BUCYRUS ERIE 1570W DRAGLINE - ANALYSIS OF THE UPPER MAIN SUSPENSION SYSTEM

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A project report submitted to the Faculty of Engineering, University of the Witwatersrand, Johannesburg, in partial fulfilment of the requirement for the degree of Master of Science in Engineering

JOHANNESBURG 1989

I DECLARE THAT THIS PROJECT REPORT IS MY OWN, UNAIDED WORK. IT IS BEING SUBMITTED FOR THE DEGREE OF MASTER OF SCIENCE IN ENGINEERING AT THE UNIVERSITY OF THE WITWATERSRAND, JOHANNESBURG. IT HAS NOT BEEN SUBMITTED BEFORE FOR ANY DEGREE OF EXAMINATION IN ANY OTHER UNIVERSITY.

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ABSTRACT

Boom supporting pendants are considered to be one of the more critical dragline components. They are subjected to a high degree of dynamic loading throughout their relatively short, operational life and are inaccessible to conventional maintenance and inspection practices. The consequences of failure are extreme and a need exists to understand the physical and mechanical properties of the support mechanism.

Therefore, this report discusses the results of an investigation to determine the pendant loadings of the boom support system, with particular emphasis on the degree of load-sharing along with the operational factors, which affect their performance. The equilibrium conditions of the upper main suspension are determined by means of a computer analysis, and the factors which identify the pendant equalisation link as the source of a major geometric problem are discussed. A number of practical techniques are presented to overcome the deficiencies identified by the investigation.

ACKNOWLEDGEMENTS

The author would like to express his appreciation and indebtedness to the following:

Mr D. Cipolat, Faculty of Engineering, Project Supervisor, for his valuable counsel and guidance.

Mr Andrezej Klos, AAC Mechanical Engineering Department for the benefit of frequent discussions and his interest and encouragement, as well as his colleague, Mr Alf van Dijk, for his assistance during the test period.

The permission of the management of the Coal Division of Anglo American Corporation to publish this report is acknowledged. To them, and to Mrs Ines Lowe and Mrs Adele Palmer, who did the typing, I express my thanks.

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LIST OF SYMBOLS

Quantity	Symbol
1. Mensuration	
angle	
area	A
co-ordinates	x, y, z
diameter	d
distance along path	s
height	h
length	l, L
radius	r
2. Mechanics	
acceleration	a
coefficient of friction	μ
force	F
mass	m
moment of force	M
horizontal force	H
vertical force	V
strain	ϵ
stress	σ
weight	G
Young's modulus	E

CHAPTER 1

1 INTRODUCTION

1.1 The Evolvement of the Walking Dragline

The United States of the 19th Century still relied on slow and costly wagon haulage for the distribution of bulk commodities to the interior of the country. Water transport was the key to the opening of the interior, but at that time canal building was painfully slow with only about 170 kilometres of canals having been completed. This problem, together with the start of railway construction, brought a market for practical and economical dry-land excavating machines.

Hollingworth,[1] in his History of Strip Mining, described how the first excavating machinery was developed originally for dredging rivers and harbours. Dredging machinery dates back to at least the 18th Century, as in 1796 the firm of Boulton and Watt built and installed a 3kW steam engine of Watt's design in a scow, from which it was used to operate spoon dredge machinery. Some seven years later, steam power was applied to elevator dredges and their use spread rapidly.

Adopting the existing design of steam dredge to a land based requirement was so difficult, that none of the early attempts was really successful. Then in 1835 a new type of excavating machine, a power shovel, was developed by a young American, William Smith Otis. His invention, illustrated in Figure 1.1, was a major step forward in excavation, swung from the centre, and had a power thrust to adjust the radial thickness of the cut. These features were combined with existing designs of a self-propelled car body, mounted on standard gauge railroad wheels and axles. The three basic motions: hoisting, swinging, and thrusting were all operated by steam power to produce a machine recognizable today. The terminology is self-explanatory. Hoisting refers to the raising of the bucket by chains or cables passing over a supporting boom. Swinging refers to the excavator's turning motion, which in the original machine was delivered by a chain wrapped around a swing circle. Thrusting (crowding) is the motion of the bucket and bucket handle away from the boom towards the work. When excavating sites moved away from rails, as in the case of highway building and urban construction, additional mobility was required. This was first supplied by traction wheels and later by crawler mountings. Meanwhile, the more efficient and versatile full-revolving design was introduced. Next came the replacement of steam power by more easily managed and economical internal combustion engines and electric motors.

