

https://dspace.org/

The dynamics of mine hoist catenaries

Constancon, Charles Peter

2011-04-20
http://hdl.handle.net/10539/9610

Downloaded from a DSpace repository. https://dspace.org/

THE DYNAMICS OF MINE HOIST CATENARIES

Charles Peter Constancon

A Thesis Submitted to the Faculty of Engineering,
University of the Witwatersrand, Johannesburg, South Africa
for the Degree of Doctor of Philosophy.

Johannesburg 1993

Abstract

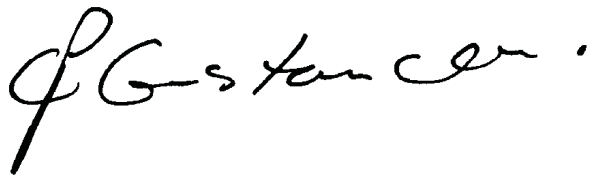
The dynamic analysis of catenary vibration of mine hoist ropes on South African mines is examined. This research has been preceded by studies in the mining industry, which have laid the foundation for the definition of design guidelines of hoist systems to avoid catenary vibrations or *rope whip*. These guidelines are based on a classical linear analysis of a taut string, and in essence rely on ensuring that the frequency of excitation at the winder drum due to the coiling mechanism, does not coincide with the linear transverse natural frequency of the taut catenary. Such an approach neglects the nonlinear coupling between the lateral catenary motion and the longitudinal system response. Although previous research suggested the possibility of autoparametric coupling between the catenary and vertical rope, this was not developed further on a theoretical level. The possibility of such behaviour is defined by considering the equations of motion of the coupled system.

A design methodology is developed for determining the parameters of a mine hoist system so as to avoid rope whip. The methodology accounts for the nonlinear coupling between the catenary and longitudinal system. In order to implement the proposed methodology, two phases of the analysis are developed. In the first phase the stability of the linear steady state motion is examined in the context of the nonlinear equations of motion, by applying a harmonic balance method. The stability analysis defines regions of secondary resonance, where it is shown that such regions may arise at sum and difference combinations of the linear lateral and longitudinal natural frequencies due to autoparametric excitation. Prior to this research, this phenomenon had not been appreciated in the context of the mine hoist system. A laboratory experiment was conducted to confirm the existence of these regions experimentally. In reality, the system is non-stationary since the dynamic characteristics of the system change during the winding cycle, and hence the steady state stability analysis can only describe potential regions of nonlinear interaction on a qualitative basis. The second phase of the analysis deals with a non-linear numerical simulation of the hoist system, which accounts for the non-stationary nature of the systems dynamic characteristics, and includes transient excitations induced during the wind.

The methodology developed is assessed by considering the Kloof mine rope system, where rope whip was observed. This study demonstrates that although an appreciation of the steady state system characteristics is useful, the stability analysis alone is not sufficient. It is necessary to account for the non-stationary aspects of the winding cycle if a realistic interpretation of the observed behaviour is to be achieved. To compliment this study, a motion analysis system was developed to record catenary response on an existing mine hoist installation. Such data has not been recorded before. This data provides direct evidence of the autoparametric nature of the coupled catenary/vertical rope system.

DECLARATION BY THE CANDIDATE

I, Charles Peter Constancon, declare that this thesis is my own work, that its substance neither in whole nor part has been submitted to, or is to be submitted to, any institution other than the University of the Witwatersrand, Johannesburg.



A man breaking his journey between one place and another at a third place of no name, character, population or significance, sees a unicorn cross his path and disappear. That in itself is startling, but there are precedents for mystical encounters of various kinds, or to be less extreme, a choice of persuasions to put it down to fancy; until - "My God," says a second man, "I must be dreaming, I thought I saw a unicorn." At which point a dimension is added that makes the experience as alarming as it will ever be. A third witness, you understand, adds no further dimension but only spreads it thinner, and a fourth thinner still, and the more witnesses there are the thinner it gets and the more reasonable it becomes until it is as thin as reality, the name we give to the common experience.... "Look, look!" recites the crowd. "A horse with an arrow in its forehead! It must have been mistaken for a deer."

Tom Stoppard.

ACKNOWLEDGEMENTS

Professor B W Skews, who has provided support and advice concerning the development and execution of this thesis.

Dr M E Greenway, who provided a forum for discussion concerning the direction in the thesis, as well as access to data captured by his department on Elandsrand Gold Mine.

The Chamber of Mines for the grant which provided funds for the purchase of the motion analysis equipment.

Professor C Dimitriou, who has acted as a mentor throughout my undergraduate and graduate career.

