of Steel Cord Conveyors: A Pseudo 3D Viscoelastic Finite Element Analysis, *C.A. Wheeler & P.J. Munzenberger*..37
A.4 The possibilities of Decreasing the Belt Conveyors Main Drive Power Demand, *L. Gladysiewics & W. Kawalec*...49
A.5 Theoretical and Experimental Noise Examinations on the RopeCon System, *S. Wirth*.........55
A.6 Lay-out Considerations for Multiple Driven Belt Conveyor Systems, *A.J.G. Nuttall & G. Lodewijks*..61

B Pneumatic Conveying..69

B.1 Profiling the Dilute Phase Flow Parameters of Large Throughput Coke Suction Cranes, *M.G. Jones, K.C. Williams & S. Keys*..71
B.2 Horizontal Dense Phase Pneumatic Conveying of Bulk Solids, *L. Lecreps & K. Sommer*........79
B.3 Investigations on Single Slugs to Explain High Pressure Loss by Horizontal Dense Phase Pneumatic Conveying, *L. Lecreps & K. Sommer*...87
B.4 On the Modelling of Pressure Drop for Dense Phase Pneumatic Conveying of Powders, *S.S. Mallick & P.W. Wypych*..99
B.5 Pneumatic Conveyor Design – How Good is your Computed Aided Program, *D. Mills*.......113
B.6 Power requirements for Pneumatic Conveying Systems, *D. Mills*.....................................123

C Silo & Dry Bulk Terminal Technology..133

C.1 Experimental and Design Loads of Pressure of Bulk Materials against Silo Walls, *K. Mieczyslaw, M. Marek* 135
C.2 Cylindrical Corrugated Steel Silos in Brazil: Failure Modes, *C. Calil Jr., G. Palma & A.B. Cheung*..141
C.3 Avoiding and Curing Hopper Problems, *L. Bates*..149
C.4 Modern Coal Storage – a Safe and Efficient Storage Solution, *R. Spaargaren*...................161
C.5 Storage Facilities on Bulk Terminals: Open versus Closed Facilities, *D. Schott & G. Lodewijks*..169
C.6 Modern Dry Bulk Terminal Design, *G. Lodewijks, D.L. Schott & J.A. Ottjes*........................177

D Environmental Aspects......193

D.1 Environmental Management Accounting as a Selection Tool for Storage Systems, *D.L. Schott, O.C. Maan, R. Spaargaren, J. Ruijgrok, J.H. Welink, J.A.J.M. Deckers & G. Lodewijks*......195

D.2 Reducing Dust Emissions from Ship Holds during Loading of Bulk Materials, *S. Wiche, C.A. Wheeler, A.W. Roberts, T. Krull & D. Ilic*......207

D.3 Dust Explosion Protection using Flameless Venting, *J. Snoeys*......219

D.4 Noise Reduction of Nozzles by Use of Adequate Geometrics, *S.J. Wirth & M. Prenner*......229

D.5 Determination of the Dustiness Characteristics of Bulk Solids through the Use of Experimental Procedures and Test Apparatus, *M. Djukic*......237

D.6 Continuous Particulate Emission Monitors for Industrial Processes, *V. Schmitz*......247

PREFACE

The first international conference in Europe dedicated to Storing, Handling and Transporting Bulk Solids Material, *BulkEurope2006*, was held in Barcelona on the 5th and 6th of October, 2006. The theme of *BulkEurope2006* was innovation. *BulkEurope2006* aimed at bridging the gap between academia and industry by providing a forum where new ideas, concepts, methods and technologies were shared and exchanged between the academic institutes, the engineering and consultancy companies and the end users of bulk material handling, transporting and storing systems. It also served as a networking platform for all attendants to make new contacts and to refresh the old ones, to strengthen and enlarge their private network and broaden their possibilities. The scope of *BulkEurope2006* was all aspects of design, operation, maintenance, surveillance and automation of bulk materials handling plants and individual parts of equipment for bulk material transport, storage, handling and mechanical processing in various industries such as mining, cement, power generation, steel making, ship-, rail-, and road transport and others. *BulkEurope2006* turned out to be a great success.

In 2008, on September the 11th and 12th, the second edition of the conference, now called *BulkEurope2008*, was organized in Prague, the Czech Republic. Where BulkEurope2006 was dedicated to Innovation, *BulkEurope2008* was dedicated to the environmental impact of systems used for Storing, Handling and Transporting Bulk Solids Material. The focus of the conference was on technologies, methods and measures used to reduce the generation of dust, spillage and noise. The aim and scope of *BulkEurope2008* was the same as those of *BulkEurope2006*. Once again, *BulkEurope 2008* succeeded in bringing together an audience from academia and industry. Many fruitful discussions, new contacts and meeting old friends from all over the world were the outcomes!

On both conferences together over hundredth papers were published and presented. The experience of the editor of this book however, who chaired *BulkEurope2006* and *BulkEurope2008*, is that the reach of conference papers is limited. Therefore it was decided to collect the best and most informative papers and publish them in book form. This book, titled *Current Developments in Bulk Solids Handling*, is the results of this exercise. The number of papers, reworked into chapters, presented in this book is limited because the size of the book had to be limited. Therefore four main topics were selected focussing on innovation and the environmental impact of bulk material handling, transporting and storing systems: A: belt conveyors, B: pneumatic conveyors, C: silo and dry bulk terminal technology and D: environmental aspects. It is however acknowledged that on both conferences many more excellent papers were presented on other than the four mentioned topics.

In 2010 the third edition of the conference, logically name BulkEurope2010, will be held in Glasgow, Scotland. I hope that this conference will be equally successful and that the bulk material handling, transporting and storing community once more finds a forum and a meeting place to exchange ideas and new developments. I look forward to see you all again in Glasgow in 2010.

Yours sincerely,

Delft, February 2010

Prof.dr.ir. Gabriel Lodewijks (Editor)
Chairman of BulkEurope 2006/2008/2010

A: BELT CONVEYING

Current Developments in Bulk Solids Handling
G. Lodewijks (Editor)

A.1 Design Considerations to Reduce the Costs of Conveyor Systems

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1 INTRODUCTION

Belt conveying systems are used extensively in the mining and minerals processing industries to continuously transport bulk material. The investment and ongoing costs associated with these systems are substantial and can represent a significant proportion of overall plant costs. This chapter will discuss factors that influence the motion resistance of belt conveyors, and therefore the energy costs associated with the operation of these systems. While energy costs are an important design consideration, the economic performance of the system over the life of the installation should also be considered and will be briefly discussed using life cycle cost analysis.