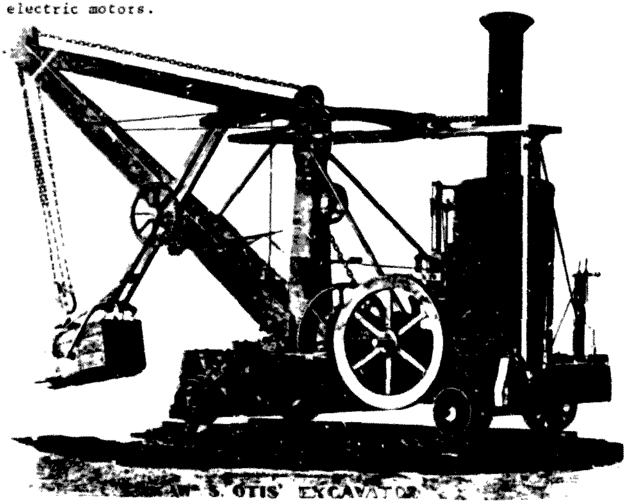


Figure 1.1 Power Shovel developed by W.S. Otis in 1835 (Hollingworth (1))

From this beginning the walking dragline has been developed primarily to suit various applications in the United States. The great flood control projects on the Mississippi and other rivers required long cantilevered boom machines with low bearing pressures to excavate river channels and to build levees in the soft river bottom lands.

Draglines were used as auxiliary machines for many years in conjunction with stripping (power) shovels in surface coal mining operations, but it was not until 1930 that they began to be considered as primary excavators for overburden removal. By 1940, while stripping shovels were considered the favoured tool in most operations, the dragline was being applied where overburden depths were too great for stripping shovel stacking capacity.

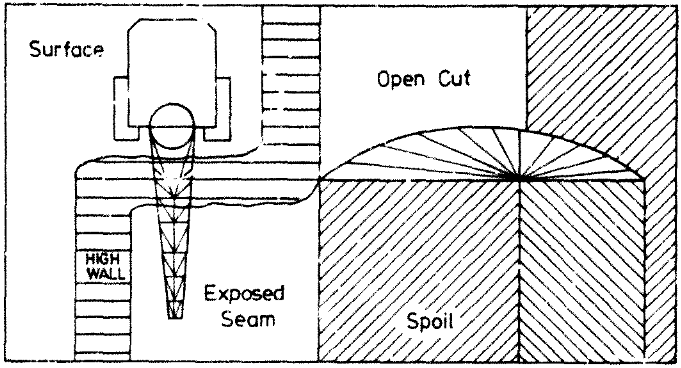
In the years between 1940 and 1960 about nine walking draglines with an average bucket capacity of 11 cubic metres were placed in service. Two thirds of them were applied to coal mining operations and the bulk of the remainder to other mining operations, primarily phosphate. Almost all of this capacity was utilised in the United States. The following decade reflected a dramatic increase in dragline capacity as a result of the intensive development in dragline design and application. Maximum machine size increased tenfold and boom lengths doubled during this period. Since that time bucket capacity has continued to increase and while gradual increases in boom length have continued, machine size has remained stable.

It is significant that in the past twenty years dragline operations outside the United States have absorbed a third of the new capacity as compared to a minor proportion of machines before that period. The first large walking dragline in South Africa was commissioned during August 1971 and since that time the local dragline population has increased to 22 units.