Professor E Dockumaci, who during his visit to the University in October 1990, both stimulated and provided theoretical support for the study.

D Massey, a colleague, who has consistently availed himself to lengthy discussions regarding the thesis and given unfailing encouragement and moral support throughout the thesis.

R A Backeberg, a former colleague, who has consistently availed himself to lengthy discussions regarding the thesis and given unfailing encouragement and moral support throughout the thesis.

M J Howell for the unenviable task of proof reading the script.

To family and friends for their patience and support.

Outline of the Thesis

The topic of catenary vibration or rope whip in hoisting systems on South African mines was examined in the early 70's by Dimitriou and Whillier[1973], and Mankowski, Whillier and Louw[1974]. This research resulted in the development of design guidelines for the avoidance of rope whip, based on classical taut string theory. The possibility of autoparametric response of the system was intuitively described by Dimitriou and Whillier[1973], but not formalised. Mankowski[1982] researched the forced nonlinear response of the system numerically, culminating in a Ph.D. thesis submitted to this University. His thesis examined, and alluded to, the difficulty associated with modelling nonlinear aspects of the Kloof mine hoist system, and consequently of nonlinear hyperbolic partial differential equations. Numerical results defined the forced response of the modelled system. Unfortunately, the simulation exhibited numerical instability, and a complete simulation of the winding cycle of Kloof Mine was not achieved. Nevertheless, Mankowski's contribution provided invaluable information concerning the behaviour of the mine hoist system, as well as the application of dynamic simulation techniques employing Bergeron's[1961] impedance method. However, this effort did not clarify or extend the existing design guidelines.

The current thesis intends to compliment previous research, by developing a methodology pertaining to the avoidance of catenary whip on mine hoist systems. In this regard, during the initial design phase, the nonlinear forced resonance of the system is not of primary concern. It is the definition of system parameters likely to reduce the potential for rope whip, and hence the avoidance of significant nonlinear interactions, which is of primary importance. Hence the nonlinear equations of motion of the catenary are linearised about the steady state first order solution of the system, and the stability of this motion to small disturbances is examined. The philosophy of this approach is to capture the fundamental nature of mechanisms relating to the growth of the catenary response, in such a manner that a routine approach may be employed during the design phase of a shaft head layout to avoid such regions. This approach may appear to discard the reservations apparent in recent literature regarding the study of nonlinear systems, specifically with respect to the discovery of chaotic behaviour and consequently the importance of retaining the integrity of the nonlinear nature of the system. This is particularly important in terms of the final validation of the system parameters selected, since transient excitation, as well as the non-stationary nature of the system may lead to unexpected behaviour. To address this, a numerical simulation retaining nonlinear terms as well as transient excitation was developed. The simulation is computationally intensive, and is intended to be used as the final validation of the design parameters.

In Chapter 1 of this thesis, a description and definition of the scope of the work and its context in the South African mining industry is presented. The components of the mine hoist system are examined and sources of excitation are identified. Relevant research pertaining to studies of the dynamics of the general system are reviewed. A detailed account of past research concerning catenary dynamics of mine hoist systems, and the current guideline employed for the design of a system in order to avoid catenary whip is presented.

In Chapter 2 a discussion relating to research conducted in the literature concerning the dynamics of strings and cables is presented. The purpose of this chapter is to establish a perspective on the types of nonlinear behaviour associated with taut strings and cables. These studies consider strings and cables fixed at each end. Such boundary conditions are relevant to most cable systems, where in practice the longitudinal wave speed is significantly larger than the lateral wave speed; this permits simplification of the equations of motion by neglecting the longitudinal inertia of the system, thereby introducing a quasi-static definition for the longitudinal response. This represents the point of departure between studies presented in the literature, and the mine hoist system, where due to the coupling between the catenary and vertical cable across the sheave, retention of the longitudinal system inertia is essential.

In Chapter 3 the derivation of the equations of motion of the coupled catenary and longitudinal system is presented. The equations are derived by applying Hamilton's principle. In the most general form, the equations of motion include rope curvature, axial transport velocity of the rope, and utilise a nonlinear strain definition relating the axial tension in the cable to motion in three orthogonal directions. These equations form the basis for further development.