The motion resistances that occur along the length of the conveyor are known as the main resistances and include the belt and bulk solid flexure resistance, the rotational resistance of the idler rolls and the indentation rolling resistance of the conveyor belt. Generally the indentation rolling resistance component is the major contributor to the motion resistance of long horizontal belt conveyors. Typically the next highest is the resistance due to the flexure of the bulk solid, followed by the rotating resistance of the idler rolls, and then the belt flexure resistance. The magnitudes of the main resistances are strongly influenced by factors such as belt speed, belt width and type, idler diameter and spacing, etc. The selection of these variables is in the hands of the conveyor designer and is traditionally approached from an empirical standpoint.

This chapter will discuss the influence of particular belt conveyor variables on the resistance to motion of belt conveyors, and thus the energy consumption. The aim of this chapter is to inform the conveyor designer of the influence of these variables and therefore provide the opportunity to reduce the energy consumption at the design stage. However, while reducing the motion resistance is an important consideration, the conveyor design should also be made with consideration to the capital cost of the system. As a result the economic performance of belt conveyor systems will be discussed using an established economic model based on life cycle costs developed by Roberts et al. ([1],[2]).

2 BULK SOLID AND CONVEYOR BELT FLEXURE RESISTANCE

Bulk solid and conveyor belt flexure resistance occurs along the length of the conveyor as the belt and the bulk solid undergoes transverse and longitudinal displacement due to belt sag. As the belt progresses from one idler set to the next the bulk solid undergoes cyclic expansion and contraction in the transverse direction, in addition to variation in height in the longitudinal direction. The relative movement of the bulk solid results in energy losses due to the internal friction of the bulk material, while the movement of the conveyor belt results in losses due to the viscoelastic nature of the rubber belt. The losses due to the flexure of the conveyor belting will not be discussed in this chapter due to the small amount that this component adds to the motion resistance, as noted by Wheeler [5].

For the purpose of analysis, bulk solid flexure can be considered to consist of both transverse and longitudinal resistance components. When the belt is supported by an idler set, as indicated by positions A and E in Figure 1, the bulk solid is forced to conform to the troughing profile, resulting in transverse compressive stresses. As the belt moves to position B, the troughed belt opens under the action of gravity allowing the bulk solid to relax transversely forming an active stress state. Longitudinally, however, the bulk solid is undergoing compressive stress due to the contraction of the bulk solid arising from the sag of the belt. Upon reaching approximately 50% to 60% of the idler spacing, as indicated by position C, the stress states theoretically reverse. A passive stress state is induced in the transverse direction due to the compressive stresses caused by the narrowing profile of the belt, while the bulk solid in the longitudinal direction dilates generating an active stress state as it moves away from the point of maximum sag.

The cyclic transverse and longitudinal flexure of the bulk solid results in flexure losses due to internal friction and friction at the belt and bulk solid interface.

In order to calculate the flexure resistance of the bulk solid the forces generated from the relative movement of the bulk solid need to be resolved, taking into account the properties of the bulk solid and the conveyor belt. The magnitude of the belt deflection, troughing configuration and belt speed each contribute to the amount of belt and bulk solid flexure, and therefore, the flexure resistance. Bulk solid properties including bulk density, bulk solid surcharge angle, internal friction and friction at the belt interface determine the pressure distribution acting on the belt and the losses attributable to the relative movement of the bulk solid.

Spaans [3] was the first to provide an analytical model to calculate the flexure resistance of the bulk solid due to the cyclic transverse and

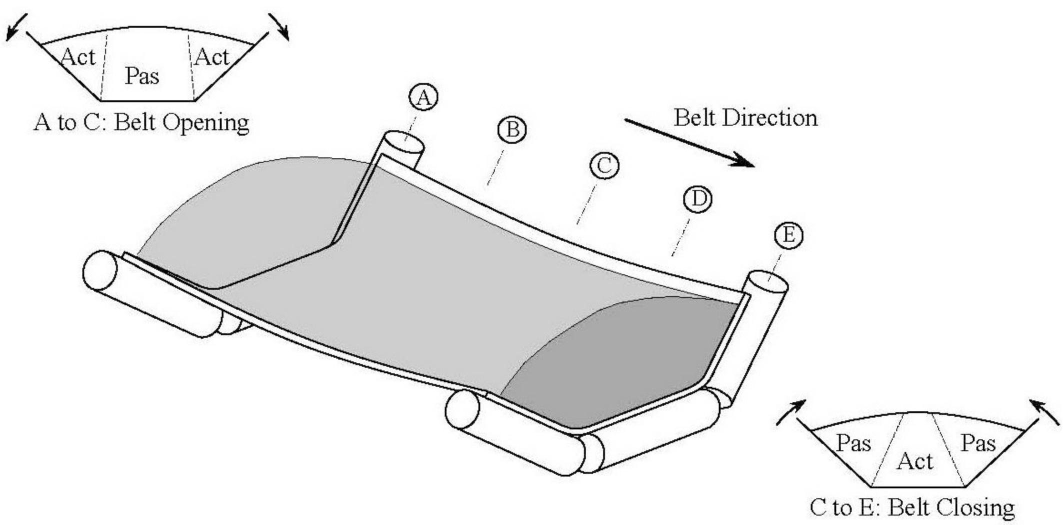


Figure 1: *Induced active and passive stress states for a loaded conveyor belt.*

longitudinal deformation. The transverse flexure resistance is modeled by calculating the difference between the work done during the opening and closing of the belt, as the belt moves between consecutive idler sets. The normal forces acting on the side idler rolls are calculated using a method developed by Krause and Hettler [4], who provide an analysis of the total force acting on the idler rolls due to the formation of active and passive stress states within the cross-section of bulk solid. Spaans [3] also calculated the longitudinal flexure resistance. This analysis involved considering an elemental volume of bulk solid with vertical boarders on top of a flat belt. As the element moves between successive idler sets it undergoes compressive forces due to the belt sag, and since the bulk solid possesses internal friction, energy is absorbed in the bulk solid in the form of flexure resistance.