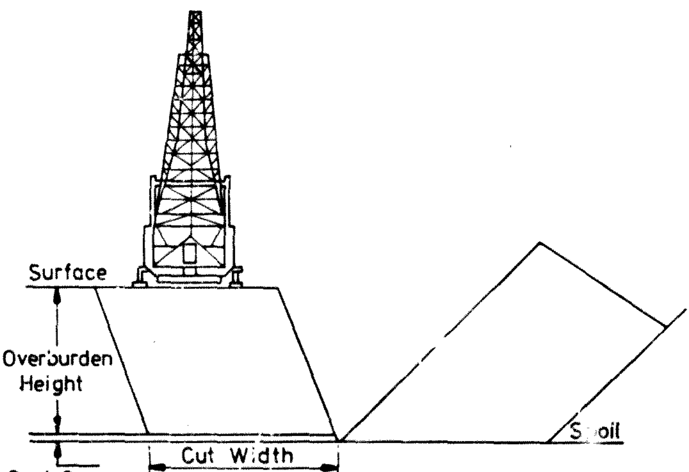
1.2 Dragline Application

Most draglines have been applied to coal deposits, where the average overburden depth does not exceed 40 metres. Figure 1.2 shows a typical dragline pit. The type of overburden may vary from loosely consolidated to highly consolidated sedimentary materials, such as sandstone or limestone, which require fragmentation by drilling or blasting. The major function of the dragline is to excavate a long narrow trench in the overburden, exposing a strip of coal. After the coal has been removed, the dragline reverses and excavates a new seam parallel to the previous cut dumping the overburden where the coal has been removed. For this reason, draglines require long booms to increase the range between the drag and dump points. Other important factors are large bucket capacities and rapid swing cycles.

A dragline supports the bucket with wire ropes. (See Figure 1.3). The bucket is hoisted by pulling ropes over sheaves at the boom point and is dragged by ropes passing near the boom foot. The hoist and drag functions must be co-ordinated to obtain the digging operation. As long as the ropes remain taut, the bucket will remain upright. When the drag ropes are relaxed, the bucket dumps. For a normal digging cycle the bucket is lowered vertically from the boom point until it contacts the overburden to be removed, where it is dragged towards the foot thus filling the bucket. With hoist and drag ropes taut, the bucket is reeled out and up until it is almost at the boom point, where the drag rope tension is released, the hoist ropes stopped and the bucket dumps. The swing motion is co-ordinated with hoist and drag to dig and dump at almost any desired swing cycle. In the simple case shown in Figure 1.2 the dragline sits on top of the material to be excavated and swings through an arc of between 45° and 90° to dispose of the material. A typical average cycle time for this operation is 45 seconds.