In Chapter 4 the theoretical basis for the first phase of the proposed methodology is presented. The equations of motion developed in chapter 3 are simplified. A *datum* solution is then formulated. This solution is only valid in the absence of primary or secondary resonance conditions. The variational form of the solution is developed by substitution of this solution into the nonlinear equations of motion followed by linearisation, and normal mode orthogonalisation. This results in a set of coupled Hill type ordinary differential equations with periodic coefficients. The stability of the system is firstly examined by applying a harmonic balance method. The harmonic balance method was applied, since it is possible to prepare a general numerical code for assessing the stability of the system. Although the definition of regions of stability may be achieved in closed form, by applying a perturbation method such as multiple scales, it is likely that a number of regions may occur simultaneously, resulting in anomalous regions requiring special attention. In addition, perturbation methods require the concept of a small parameter, as well as a higher level

of skill from the designer. However, a multiple scales method is applied to substantiate the regions determined, as well as to define conditions of tuning which would lead to internal resonance. The results from a simple laboratory experiment are presented to further validate the numerical programme developed, as well as to illustrate the significance of the secondary resonance regions at an additive combination of the longitudinal and lateral natural frequency. Finally the stability analysis of the Kloof Mine rope system is presented and discussed.

Chapter 5 presents the numerical simulation developed as the second phase of the methodology. The simulation accounts for the non-stationary nature of the system, and transient excitations. These effects could not be accounted for in the steady state stability analysis. An important aspect of a numerical simulation concerns the damping mechanism applied. Experimental results relating to a definition of a damping mechanism applicable to mine hoist cables are scarce. The damping mechanism applied in the simulation is discussed and motivated. A numerical simulation of the Kloof Mine rope system is presented for both the ascending and descending wind. The results confirm the observations of Dimitriou and Whillier[1973], where severe rope whip occurred on the ascending cycle but not on the descending cycle.

Chapter 6 concludes the study, where a critical appraisal of the work is presented, and suggestions for further research are discussed.