The results presented in this chapter were generated from the method detailed by Wheeler [5]. This method adopts a similar approach to that of Spaans [3] by individually calculating the transverse and longitudinal components of the bulk solid flexure resistance. The analysis uses orthotropic plate mechanics to calculate the belt deflection to provide a means of predicting the flexure resistance due to the relative movement of the bulk solid. A similar approach to that of Krause and Hettler [4] is used to predict the active and passive stress states that are formed within the bulk solid as the belt opens and closes between successive idler sets. The pressure factors given by Krause and Hettler [4] are used, but rather than calculating the resultant normal force acting on the conveyor belt due to the induced stress states, the analysis calculates the pressure distribution over the surface area of the conveyor belt.

Figure 2 represents typical results generated from the program. This example demonstrates the influence of the internal friction angle and idler spacing. While the conveyor designer typically has little control over the properties of the bulk solid being conveyed, and in particular the internal friction angle, it is still worth noting its influence on the bulk solid flexure resistance. As the internal friction angle increases the ratio between the passive and active stress factors also increases. This has the effect of increasing both the longitudinal and lateral components of the bulk solid flexure resistance.

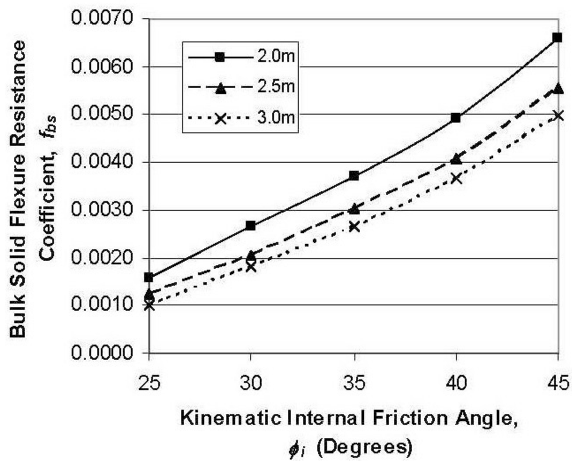


Figure 2: *Bulk solid flexure resistance coefficient versus kinematic internal friction angle for a range of idler spacings. (belt speed = 5m/s, belt width = 1.2m, sag ratio = 2%, bulk density, ρ = 1000kg/ m^3, friction angle with the belt conveyor, φ_w = 30°).*

Figure 2 also shows the reduction in the bulk solid flexure resistance coefficient with increasing idler spacing. This occurs since the magnitude of the flexure resistance per idler set only increases marginally with idler spacing since in the present example the sag ratio is maintained at 2%. Consequently, the flexure resistance force per unit length decreases with increasing idler spacing providing belt tension is increased accordingly to maintain 2% sag.

An important aspect of the analysis is the allowance for the influence of belt speed. As indicated previously the transition between the stress states typically occurs at 50% to 60% of the idler spacing. The exact location of the transition is heavily dependant on the belt speed since as the belt speed increases the transition, and therefore the point of maximum sag moves further away from the midpoint of the idler spacing. To account for these dynamic effects in the program an iterative procedure is employed. The procedure initially assumes that the transition occurs at the midpoint of the idler spacing and then with each iteration the profile of the belt alters as a result of the momentum of the moving bulk solid. Typically at high belt speeds the transition will occur at 55 to 60% of the idler spacing. Since the bulk solid flexure resistance is calculated from the difference between the work done during each stress state, increasing belt speed has the effect of increasing bulk solid flexure resistance, as shown in Figure 3. The increasing flexure resistance occurs for each of the sag ratios shown and is slightly more pronounced with higher sag ratios.

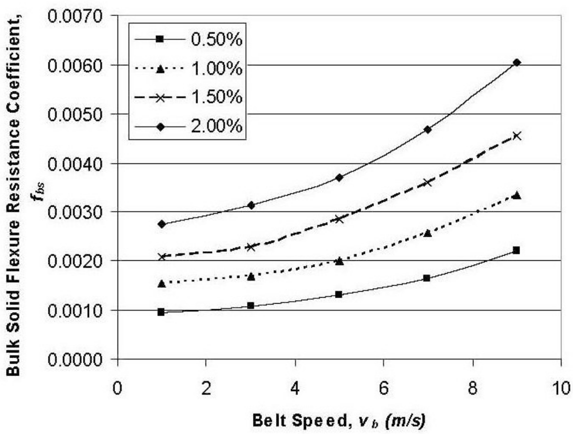


Figure 3: *Bulk solid flexure resistance coefficient versus belt speed for a range of sag ratios. (belt width = 1.2m, idler spacing = 2m, kinematic internal friction angle, φ_i = 35°, bulk density, ρ = 1000kg/m^3, friction angle with the belt conveyor, φ_w = 30°).*

Given the influence of particular conveyor variables on the bulk solid flexure resistance it is clear that the conveyor designer has control over many of these at the design stage. While the bulk solid properties are typically design constraints, other variables such as idler spacing, troughing configuration, belt speed and tension are able to be selected by the conveyor designer to minimise the life cycle cost of the belt conveyor system.

3 ROTATING RESISTANCE OF IDLER ROLLS

Predicting the cumulative resistance of idler rolls is vitally important in calculating the belt tension and therefore power requirements of a system, particularly on long overland conveyors where there are typically more than one thousand idler rolls per kilometer of belt. The rotating resistance occurs due to the friction of the rolling elements in the bearings, the viscous drag of the lubricant and the friction of the contact lip seals.

The rotating resistance of the idler rolls is primarily dependent on the seal type and configuration, the type of bearings, the temperature of the lubricant and the rotational speed of the idler roll. Contact lip seals and grease filled labyrinth seals form the boundary preventing dust and water ingress into the rolling elements of the bearings. The labyrinth seals are usually packed with grease to optimize the sealing efficiency of the labyrinth, resulting in viscous drag generated from the shearing of the grease between the layers of rotating and stationary surfaces. An outer contact lip seal typically forms the primary boundary between external contaminants entering the labyrinth seal, while an inner lip seal contains the lubricating grease within the bearing. The outer and inner contact lip seals add to the rotating resistance of the idler roll due to the nature of the sealing mechanism. In addition to the resistances associated with sealing, conventional idler rolls use rolling bearings where the friction primarily depends on the bearing type and size, the operating speed, the properties and quantity of the lubricant and the load. The total resistance to rolling in a bearing is made up of the rolling and sliding friction between the rolling elements and the cage and guiding surfaces and, the friction in the lubricant, as noted by Palmeren [6].