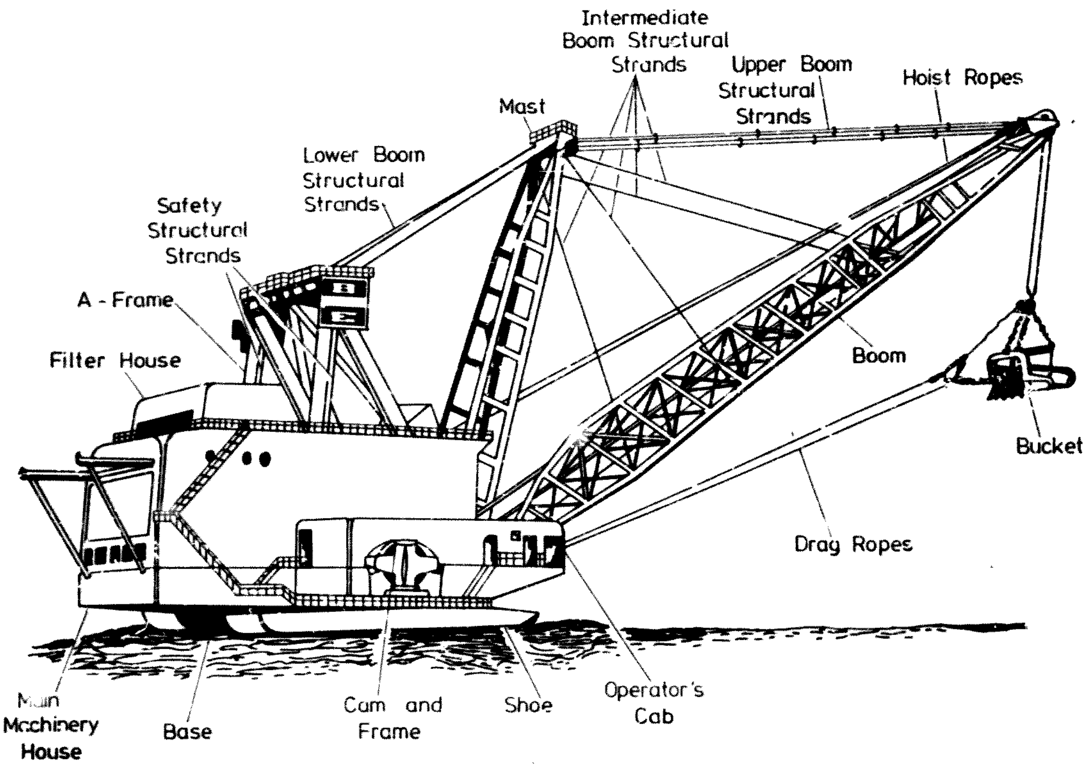
PLAN VIEW



SECTION VIEW

TYPICAL PIT ARRANGEMENT

FIG. 1-2



SCHEMATIC VIEW OF BUCYRUS ERIE 1570W WALKING DRAGLINE

FIG. 1-3

Draglines used in opencast mining typically range in size from machines equipped with 5 cubic metre drag buckets on 35 metre booms to a monster 168 cubic metre drag bucket on a 95 metre boom. A number of Bucyrus Erie 1570W draglines are operating in South Africa. The dragbuckets on these machines range in size from 53,5 to 61 cubic metres depending on the boom length.

All large draglines are powered by electric motors, which are fed by 6600 volt trailing cables supplied from an input 22 kV overhead powerline. Mobility for these large machines is provided by a cam and shoe mechanism on each side of the machine. These lift the rear of the machine and simultaneously move it forward. Walking speed is in the region of 225 metres per hour.

1.3 Statement of Problem

In South Africa today business is at a crossroads. For many years industry has not developed the skills of technological expertise, due to its easy availability from foreign sources. With the advent of sanctions, industrialists will now be forced to compensate by long term development of staff and technological expertise, viz-a-viz the short term needs to apply technology for immediate commercial gains. Emphasis must be placed on looking internally rather than externally for the solution of these difficulties. Therefore a conscious effort has been made to formulate a strategy to offset the loss of foreign technology with the ultimate objective of becoming self-sufficient.

As in the past, the trend in the mining industry is towards larger equipment. Mining companies usually seek out the single dragline, which will be able to handle the entire stripping requirement. This together with an operational requirement of about 7500 hours per annum requires that emphasis be placed on developing components with high levels of reliability and predictability. This will ensure that repairs and replacement of components can be scheduled at times that will least affect the overall mining operation.

The greatest assets of a dragline are inherent in the actual design of the machine. The greater the reach and dumping radius, the deeper the level of overburden, which can be removed. Overburden up to 35 metres is well within the dumping reach of existing dragline capacities and removal can be achieved at substantially less total cost than by any other method. As overburden depth increases longer booms are required to dispose of the spoil at an adequate distance. Machines with 120 metre booms are going into service, and it seems likely that boom length will increase further. To maintain the same excavating capacity larger machines are required and both investment and operating costs per cubic metre will be higher than with a machine of the same capacity with a shorter boom.

From this objective, providing Bucyrus Erie 1570W draglines with a local supply of suspension rope pendants has proved to be a difficult and time-consuming operation. It highlighted the complication of working to extremely tight tolerances over long lengths and determined a need for effective length compensation. Following installation of the first set of locally produced pendants it was observed that the catenaries in adjacent ropes, connected to the same link, were different. It was naturally assumed that this was caused by errors in the lengths of the ropes, which resulted in differing rope tensions and catenaries. This initiated a set of field tests[8] to measure the loads on both the upper and lower main suspension pendants. The results of these tests were reported in September 1984 and January 1986. They established that indeed the loads in the ropes were not shared and variations in rope tensions in excess of 100% were found.

The effect of unequal load sharing magnifies the already high degree of static and dynamic load as seen by the structural members in normal operation. High static and dynamic load levels can increase the susceptibility of the wires to deteriorate through normal fatigue mechanisms. The rate of deterioration can be significantly worsened by the effects of environment. With maintenance and inspection of the pendants difficult, it is vitally important to limit these deficiencies as much as possible.

A further factor for consideration is damage or failure of a component of an operational dragline boom. The damage could be as a result of an accident or a fatigue-induced failure during operation. This could be confined to a particular area of the boom where collapse may occur unless there is some geometric stability remaining after the damage.

1.4 Scope of Project

In order to meet these challenges a need exists to study critically and evaluate dragline operation and boom loadings in detail. Therefore, the initial purpose of the research is to develop an understanding of the physical and mechanical properties of the boom suspension system. Pendant loadings are to be determined by means of a strain gauge analysis and those operational factors, which have an influence on their performance, are to be identified. In addition, it is required to establish the origin of the poor load sharing previously noted between a set of upper and lower main suspension pendants. Critical components are to be identified and their role in boom structural integrity ascertained. Practical solutions will be presented to overcome any deficiencies highlighted by the analysis.