Contents

1 Introduction **1**

1.1 The Mine Hoist System 4

1.1.1 Excitation Sources 6

1.2 The Study of Dynamics in the Industry 9

1.2.1 Longitudinal Dynamic Studies 9

1.2.2 Coupled Dynamic Studies 11

1.3 Scope for Analytical Research 21

1.4 Scope of the Study 23

2 The Dynamics of Strings and Cables **24**

2.1 Introduction 24

2.2 Static Response of Cables 25

2.3 Dynamics of Strings and Cables 29

2.4 Taut String Analyses 31

2.4.1 Nonlinear Taut String Analyses 31

2.4.2 Travelling Taut Strings 34

2.5 Small Amplitude Vibration of Sagged Cables 37

2.5.1 Stationary Sagged Cables 37

2.5.2 Travelling Sagged Cables 41

2.6 Large Amplitude Vibration of Sagged Cables 44

2.7 Conclusion 48

3 Equations of Motion **50**

3.1 Introduction 50

3.2 The Sagged Travelling Catenary 51

3.2.1 Equations of Motion 51

3.2.2 Hamilton's Principle 54

3.3 The Mine Hoist System 58

3.3.1 Equations of Motion and Equilibrium 60

3.3.2 Flat Sag Cable Approximation 66

3.4 Conclusion 70

4 Stationary Analysis of the Mine Hoist System **72**

4.1 The Steady State Analysis 74

4.2 The Linear *Datum* Solution 76

4.3 Stability of the *Datum* Solution 79

4.3.1 Stability of the Variational Equations 81

4.3.2 Experimental Validation 82

4.4 The Kloof Mine Hoist System 89

4.5 Conclusion 94

5 Nonlinear Numerical Simulation **95**

5.1 Nonlinear Equations of Motion 96

5.1.1 Simplifications Applied in the Modelling Process 96

5.1.2 Reduced Equations of Motion 97

5.2 Normal Mode Model of the Equations of Motion 98

5.3 Laboratory Measurements 107

5.4 Quasi-Static Laboratory Model 109

5.4.1 Experimental Correlation 114

5.5 Quasi-Static Mine Hoist Model 119

5.6 Longitudinal and Lateral Damping Mechanisms 127

5.6.1 Longitudinal Damping 127

5.6.2 Lateral Damping 128

5.7 Simulation of the Equations of Motion 129

5.8 Kloof Mine Simulation 131

5.9 Simulation Results 134

5.9.1 Descending Cycle 134

5.9.2 Ascending Cycle 142

5.9.3 Dimitriou and Whillier's Observations 151

5.10 Conclusion 153

6 Closure **156**

Appendices **160**

A Excitation Definition **160**

A.1 Lebus Coil Cross-Over Excitation 162

A.2 Acceleration/Deceleration Excitation 171

A.3 Layer Change Excitation 172

A.4 Excitation Definition for the Quasi-Static Model 177

B Parametric Response due to Longitudinal Excitation Only **179**

B.1 Stability Analysis 182

B.2 Parametric Resonance of MDOF Systems 183

B.2.1 Hsu's Perturbation Solution 184

B.3 Anomalous Cases 188

B.3.1 Case 1: 2 DOF - $s = 1,2$ $\omega \approx 2\omega_1$ 190

B.3.2 Case 2: 3 DOF - $s = 1$ $\omega \approx 2\omega_2$ 194

B.3.3 Case 3: 2 DOF - $s = 1,2,3$ $\omega \approx \omega_1$ 197

B.4 Kloof Mine Observations 202

C Linear Longitudinal System Response **209**

C.1 Undamped Natural Frequencies 212

C.2 Forced Longitudinal Damped Response 215

C.2.1 Determination of the Modal Damping Coefficient 218

C.3 General Proportional Damping 219

C.4 Longitudinal Response - Kloof Mine 221

C.4.1 Relative Proportional Damping Model 220

C.4.2 General Proportional Damping Model 221

C.4.3 Forced Longitudinal Response 222

D Linear Lateral Catenary Response **226**

D.1 Forced Response 227

E Linear Coupled System Response **229**

E.1 Longitudinal Drift Term 236

F Parametric coupling Matrices **238**

F.1 Longitudinal Parametric Coupling 240

F.2 Lateral Parametric Coupling 243

F.3 Drift Terms 248

G Longitudinal Damping Estimates **251**

G.1 Damping Mechanisms 251

G.2 Drop Tests 253

G.3 Discussion 255

G.4 Conclusion 258

H Lateral Damping of the Catenary **265**

H.1 Aerodynamic Damping 267

H.1.1 Aerodynamic Drag 267

H.1.2 Planar Motion 268

H.1.3 Non-Planar Motion 269

H.2 Equivalent Aerodynamic Power Losses 271

H.3 Damping Model Applied in the Simulation 272

I Video Measurement System **273**

I.1 Video System 273

I.2 Rope Tracking 274

I.3 Hardware Configuration 274

I.4 Dual Camera Measurement 275

I.5 Commissioning Tests 276

I.5.1 Measurements Performed 276

I.5.2 Measurement Results 276

I.6 Discussion 278

I.6.1 Catenary Response 278

I.6.2 Lateral Response of the Vertical Rope 279

I.7 Summary 281

J Stationary Stability Analysis - Further Considerations **290**

J.1 Nonlinear Equations of Motion 291

J.2 Nonlinear Methods of Analysis 293

J.3 The Analysis of Nonlinear Systems with Quadratic and Cubic Nonlinearities. 297

J.4 Secondary Resonance Conditions 312

J.5 Conclusion 317

K Stability of Liner ODE's with Periodic Coefficients **318**

K.1 Parametrically Exited Systems 318

L Numerical Sensitivity Study **326**

M Equations of Motion Including Sheave Wheel Constraints **350**

References **356**

List of Figures

1.1 Hoisting system configurations 5

1.2 Mankowski[1982], Figure 2.4(a): Winder drum fitted with a Lebus liner 7

1.3 Mankowski[1982], Figure 2.4(b): Rope coiling pattern 8

1.4 Dimitriou & Whillier[1973], Figure 8: Pulse initiation by coil cross-overs at the drum 8

1.5 Dimitriou & Whillier[1973]: Linear frequency map 12

1.6 Mankowski[1982], Figure 11.28: Longitudinal frequency content of skip velocity - simulation model and theoretical modes of double mass model. 16