Wheeler [5] provides methods to calculate the individual components of the rotating resistance of the idler rolls. The analysis provides theoretical estimates for the friction due to the bearing, labyrinth seals and lip seals. The contribution of the labyrinth seal viscous drag is approximated by calculating the torque required to shear the grease using a force momentum balance for a Newtonian fluid. The resistance due to the rolling bearings is approximated by calculating the no-load and load moments acting, while the contact lip seal resistance is calculated from a derived empirical formula. In addition to providing a theoretical approximation of the rotating resistance, an experimental apparatus to measure the rotating resistance of conveyor idler rolls under simulated operating conditions was also developed. Figure 4 shows a photograph of the test facility.

The idler roll to be tested is supported on knife-edge supports that enable the vertical force at each end of the shaft to be measured independently using load beams. Collars are attached to each end of the idler shaft that rest on the knife-edge supports and allow the shaft to rotate freely about the knife-edge. The rotating resistance of the idler roll is measured using a load beam which measures the torque required to hold the shaft stationary. A flat drive belt applies a vertical load and a driving torque to the idler through a variable speed drive, which can be ramped up to the required belt speed to represent the starting characteristics of the conveyor. The flat drive belt has the added advantage of damping the vibrations induced from the radial runout of the idler roll. The vertical load is provided by a pivoted mass carrier and can be applied at any position along the length of the idler roll. The device can accept idler rolls up to 1250mm long and 178mm in diameter.

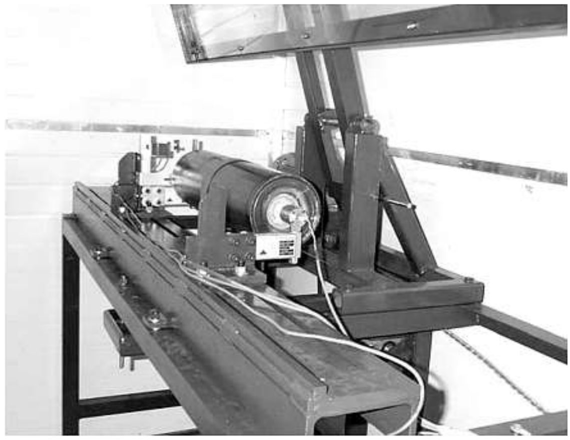
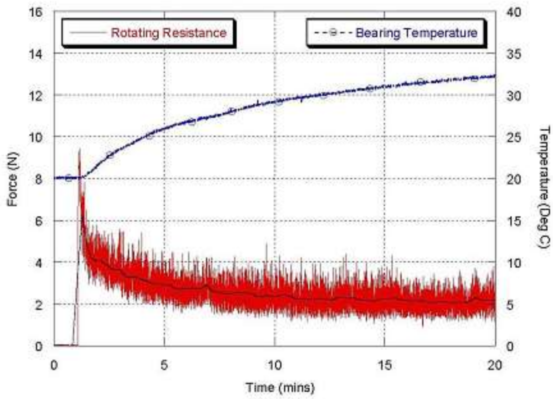
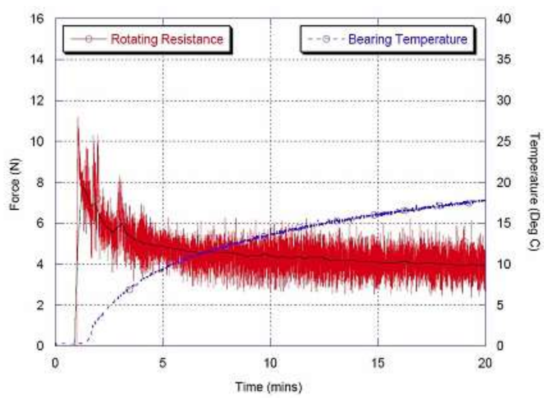


Figure 4: *Idler rotating resistance measurement apparatus.*

The measurement apparatus is housed within a temperature controlled room where the ambient temperature can be set from -10°C to +60°C. Additionally, if required the bearing temperature can be monitored using a thermocouple located beneath the inner race of the bearing. The test procedure involves measuring the rotating resistance for a particular idler roll under simulated operating conditions. The idler roll is subjected to the required vertical load and ramped up to an equivalent operating belt speed over the specified conveyor starting time. The test is carried out until the rotating resistance force stabilises signifying the equilibrium temperature for the grease. The test is repeated over a range of ambient temperatures determined from the climate in which the conveyor is to be operated. Typically new conveyor idler rolls are required to be run for a period of time to ensure the lip seals are worn in and the grease is allowed to circulate under normal operating temperatures.



(a) 20 °C ambient temperature



(b) 0 °C ambient temperature

Figure 5: *Idler roll rotating resistance versus time for a Ø152mm at 6m/s.*

3.1 Experimental Results

The following test results are presented to demonstrate the use of the measurement apparatus for determining the rotating resistance of idler rolls. Figure 5 shows the total rotating resistance for a ∅152mm idler roll operating at 6m/s and ambient temperatures of 20 °C and 0 °C. Both results clearly demonstrate the reduction in rotating resistance with running time, as the temperature of the grease within the bearings and the labyrinth seals increase. The roll tested at the lower ambient temperature will stabilize at a higher rotating resistance force. The rate at which the rotating resistance reaches a constant value is dependent on the properties of the grease, belt speed and ambient temperature. The influence of the temperature of the grease on the overall rotating resistance will vary depending on the contributions of the viscosity dependent factors. The variation in rotating resistance with both time and ambient temperature clearly demonstrate the need to undertake testing at simulated operating conditions.