C H A P T E R 2

2 CONSIDERATIONS IN BOOM DESIGN

2.1 Boom Loading

The boom is probably the most critical single component of a dragline, from an engineering standpoint. This seemingly simple structure must be designed to withstand a medley of forces that interact and reverse continually during the loading, hoist, swing, dump and return operations.

Loading of a boom can be categorised into three major areas.

(1) The compressive load as the boom itself is basically loaded as a column.

(2) Because the boom is not vertical, but supported at a lower angle, the column deflects due to deadweight with consequential loadings, which are additive to the working loads applied to the boom.

(3) The dynamic loadings which occur as a result of the swing cycle.

For a better understanding of the overall loading[2] of a boom, together with design considerations and rationale, consider the following:

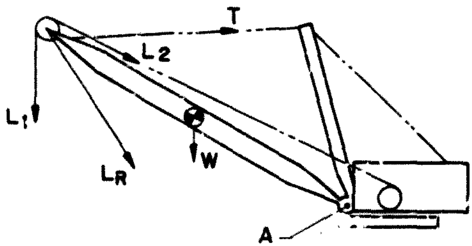


Figure 2.1 Simplified Boom Loading Diagram
(Sankey (2))

In Figure 2.1, the bucket load is shown as Force L_1, which is supported over the boom point sheave and a Weight W of the boom. The Loads L_1, supporting the load and L_2 to the winch, can be resolved into one Load L_R acting on the end of the boom. The resultant Force L_R and the weight of the boom are supported by the boom support pendants, which are in tension, and the boom itself, which is in compression.

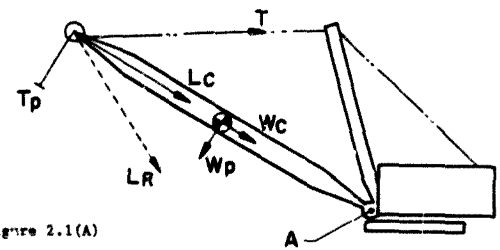


Figure 2.1(A)

This shows the boom loads resolved into forces of compression L_C, which act down the centreline of the boom through the boom foot, and forces L_P perpendicular to the boom, which tend to rotate the boom about Point A. The load in the support pendants T will be that force necessary to prevent rotation of the boom and maintain static balance.

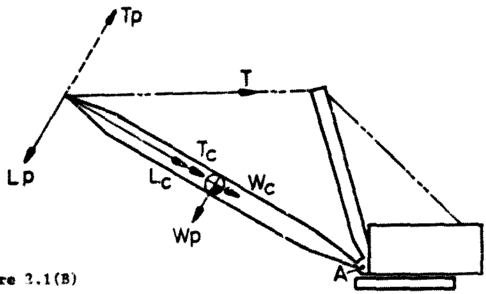


Figure 2.1(B)

The Force T is resolved into forces T_p perpendicular to the boom to prevent rotation and its component T_C down the boom, which is additive to the compressive forces. Thus, one can see that the loads perpendicular to the boom are cancelled to maintain equilibrium, and the resultant force on the boom is the additive effect of the compressive Loads L_C due to the working load, T_C due to the rope tension, and W_C due to the boom weight.

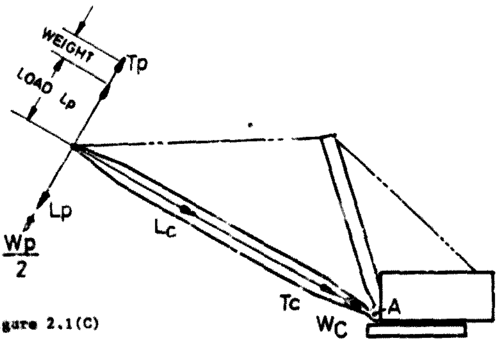


Figure 2.1(C)

The resultant working force on the boom is illustrated in Figure 2.1.(C) as a compressure force acting along its centroid.