1.7 Mankowski[1982]:Frequency content of the simulation and the double mass model. 18

1.8 Backeberg[1984], Figure 4.1: Mine rope model 20

2.1 Irvine (1981): Equilibrium of an element of cable 25

2.2 Mote (1966): Nonlinear fundamental period of an axially translating string 35

2.3 Irvine (1981): In-plane natural frequencies of a cable. 39

2.4 Perkins and Mote (1987): In-plane natural frequencies of a cable - Comparison with Irvine's theory. 43

2.5 Perkins and Mote (1987): In-plane natural frequencies of a translating cable - Comparison with Simpson's theory. 43

3.1 Catenary configuration 51

3.2 Displacement of a differential element of cable from χ^i to χ^f . . 53

3.3 Mine hoist configuration 58

3.4 Catenary - sheave interface 60

3.5 Control volume of segment of cable. 65

3.6 Flat sag inclined cable 67

3.7 Mine hoist model 68

4.1 Laboratory model of the mine hoist system 83

4.2 Stability chart of the laboratory model 87

4.3 Stable/unstable catenary motion - Case 3. 88

4.4 Kloof Mine: Linear dynamic characteristics. 91

4.5 Stability chart of the steady state datum solution - Ascending . 92

4.6 Stability chart of the steady state datum solution - Ascending . 93

4.7 Stability chart of the steady state datum solution - Descending . 93

5.1 Kloof Mine simulation - Normal mode model 103

5.2 Experimental steady state amplitudes 108

5.3 Quasi-static laboratory model 109

5.4 Experimental measurements vs simulation results 115

5.5 Sine sweep simulation - positive sweep rate. 117

5.6 Sine sweep simulation - negative sweep rate. 118

5.7 Quasi-static mine hoist model 119

5.8 Stationary lateral natural frequencies vs shaft depth - Kloof Mine 124

5.9 Descending cycle: Kloof Mine hoist system: 14.8 m/s 136

5.10 Descending cycle: Kloof Mine hoist system: 14.8 m/s 137

5.11 Descending cycle: Out-of-plane modal amplitudes: 14.8 m/s . . 138

5.12 Descending cycle: In-plane modal amplitudes: 14.8 m/s 139

5.13 Descending cycle: Kloof Mine hoist system: 14.6 m/s 140

5.14 Descending cycle: Kloof Mine hoist system: 15 m/s 141

5.15 Ascending cycle: Kloof Mine hoist system: 14.8 m/s 144

5.16 Ascending cycle: Kloof Mine hoist system: 14.8 m/s 145

5.17 Ascending cycle: Out-of-plane modal amplitudes: 14.8 m/s . . . 146

5.18 Ascending cycle: In-plane modal amplitudes: 14.8 m/s 147

5.19 Ascending cycle: First mode polar plot 148

5.20 Ascending cycle: Second mode polar plot 148

5.21 Ascending cycle: Third mode polar plot 149

5.22 Ascending cycle: Fourth mode polar plot 149

5.23 Ascending cycle: Polar plot of the response, $s = t_c/4$ 150

A.1 Mankowski[1982], Figure 2.4(a): Winder drum fitted with a Lebus liner 161

A.2 Geometry of rope layers 165

A.3 Cross-over geometry 165

A.4 Physical rope motion per drum revolution 166

A.5 Periodic motion per drum revolution 167

A.6 Displacement spectra 168

A.7 Simulated Fourier vs. physical excitation - 2 Harmonics 169

A.8 Filler geometry 171

A.9 Layer change transient excitation 175

A.10 Lebus coil-cross over excitation - Versine functions 177

B.1 Dimitriou and Whillier (1973): Kloof Mine shaft dynamic characteristics 201

B.2 Ascending Kloof Mine shaft dynamic characteristics 202

B.3 Main parametric resonance regions - Case I 206

B.4 Parametric resonance regions - Harmonic balance method 207

C.1 The longitudinal system 209

C.2 Modal damping factors - Relative proportional damping 223

C.3 Modal damping factors - General proportional damping 223

C.4 Linear steady state forced response 224

G.1 Drop Test : Modal damping ratio ζ_i verses conveyance load . . 260

G.2 Drop test free response: AAC accelerometer record 261

G.3 Frequency response functions: AAC accelerometer record 262

G.4 Curve fitted data: Based on first modal damping estimates. . . 263

G.5 Damping data and curve fit for the first longitudinal mode. . . 263

G.6 Greenway (1993): Global longitudinal damping estimate 264

G.7 Global longitudinal damping estimate based on *linear* ζ_1 264

I.1 Video camera configuration 282

I.2 Mixed video image of the rope 282

I.3 Test 1: Time trace 283

I.4 Test 2: Time trace 284

I.5 Cascaded spectral contour plot 285

I.6 Cascaded spectral contour plot 286

I.7 Test 3: Time trace 287

I.8 Test 3: Transformed time trace 288

I.9 Expanded time trace: Test 3 289

J.1 Nayfeh & Mook (1983): Figure 6.8 p 386 299

J.2 Nayfeh & Mook (1983): Figure 6.13 p 406 301

J.3 Nayfeh & Mook (1983): Figure 6.14 p 407 301

J.4 Nayfeh & Mook (1983): Figure 6.17 p 410 302

J.5 Nayfeh & Mook (1983): Figure 6.18 p 411 303

J.6 Nayfeh & Mook (1983): Figure 6.20 p 416 305

K.1 Takahashi (1982): Figure 1 - Regions of simple parametric resonance 320

K.2 Takahashi (1982): Figure 2 - Regions of parametric instability for coupled Mathieu equations. 324