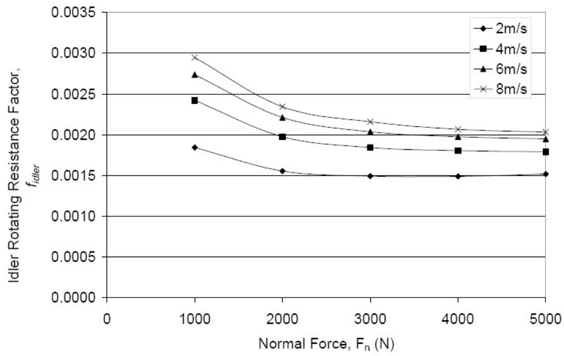


Figure 6: *Idler rotating resistance factor for a Ø152mm idler roll with 6307 series deep groove ball bearings shown for a range of belt speeds and normal loads.*

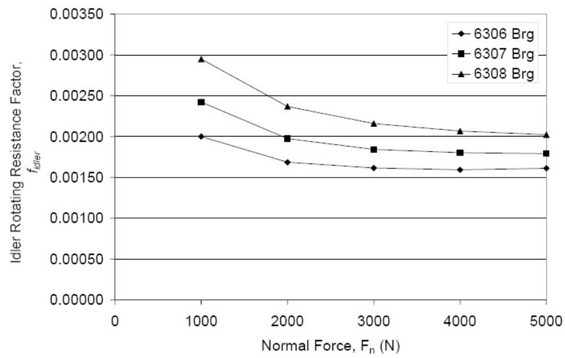


Figure 7: *Idler rotating resistance factor for a Ø152mm idler roll operating at 4m/s for a range of normal loads and a variety of deep groove ball bearings.*

Figure 6 shows the calculated rotating resistance factor versus normal load for a Ø152mm idler roll with 6307 series deep groove ball bearings for a range of belt speeds. The rotating resistance factor increases with belt speed primarily since the shear rate of the grease increases. Furthermore the rotating resistance factor increases to a lesser extent with increasing normal load.

Figure 7 shows the calculated rotating resistance factor versus normal load for a Ø152mm idler roll operating at 4m/s with three commonly used deep groove ball bearings. The larger bearings show greater rotating resistance factors for the range of normal loads. Also the rotating resistance factors show a decrease with increasing normal force, proving that the force per unit length will decrease with increasing load up to a particular limit, such as in the case of increasing idler spacing. Evaluations of this type should be made in consultation with the idler roll manufacturer to determine maximum operating load.

4 INDENTATION ROLLING RESISTANCE

Indentation rolling resistance occurs due to the viscoelastic nature of the bottom cover of the belt. As the belt travels over the idler roll the bottom cover of the belt is indented due to the weight of the belt and bulk material. The cyclic indentation of the bottom cover of the belt as it passes over the idler rolls generates a resistance to motion due to the formation of an asymmetric pressure distribution within the contact area of the idler roll and belt due to hysterisis losses.

A finite element method has been developed to calculate the indentation rolling resistance by calculating the asymmetric pressure distribution within the contact area of the rubber belt and the idler roll. The roll diameter will prescribe the lower boundary conditions, while an iterative procedure is used to increase the depth of indentation until a specified vertical load is reached. Analyzing stress and deformations in the contact region relies on the formulation of the stress-strain-time relation for the response of the bottom cover material. The analysis assumes that the bottom cover of the belt is homogeneous and isotropic. The speed of the belt is considered to be constant so a steady state viscoelastic stress analysis may be applied.

Given the initial displacements of the nodes within the contact zone, the vertical forces acting on the boundary nodes in the contact zone are calculated. As a result of the viscoelastic response of the bottom cover material, the contact length will not be symmetric about the centerline of the roll and there will be less contact on the exit side of the idler roll. Consequently, since the contact zone is initially assumed to be symmetric, negative forces occur at nodal points on the exit side of the idler roll, representing induced tension as the nodes are forced to conform to the profile of the idler roll. Tensile forces are not possible without adhesion between the roll and the cover material and, therefore any negative forces acting on nodal points along the boundary in the contact zone are reassigned as free boundary nodes.

The horizontal force generated by the load acting within the contact zone is calculated from the sum of the vertical forces about the centerline of the idler roll. The sum of the vertical forces, acting at the nodal points within the contact zone results in a moment acting to oppose the motion of the conveyor belt due to the resulting asymmetric force distribution. The total horizontal force acting at the interface between the belt and the idler roll is equivalent to the indentation rolling resistance force.

4.1 Experimental Results

In order to verify the linear viscoelastic finite element analysis experimental tests were undertaken on a recirculating conveyor belt test facility at The University of Newcastle, Australia. Figure 8 details the measurement method which enables the measurement of the indentation rolling resistance force for a sample of flat

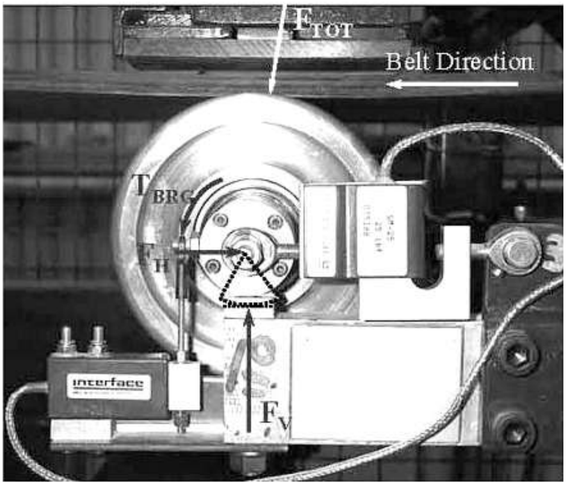


Figure 8: *Indentation rolling resistance measurement details*.

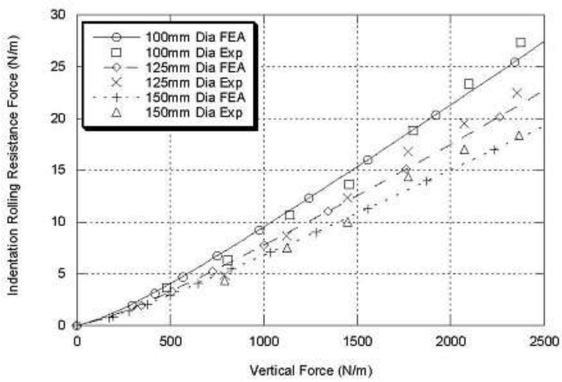


Figure 9: *Indentation rolling resistance force versus vertical force; FEA and experimental results for Ø100 mm, Ø125 mm and Ø150 mm idler rolls at a belt speed of 2 m/s*.

conveyor belt on the recirculating test facility. The belt speed, idler roll diameter and vertical load can be varied and the influence of each parameter measured and compared with that predicted by the finite element analysis.