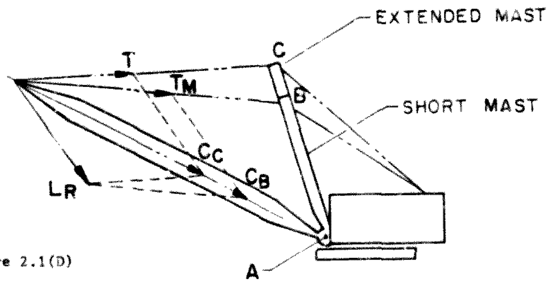


Figure 2.1(D)

Figure 2.1(D) illustrates one method the engineer has for changing the magnitude of force down the boom. Since the boom is *a pinned structure, the compressive* load must always act down the centroid of the boom. By combining the Loads L_R and T_M which are pinned off at Point B, the resultant load is shown as Force C_B. If the mast is extended to Point C, then the combined load of L_R and T_C is shown by the Load C_C. By observation C_C is smaller than C_B. thus, one can conclude that the higher the mast, the lower the compressive load in the boom for a given work load.

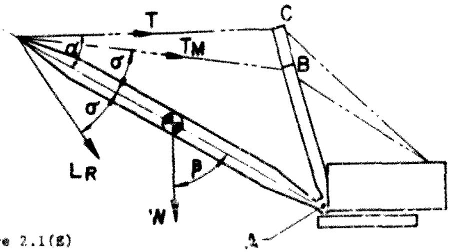


Figure 2.1(E)

Figure 2.1(E) is a mathematical representation of the same principle, which shows the compressive load down the boom to be a function of the Load T in the ropes and the angle between the ropes and the boom. This force varies as the cosine of the angle, thus as the angle between the rope and the boom increases for a higher mast - the compressive load down the boom decreases.

CHAPTER 3

3 SUSPENSION PENDANTS AND ATTACHMENTS

3.1 Pendant Configuration

The boom of a Bucyrus Erie 1570W dragline is supported by four upper main fixed length suspension pendants. These pendants are solidly connected at the boom point box, and by bronze bushed equalizer links to the mast head.

Two sets of intermediate suspension structural pendants support the lower boom chords at two points on either side of the boom. The structural pendants are equipped with hydraulic adjusting jacks and are connected to the boom chords by open strand sockets mounted on spherical ball bushed pins.

Spacer blocks are located along the length of the upper main boom support pendants. The spacer blocks are clamped between the two structural pendants on each side of the boom and are used to minimize pendant vibration during the digging cycle.

3.2 Stress Strain Properties of Steel Cables

Three different types of steel cables are common in structural applications: structural strand with individual wires wound round a core, structural rope with several strands of wire, finer in diameter than in the strand, wound around a core and parallel wire strands with the wires bundled not twisted. Due to the differing methods of manufacture the effective modulus of elasticity varies. The more pronounced influence of the helices gives the rope the smallest modulus, with an approximate value of E equal to 140 GPa. This may be contrasted with 170 GPa for a structural strand and 190 to 210 GPa for a parallel wire strand.

Because the elastic limit is not clearly defined (see Figure 3.1) the modulus of elasticity is usually calculated from the slope of the straight line that connects the 10 percent breaking load with the 90 percent pre-stretching load of the cable specimen. A pre-stretching load of about 55% of the breaking load is usually applied to remove the constructional looseness of the cables. This results in extension by the bedding down of the assembled wires with a corresponding reduction in the overall diameter. This reduction in diameter creates an excess length of wire which is accommodated by a lengthening of the helical lay. When sufficiently large bearing areas have been generated on adjacent wires to withstand the circumferential compressive loads, this mechanically created extension ceases and elastic extension commences.

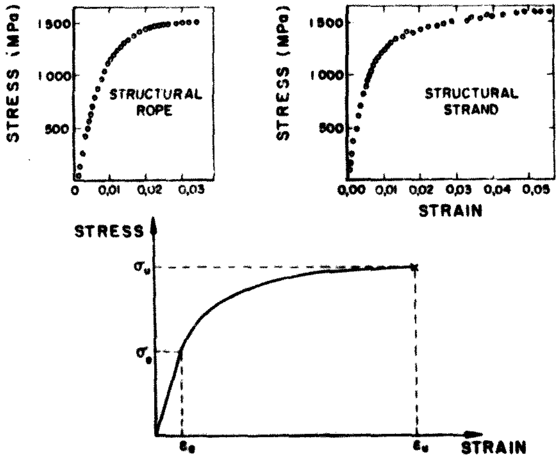


Figure 3.1 Stress-Strain Relationships of Steel Wire Cables (Irvine (4))

Typically the elastic limit σ_e is reached at about 50 percent of the ultimate tensile strength σ_u. Ultimate tensile strengths of 1500 MPa are regularly achieved, with ultimate strains of around 3 percent for rope and 6 percent for strand. Tests show that strand is stronger than the more flexible rope of the same size. Strength and stiffness, based as they are on the nominal cross-sectional areas, are affected by the class of zinc coating. Factors of safety vary, but a working stress of approximately 1 MPa (giving a factor of safety of 6 on the ultimate tensile strength) is routinely used on dragline applications.