L.1 Ascending cycle - Kloof Mine hoist system: 14 m/s 328

L.2 Descending cycle - Kloof Mine hoist system: 14 m/s 329

L.3 Ascending cycle - Kloof Mine hoist system: 14.2 m/s 330

L.4 Descending cycle - Kloof Mine hoist system: 14.2 m/s 331

L.5 Ascending cycle - Kloof Mine hoist system: 14.4 m/s 332

L.6 Descending cycle - Kloof Mine hoist system: 14.4 m/s 333

L.7 Ascending cycle - Kloof Mine hoist system: 14.6 m/s 334

L.8 Descending cycle - Kloof Mine hoist system: 14.6 m/s 335

L.9 Ascending cycle - Kloof Mine hoist system: 14.8 m/s 336

L.10 Descending cycle - Kloof Mine hoist system: 14.8 m/s 337

L.11 Ascending cycle - Kloof Mine hoist system: 15 m/s 338

L.12 Descending cycle - Kloof Mine hoist system: 15 m/s 339

L.13 Ascending cycle - Kloof Mine hoist system: 15.2 m/s 340

L.14 Descending cycle - Kloof Mine hoist system: 15.2 m/s 341

L.15 Ascending cycle - Kloof Mine hoist system: 15.4 m/s 342

L.16 Descending cycle - Kloof Mine hoist system: 15.4 m/s 343

L.17 Ascending cycle - Kloof Mine hoist system: 15.6 m/s 344

L.18 Descending cycle - Kloof Mine hoist system: 15.6 m/s 345

L.19 Ascending cycle - Kloof Mine hoist system: 15.8 m/s 346

L.20 Descending cycle - Kloof Mine hoist system: 15.8 m/s 347

L.21 Ascending cycle - Kloof Mine hoist system: 16 m/s 348

L.22 Descending cycle - Kloof Mine hoist system: 16 m/s 349

M.1 System configuration with geometric constraints 354

M.2 Geometric constraint relationships 355

List of Tables

4.1 Laboratory model parameters 83

4.2 Tuning conditions of the laboratory model 84

4.3 Kloof Mine - System parameters 92

5.1 Simulation variables 130

5.2 Simulation variables - Kloof Mine 131

B.1 Anomalous regions of parametric resonance 188

C.1 Kloof Mine - System parameters 225

G.1 Deelkraal Data - Greenway[1989] 259

G.2 Elandsrand - Drop test measurements - AAC accelerometer . . 259

G.3 Elandsrand - Drop test measurements - AAC LVDT 259

G.4 Elandsrand - Drop test measurements - WITS load cell 260

H.1 Damping coefficients - Mankowski[1988] -Kloof Mine 271

H.2 Damping coefficients - Aerodynamic -Kloof Mine 271

Nomenclature

c	Longitudinal wave speed.
$\overline{c}$	Lateral wave speed.
E	Youngs Modulus.
J	Mass moment of inertia of the sheave wheel.
l_c	Catenary length.
l_v	Vertical rope length.
M	Conveyance mass.
m	Linear rope mass.
p_i	i^{th} longitudinal normal mode.
q_i	i^{th} lateral in-plane normal mode.
r_i	i^{th} lateral out-of-plane normal mode.
s	Arc length co-ordinate.
u	Longitudinal displacement in the catenary.
v	Lateral in-plane displacement.
w	Lateral out-of-plane displacement.
$\overline{u}$	Longitudinal displacement of the vertical rope.
u_1	Longitudinal displacement at the sheave.
u_2	Longitudinal skip displacement.
$\phi_i(s)$	i^{th} longitudinal mode shape of the longitudinal system.
$\Phi_i(s)$	i^{th} lateral mode shape of the catenary.
ω_i	i^{th} Longitudinal natural frequency.
$\overline{\omega}_i$	i^{th} Lateral natural frequency.

Chapter 1

Introduction

The development of the mining industry in South Africa has stimulated both technical and social change. The primary function of the industry is the extraction of mineral rich ore from natural reserves. During the early stages of development, this necessitated ore extraction from depths measurable in hundreds of meters. Currently the local gold mining industry is internationally distinguished for developing mines to depths of thousands of meters. The need for such depth has dictated the development of high speed hoisting equipment to accommodate production requirements. Due to the capital cost and stringent safety criteria defined by legislation, an active area of research concerning the technical aspects of hoisting has emerged. Research has been pursued under the auspices of The Chamber of Mines Research Organisation (COMRO), which supports research in the *Shafts* programme. Backeberg[1990] states that this programme was established to solve problems associated with access to the reef, and incorporates two main areas of investigation within conventional hoisting, namely, winding ropes and shaft steelwork/conveyances.