The total horizontal force acting on the idler roll pictured in Figure 8 is due to the indentation rolling resistance and the rotating resistance of the idler roll. The total horizontal force is measured using an instrumented idler roll which also measures the idler rotating resistance as a separate component, enabling the indentation rolling resistance to be isolated. The idler rolls are supported at each end by a collar that is attached to the shaft. The collar is supported on a knife-edge and rocker support that enables the vertical force, F_V to be measured by a load beam while the horizontal force, F_H is measured by a s-type load cell, detailed in Figure 8. The rocker support facilitates measurement of the vertical force while allowing horizontal movement which is only restricted by the s-type load cell. The knife-edge support allows the idler shaft to rotate freely about the knife-edge but is restricted by the load beam which measures the torque, T_{BRG} resulting from the rotating resistance of the idler roll.

To verify the finite element analysis a structured testing program was undertaken to test a range of operating variables. The results presented in this chapter are for tests conducted at $20^{o}C$ using an SBR bottom cover compound, using a range of idler roll diameters and belt speeds. Figure 9 shows calculated and measured indentation rolling resistance forces plotted against the applied vertical loads for a belt speed of 2 m/s.

The test results show a good correlation between experimentally measured values and those calculated using the finite element analysis. Experimentation has shown that for high belt speeds and high vertical loads, the finite element analysis can over estimate the indentation rolling resistance force, since the influence of the lagging tail in the recovery zone is not fully modelled. Research has shown that higher belt speeds can be more accurately modelled by extending the length of the analysis zone, however this comes at the expense of additional computational time, and is an area of ongoing research.

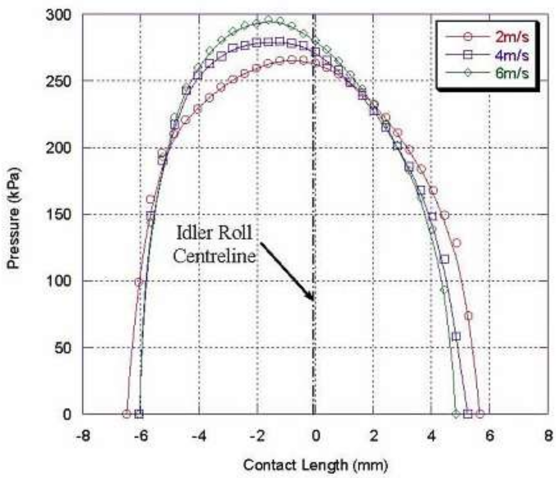


Figure 10: *Calculated pressure distribution for a Ø150 mm idler roll under a simulated vertical load of 2.5 kN/m at belt speeds of 2, 4 and 6 m/s.*

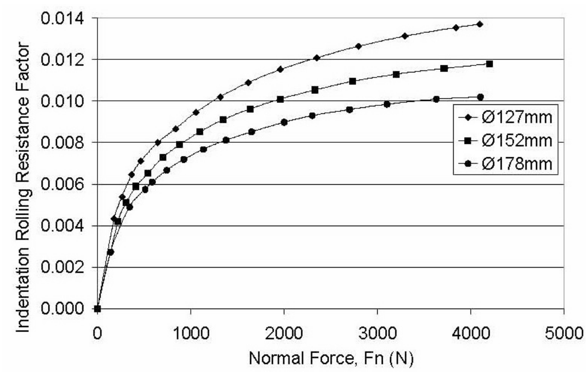


Figure 11: *Calculated indentation rolling resistance factor versus vertical force for Ø127mm, Ø152mm and Ø178mm idler rolls at a belt speed of 4m/s.*

As mentioned earlier, the resistance to motion comes about due to the hysteresis losses and the formation of an asymmetric pressure distribution within the contact zone. The asymmetry of the pressure distribution is clearly evident in results shown in Figure 10, and becomes more pronounced with increasing belt speed. Furthermore, as the belt speed increases, the magnitude of the pressure distribution increases since the same vertical load is applied over a smaller contact length.

Another important variable is idler roll diameter, which not only influences the rotating speed of the roll and therefore the rotational resistance, but more importantly the indentation rolling resistance. The results shown in Figure 11 highlight the influence of the idler diameter on the indentation rolling resistance. The results clearly demonstrate the advantage of larger roll diameters to reduce indentation rolling resistance. Furthermore, the relationship with normal load demonstrates a gradual decrease with increasing load. The finite element analysis provides an ideal mechanism to evaluate the influence of each variable at the design stage.

5 ECONOMIC CONSIDERATIONS

Clearly from the foregoing analysis the selection of particular variables can have a significant influence on the motion resistance of a belt conveyor. While reducing the energy consumption is an important consideration, the conveyor design should also be made with consideration to maintenance and capital costs of the system. As a result the economic performance of a belt conveyor system is suited to evaluation using life cycle cost analysis.

Roberts et al [1][2] provides a detailed economic analysis of belt conveyor systems based on life cycle costs. Cost functions were derived to take into consideration the energy costs and annual equivalent costs of conveyor components for the design life of the system. Component life, salvage value, taxation rate, and rate of return were considered in the latter. Optimum designs for a minimum annual equivalent cost were determined based upon performance, and geometric and design constraints.