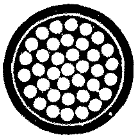


Figure 3.2 Cross Section of a typical Bridge Strand (Haggie Rand (11))

Bridge strand used on Bucyrus Erie 1570W draglines have the following specifications:

	Lower Main	Upper Main	Intermediate
Diameter (mm)	98	89	57
No. of wires*	283	243	146
Wire tensile (MPa)	1570	1570	1470
Metallic Area (mm^2)	5639	4700	1968
Modulus of Elasticity (GPa)**	147,15	148,6	171,7
Mass/m (kg/m)	47,95	39,73	16,27
Estimated breaking force (kN)	7985	6762	2766

* 283 is laid up (54/48/42/36/32/26/20/14/8/3) All
243 is laid up (51/54/39/33/27/31/15/9/3) right hand
146 is laid up (40/34/24/18/12/6+6/6/1) lay

** $\pm$ 3% after prestressing.

Table 3.1 Specifications of Bridge Strand used on Bucyrus Erie 1570W Dragline Suspensions

3.3 Manufacture of Pendant Assemblies

As illustrated in Figure 1.3 the boom of the dragline is supported by four fixed length suspension pendants. The dragline manufacturers specification requires that the pendants are measured under load such that the finished lengths are

Upper Mains 62,176 m ± 0,006 at 1197 kN

Clearly this requires that close attention be paid to all aspects of measurement and the attachment of the terminations.

Bridge strand usage in South Africa is almost exclusively restricted to the replacement of dragline suspension pendants. As this constitutes a low and intermittent consumption, local manufacture has not been considered and alternative means to supply these critical and varied items has had to be found.
Long lengths, up to 1000 m, of prestressed strand was imported by the local wire rope manufacturer from which the assembly of the various lengths of pendants can be undertaken. The local manufacturer, Haggie Rand, have installed a 66 m long tensioning/compression frame, which enable the strand to be loaded to 1200 kN and thus produce matched sets of pendants well within the specified tolerances.

For this investigation a full set of upper main suspension pendants was calibrated by cycling the rope at 1600 kN until there was no measurable change in length, a minimum of three load cycles being specified. In the tests carried out three cycles were in fact sufficient to "settle" the rope. Once this was done the measurement of length was undertaken.

The technique used to measure length was to hang plumbobs from the centre of a permanent socket pin to the corresponding socket pin of a dummy socket. The bobs are suspended directly above graduated measuring scales fixed to the foundations of the tensioning frame. The distance between the measuring scales had been set up using a calibrated steel tape. Temperature effects which might alter the absolute value of length measurement obtained are minimised by measuring during the night. Minor variations in temperature are standardised by a length adjustment using a coefficient of linear expansion for strand of $12{,}5 \times 10^{-6}$ per ^{o}C of temperature variation.

At this point the pendant is removed from the tensioning frame for the replac[illegible] of the dummy socket by the permanent strand termination [illegible]e is taken following the establishment of the pend[illegible]h i[illegible] positioning the cast metal wedge socket [illegible]sulation to ensure specified length tolerances. The [illegible] strand is lifted into a socketing tower, which will allow at least 48 strand diameters to hang vertically downwards from the socket. Encapsulation is then undertaken in accordance with British Standard 463. The casting metal used for producing the wedge socket must be a near entectic alloy, such that the effect of temperature during casting does not result in any damage to the wire material, while ensuring a uniform distribution of the load over the wires together with a strong adhesive bond.

Following the completion of this permanent termination, the pendant is re-installed in the tensioning frame and a tensioning load of 1700 kN applied. This application of load is beneficial in standardising the effects of wedge creep on matched sets of pendants. Due to the shrinkage of the casting metal on solidification and because of the presence of micro blowholes and deformation of the casting metal, some pullback of the casting cone in the socket basket can be observed. This is dependent on the degree of applied load and should be confined to a few millimeters in a properly prepared termination. Table 3.2 shows the results of pullback of a locked coil suspension rope of 69 mm diameter with increasing load as determined by Schneider[3] in his investigation into the anchoring of suspension ropes.