The *Shafts* programme can be broadly divided into four main projects which are pertinent to this study.

- Winder Ropes - Factors of Safety
- Rope Degradation Mechanisms
- Shaft Steelwork Design
- Conveyances and Guide Rollers

Each of these projects are pertinent to current and future production, safety and maintenance costs associated with the hoisting process. The projects are interrelated and generally pursue one main theme: the extension of the safe operating envelope of single hoist winding systems. Naturally, benefits will be derived by systems operating at shallower depths. The rope factors project examines the extent to which the legal factor of safety can be lowered, thus easing the current depth constraint. Consequently, rope selection strategy, and the aspect of compensating the reduction in overall rope factor of safety by reducing dynamic loads through more stringent winder control, have been considered.

The *Rope Degradation Mechanism* programme examines wear and damage mechanisms related to the winding process and hence is not only concerned with the reduction in rope strength, but also with extending the operating life of the rope. With regard to this programme, Chaplin[1989] has identified several deterioration mechanisms, namely plastic and frictional wear due to high contact stresses generated during coiling, followed by axial slipback during the unwinding process. Fatigue loading, due to the combined effect of fluctuating tension and bending over the sheave or underlying coil layer on the winder drum surface, is also identified as a damage mechanism. An experimental programme has been suggested to study such mechanisms and the feasibility of commissioning an experimental facility is currently being considered by the Council for Scientific and Industrial Research (CSIR).

The production constraint of deep level mining will dictate higher winding speeds. Current winding speeds of 18 m/s will be extended to 25-30 m/s. Harvey[1973] attributed the cautious approach to winding speed to *"present limitations in shaft steelwork and conveyance design"* - which *"lead to problems in conveyance oscillations"*. The *Shaft Steelwork Design* programme essentially addressed this problem. The Structural Dynamics Research Corporation (SDRC) was commissioned by COMRO[1990] to investigate the mechanism of slamming behaviour of the conveyance between the guides. Slamming occurs

when the conveyance is excited between the guides to the extent that the guide rollers become inoperative, resulting in steel to steel contact between the skip and guide. Through numerical simulation, SDRC identified significant parameters leading to slamming and hence derived design guidelines. The results of the study demonstrated that slamming is influenced by guide alignment, conveyance speed, guide length and the ratio of mid span guide stiffness to support stiffness. Although this work was carried out independently by SDRC, Tondl (see Schmidt and Tondl[1986]) investigated a similar problem for the German mining industry. Tondl showed that by incorporating rotational and translational degrees of freedom , as well as the effect of periodic guide stiffness, the coupled conveyance/guide system linearised to one with periodic coefficients. He thus concluded that the system was parametrically excited and derived approximate solutions to the stability intervals. These intervals are related to the same parameters subsequently identified by SDRC. The aspect of guide alignment, which was identified as a contributory factor to slamming, has received attention, and two devices capable of measuring alignment have been designed and developed. Since it will not be possible to guarantee perfect guide alignment, the *Conveyances and Guide Rollers* Project was initiated. This project compliments the former in that it focusses on the design of conveyances and lateral conveyance suspension systems, to accommodate and minimise dynamic loads induced by the alignment of the shaft steelwork.

It is evident that the mining industry is involved in significant research. Hoisting technology is a complex field, continually under assessment and revision. The industry employs current technology, as well as investing in international research and development. The study presented in the current thesis covers a small aspect of the global effort. It was stimulated through assessing past research carried out at the University, the aim being to refine and re-initiate research into phenomena associated with catenary dynamics, in preparation for current and future application.

1.1 The Mine Hoist System

The mine hoist system consists of a headgear, vertical shaft and winding house. The headgear is necessary to elevate the conveyance above ground level to effect the dumping of the conveyance load. Thus the design necessitates the existence of an inclined catenary passing over a sheave in the headgear to the conveyance in the vertical shaft. During the design stage, parameters such as rope diameter, sheave diameter, winder drum diameter and power are determined. It is at this critical stage that the dynamic behaviour of the overall system is addressed. Every attempt should be made to ensure that the design speed of the system and the production rate of the shaft is attainable in practice, as little can be done subsequently to correct a system for adverse dynamic behaviour.

In essence three different configurations of the hoist system exist, and are illustrated in figure 1.1.

a) A single drum hoisting system.

b) A Koepe winding system.

c) A Blair multi rope hoisting system.