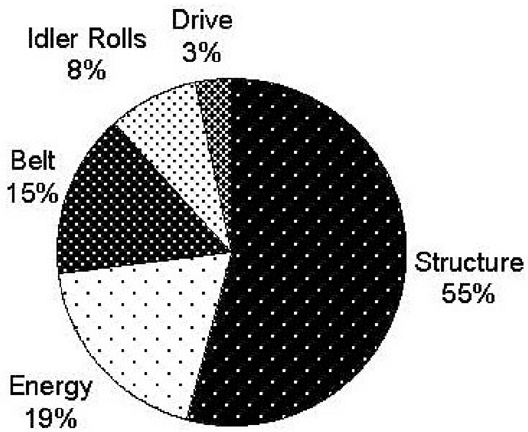


Figure 12: *Breakdown of annual equivalent component costs for a horizontal steel cord belt, throughput = 1000t/hr, length = 1000m, belt speed 3m/s, belt width = 1m (Roberts [2]).*

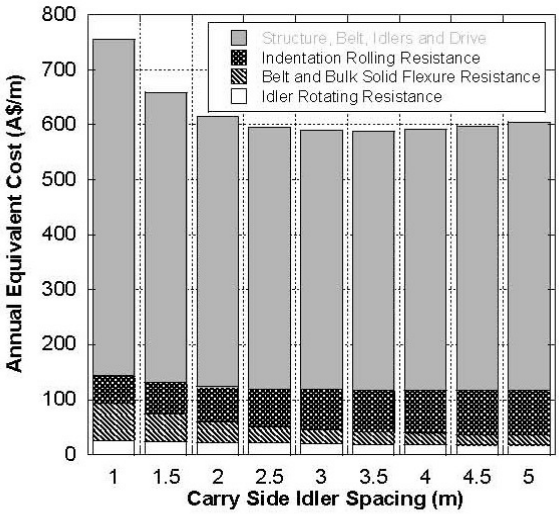


Figure 13: *Annual equivalent costs for a horizontal steel cord conveyor belt versus idler spacing. Throughput = 1000t/hr, length = 1000m, belt speed 3m/s, belt width = 1m.*

Roberts [2] provides an example of a horizontal steel cord belt conveyor with a throughput of 1000t/hr, length of 1000m conveying bulk material with a bulk density of 850kg/m^3. A breakdown of the annual equivalent costs for each of the major components versus belt width is derived, from which a belt width of 1m is shown to be the optimum belt width for this particular system. Figure 12 shows a breakdown of the annual equivalent component costs for this conveyor. The analysis was undertaken using a modified method of ISO5048 where the idler spacing is set based on the bulk density of the material and the belt width. When a detailed analysis of each of the contributing components of the main resistance is undertaken as described in this chapter, the annual equivalent cost verse carry side idler spacing is shown in Figure 13. The annual equivalent costs are based on the data given by Roberts [2] and have been adjusted for inflation using the Australian Consumer Price Index to approximate present day values.

It should be noted that the costs will vary significantly depending on the structural requirements for the particular installation. The example given by Roberts [2] is for a structure with walkways on both sides of the conveyor extending the length of the system. Other installations may require elevated or suspended structures, covers over the belts, or in the case of overland systems, minimum additional structure. These requirements will clearly influence the breakdown of the annual equivalent costs and consequently the optimum design configuration.

For the particular installation described an idler spacing between 3.0m and 3.5m is found to be optimum. The solution obtained was achieved using a fixed idler roll diameter, troughing configuration, etc, thus leaving the selection of further components which can also be optimized in a similar manner.

6 CONCLUSION

This chapter has discussed the influence of key conveyor components on the motion resistance of belt conveyors. Each of the contributing components of the motion resistance were described and the key influencing factors discussed. To highlight the influence that just one variable can have on the cost of a belt conveyor installation, an example was provided to show the influence of idler spacing on the annual equivalent cost of a long horizontal belt conveyor.

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A.2 Determination of Rolling Resistance of Belt Conveyors using Rubber Data: Fact or Fiction?

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1 INTRODUCTION

A belt conveyor is a mechanical conveyor frequently and worldwide used to continuously transport a certain material or people from a place A to a place B at a capacity C. When ordering a belt conveyor, a client normally is concerned about issues like performance (can we move C from A to B?), reliability, maximum wear rates, total cost of ownership, complexity of the system etc. During large projects the client normally provides specifications but does not specify specific types or sizes of components, although most major clients have a preferred supplier list. Assuming that the performance, reliability, maximum wear rates etc. are guaranteed by the belt conveyor supplier, they can select the actual component types and sizes.

To reduce the investment and operating costs of a belt-conveyor system it is important to determine and analyse the influences of the plant parameters and the operating parameters on the energy consumption. In terms of the indentation rolling resistance this implies that the dependence of this resistance on the roll radius, idler spacing, belt speed and radius of curvature should be known. It is also important to know the influence of the belt material and belt structure on the indentation rolling resistance and therefore on the energy consumption of the belt.

One of the most important components of a belt conveyor is the conveyor belt itself. The conveyor belt can make up till about 70% of the costs of a conveyor and the rolling resistance associated with the rubber (the indentation rolling resistance) can account for about 50% of the total rolling resistance [1]. The selection procedure of the conveyor belt should therefore be taken seriously.

It is well known that using standardised design methods like DIN 22101 or CEMA to calculate the power consumption of a belt conveyor generally leads to an overestimation of the power consumption and thus of the belt tensions. One reason is that these design methods fail to take the viscoelastic or mechanic/dynamic rubber compound properties into account. They can therefore not distinguish between the power characteristics of a belt made off one rubber compound or the other. Since the late fifties of the last century quite a few researchers worked on models that can be used to predict that part of the rolling resistance that stems from the rubber compound: the indentation rolling resistance. The use of these models provided insight into the nature of this resistance [2]. With this insight models have been developed that enable a link between the mechanic/dynamic properties of rubber compound and the later systems power consumption [3].

Although unknown power consumption may seem only a matter of costs, it also seriously affects the conveyors performance. Knowledge of rubber compound properties is therefore important because it partly determines the size and settings of components like motors and brake systems. For example, the application of a low loss rubber compound on a belt of a long overland system is a good way to reduce the overall operating costs. In case of an incline belt conveyor however, the extra costs of belting are not worth the effort since most of the power is used to raise the material. The total power consumption is therefore not noticeably decreased by the use of a low loss rubber. On a decline belt conveyor the application of a low loss rubber may be a bad idea since it may increase the size and complexity of the brake system.

2 RECENT SOUTH AFRICAN PROJECTS

In the last three years, three South African projects involving long overland belt conveyors have been realised:

1. CRU-II, Middelburg for Ingwe,
2. Optimum, Hendrina for Ingwe, and
3. Savmore, Piet Retief for the Kanga Group.

During all three projects the quality, in particular of the rubber covers, and the supplier of the conveyor belting were serious issues for discussion. The next three paragraphs explain the specific matters.

2.1 CRU-II

After adjudicating tenders from several top-ranking world contenders, Middelburg Mine Services awarded the contract for a 14,5 km overland conveyor system to BATEMAN. The project was executed by Bateman Engineered Technologies. The conveyor system is part of phase II of the R480M Coal Resources Utilisation Project (CRU II) initiated by Ingwe Coal Corporation Limited and was commissioned in May 2000.

In the tendering stage of the project, the belt conveyor system was presented, and later sold, as a high tech system utilising low indentation loss compound for the belt's covers. The biggest advantage of using a low loss rubber for the belt was a serious decrease in expected power consumption of the total system. The designs of the individual belt conveyors then were based on using belts with low loss covers. The anticipated supplier for the CRU-II belting was Bridgestone. During the course of the project however, the client requested that they could use alternative (read non low-loss rubber) belts as a replacement belt. The main reason for this was that Ingwe wanted to have a better position to negotiate for replacement belting. The design was therefore slightly altered, in particular the settings of major components as the drives and the brake systems, to enable the application of alternative belting. After completion of the system Optimum used a Dunlop SA belt to replace part of the original Bridgestone belting without any serious problem.

2.2 Optimum

Ingwe has awarded BATEMAN a turnkey contract for a 21 km overland-conveyor system to be supplied to Optimum Colliery. It includes all design, supply and erection, inclusive of civil works. The system will comprise five belt conveyors ranging in length from 2.7 km up to 6.1 km.

Figure 1: *Belt Conveyor KW-05 of the Optimum overland system.*

Figure 2: *The Savmore overland belt conveyor.*

The Optimum project, see Figure 1, knew a short-track design phase. The design of the system was in principle a further development of the system designed for CRU-II, including the application of conveyor belting with low loss rubber compounds. However, because of the short-track development there was not enough steel cord belting available on the world-market at the time. As a result, the client had to buy a mixture of Dunlop and Bridgestone belting. One of the main design principles of the Optimum overland belt conveyor system was standardisation of components. Therefore, all belt conveyors in principle should allow for the use of either Dunlop or Bridgestone belting. Because the problem with shortage of belt supply was know at a relatively early stage of the project, most components of the belt conveyors could still standardised but were tuned for the specific belting used on individual conveyors.

2.3 Savmore

Kangra Group (Pty) Ltd's Savmore Colliery, near Piet Retief in the Mpumulanga Province of South Africa, has awarded BATEMAN a contract for a 6,5 km overland conveyor. The conveyor will link Savmore's new Maquassa West shaft with the existing plant at Maquassa East and will carry 1 000 t/h of run-of-mine coal.

The Savmore project was developed at the same time as Optimum and therefore the Kanga Group had the same belt shortage problem as Ingwe. However, there was only one long overland belt conveyor in the Savmore project and Savmore decided to buy the belt directly from Goodyear and provided it to Bateman as a free issue. Although the design of the Savmore belt conveyor was based on the assumption that it should be able to utilise basically any modern conveyor belt, the dynamic/mechanic properties of the specific conveyor belt were still required to optimise the system by tuning the components. The dynamic/mechanic properties of the Goodyear belt however were not known and Goodyear was not able or unwilling to supply either rubber mechanic/dynamic properties or a sample of the specific rubber used. As a result the performance of that specific conveyor belt, and thus the conveyor system, were unknown during the commissioning stage.

3 VISCOELASTICITY

In this section a model will be presented that can be used to represent the viscoelastic behaviour of the material of a conveyor belt's cover.

Most belt covers are made of rubber or polyester material. The constitutive behaviour of these materials is viscoelastic as can be learned from the time-dependency of the stress-strain relations, [2]. The most important environmental parameters that affect the dynamic response of viscoelastic materials are temperature, frequency and the amplitude of an imposed load [4]. It is also important to know the exact compound of the material. In rubber for example the amount of carbon black influences the material properties considerably [5].

The constitutive equation for an isotropic linear viscoelastic material can be written in general tensor form [6]:

$$\sigma^d(t) = \int_{-\infty}^{t} \Psi\left(t - t'\right) \frac{\partial \gamma^d\left(t'\right)}{\partial t'} dt' \qquad (1)$$

in which $\sigma^d = \sigma - \left(\frac{1}{3} tr\sigma\right)$ is the deviatoric stress tensor and $\gamma^d = \gamma - \left(\frac{1}{3} tr\gamma\right)$ the deviatoric strain tensor. The fourth order tensor function $\Psi(t)$ is called the relaxation function and specifies the stress response to a unit strain increment. It can be written as:

$$\Psi(t) = \Psi_\infty + \int_0^\infty g(t) \exp\left(-\frac{t}{\tau}\right) d\tau \qquad (2)$$

where $g(\tau)$ is the relaxation spectrum which can be discrete or continuous and τ the relaxation time. If in the uni-axial case a pulse-spectrum $g(\tau) = \sum_{i=1}^{N} g_i \delta(\tau - \tau_i)$ is used then the relaxation function is equal to:

$$\Psi(t) = \Psi_\infty + \sum_{i=1}^{N} g_i \exp\left(-\frac{t}{\tau_i}\right) \qquad (3)$$

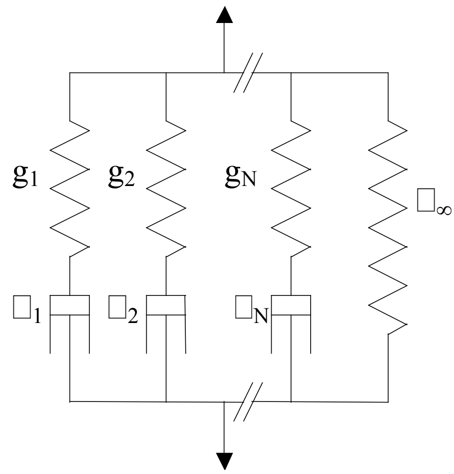

Figure 3: *Generalised uniaxial Maxwell model.*

This material model is known as the generalised Maxwell model. Figure 3 shows this uniaxial case. In this model a number of damping coefficients η_i is used which are related to specific relaxation times τ_i, in order to be able to represent the constitutive behaviour of a material for a wide range of loading frequencies. If this range is relatively small for a specific application then it is sufficient to use one relaxation time which fits for that range. In such a case a three parameter model, or a so called standard linear solid model, results that is the simplest model that can describe the relaxation of a material and situations of constant stress or high strain rates, see Figure 4.