The single drum winder configuration is the simplest design. It consists of a single drum, single cable and conveyance. This system is generally limited to shallower winds, or lower payloads.

The Koepe system is based on a bollard type friction principle. Each conveyance is attached between a head and tail rope. The system is counter balanced, where a differential torque is supplied by the winder to accommodate the difference in mass between the full ascending conveyance and the empty descending conveyance. No storage facility for the rope on the winder drum is required. Generally the Koepe winding system is used for shallower winds ($< 1000m$). The Koepe winding configuration is not addressed in this study. However tail whip is a consideration on these systems, and is viewed as a potential area for future research.

The Blair multi-rope winder is an extension of the single drum winder configuration, in that two ropes support the conveyance, driven by two single drum winders. The rope between the two winders is continuous, the winders are mechanically coupled, and a compensating sheave on the conveyance facilitates

equal load sharing between the two ropes. If it is assumed that the compensating sheave perfectly isolates the cables from one another, then this system could be approximated as two single drum winders.

This study addresses single drum and Blair multi-rope double drum winders. Although these basic systems differ in physical appearance, for the purpose of dynamic studies they may be reduced to, and treated as a single drum winding system. Thus the particular layout of the mine hoist system studied is in fact that of a single drum winder, where design parameters determined for this system would be readily applicable to either configuration.

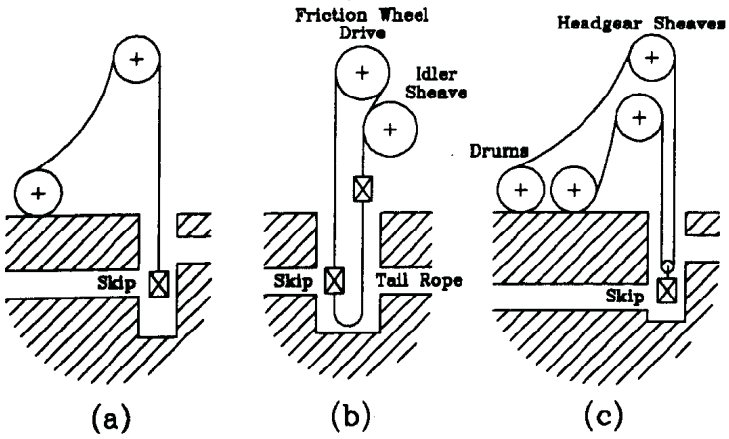


Figure 1.1: Hoisting system configurations

a) Single Drum Winder. b) Koepe Winder. c) Blair Multi-rope Winder.

1.1.1 Excitation Sources

Winder technology developed with the mining industry's need to mine to greater depths. As shaft depth increased, the cost of wire ropes increased due to a higher strength requirement. In order to maximise the life of the rope, mechanisms causing rope damage were minimised. During winding, rope damage may occur at the winder drum due to miscoiling, gaps or pull-through, which may result in crushing or even failure of the rope due to the high contact pressures developed. In order to minimise this damage, coiling mechanisms were designed to ensure a uniform controlled coiling pattern on the winder surface and on subsequent layers.

In the late $1950's$, an invention by Frank L Lebus (Dimitriou and Whillier [1973]) resolved these problems, and resulted in a system where controlled coiling at high speed could be achieved. This system achieved a neat multi-layer coiling pattern enabling the storage of cable on the drum without the damaging effects of miscoiling. This consisted of grooving the drum as illustrated in figure 1.2. The cross-over points on the Lebus drum are diametrically opposed, and as the cable passes through a cross-over, it is shifted one half of its diameter on one side of the drum, and one half on the other. When the cable reaches a drum flange, a filler guides the layer change, and spooling continues in the opposite direction. A cross section of the typical coiling pattern is illustrated in figure 1.3.

Previous research by Dimitriou and Whillier [1973][1] identified various excitation sources, the major excitation being that due to the coiling mechanism employed on the winder drum surface in order to achieve a uniform coiling pattern. In their analysis, it was shown that the longitudinal and lateral excitation due to the Lebus device forms the primary source of excitation to the system. The excitation is defined with reference to the co-ordinate system illustrated in figure 1.4. The x axis is parallel to the drum axis, whilst the y,z directions represent the normal and tangential co-ordinate directions of the drum surface.

Consider a rope being coiled onto the drum and taking up successive positions as illustrated in figures 1.3, 1.4.

- A pulse in the x direction occurs at each coil cross-over until a layer change, and then the direction of the pulse reverses.

[1]Other sources of excitation, neglected in this study, may arise due to ovality of the winder drum or head sheave, and shaft steelwork misalignment.