Load kN	Pullback mm
600	0
800	0,[illegible]
1000	0,36
1200	0,55
1600	0,90
2000	1,19
2400	1,45

Table 3.2 Pullback (creep) of Locked Coil Suspension Ropes (Schneider (3))

A matched set of pendants is assembled from the same original length of strand. The results of the length measurements for the pendants installed on the Kleinkopje dragline, following the verification test was within ± 3mm, see table 3.3.

Pendant Position	Tensioning Data		Corrected Length
	Ambient Temp. oC	Measured Length M	M
L.H. Top	12	62,169	62,175
L.H. Bottom	13	62,170	62,174
R.H. Top	13	62,169	62,174
R.H. Bottom	14	62,169	62,174

Table 3.3 Upper Main Pendant Length Measurements as installed on the Kleinkopje Dragline

CHAPTER 4

4 FIELD TESTING

4.1 Boom Raising Procedure

Raising and lowering the boom is done by using a separate boom hoist rope, reeved between sheaves on the A-frame and the top of the mast, and attached to the drag drum to provide the necessary leverage. The upper main suspension pendants, as well as the intermediate suspension pendants, are attached to the mast prior to raising. Rotation of the drag drum raises the mast, and in turn the dragline boom, into a position, where the lower main suspension pendants are connected between the A frame and the mast. Releasing the boom raising rope transfers the weight of the boom onto the structural pendants. The boom hoist rope is then released, unreeved and stored for future use.

4.2 Intermediate Suspension Ropes

Dragline booms are supported in such a manner as to eliminate, or at least minimise, the effect of deadweight bending. On Bucyrus Erie 1570W draglines two pairs of intermediate suspension pendants are positioned above and below the midpoint of the boom. This has the effect of segmenting the boom into shorter members, which support the bending components.

They are further intended to react the centrifugal forces generated when the machine is swinging. The forces would otherwise cause the boom to deflect radially outward and, since the boom is essentially a column, a buckling failure could ensue due to the eccentricity so generated.

Following installation it is a requirement of the manufacturer that these ropes be pretensioned to a normal static tension of 260 kN for the upper intermediate ropes and 285 kN for the lower intermediate ropes. Tensioning is achieved by means of a portable hydraulic jacking system (Figure 4.1). A portable jacking unit is connected to the cylinders and when the correct pressure is achieved, the tension is mechanically secured by means of locknuts. Tensions on both the upper and lower, left and right hand ropes are set independently but simultaneously. It was found that, to achieve the specified preloads, several applications of tension by the jacking system on individual ropes were necessary to obtain a balanced result.

4.3 Measurement of Pendant Loadings

Two basic techniques have been used for the measurement of suspension pendant loads either direct strain readings on legs of sockets or indirectly from the pressure gauge provided for the intermediate pendant tensioning system. Figures 4.1 and 4.2 show the principles of both types of measurement. It is essential that strain gauge bridge circuits are arranged, so that extraneous bending and torsion effects are eliminated, and so that the correct loads are produced, when the loading between the legs of sockets is not perfectly central.

In most cases, calibration is desirable, but this has not been possible in the absence of a suitable spare socket. Exploratory work has beeu undertaken however, by employing the results of a previous calibration exercise on a socket analagous to that used in the Kleinkopje measurements. It is estimated that the accuracy obtained by this method is within 10-15 percent of actual values. This exercise is detailed under Appendix B.

Prior to the recording of suspension pendant tensions, the zero stress reference position was established. This refers to the

PRELOAD INSTRUCTIONS

MODEL	BOOM LENGTH	WORK ANGLE	LOT NO	PRELOAD OF EACH UPPER INTER. SUSP STRAND			PRELOAD OF EACH LOWER INTER SUSP STRAND		
				BOOM HORIZ	BOOM 0-S L	EMPTY BUCKET	BOOM HORIZ	BOOM 0-S L	EMPTY BUCKET
1 570W	94.488M	38°	33	333kN	260kN	327kN	401kN	28[illegible]kN	292kN

NOTE.- 1- PRESSURE GAUGE READING INDICATES HALF THE PRELOAD DIRECTLY IN KN

2- S.L. MEANS SUSPENDED LOAD.
O-S.L. DETERMINES LOAD CASE 1

3- EMPTY BUCKET MEANS EMPTY BUCKET SUSPENDED BENEATH BOOM POINT. DETERMINES LOAD CASE 2

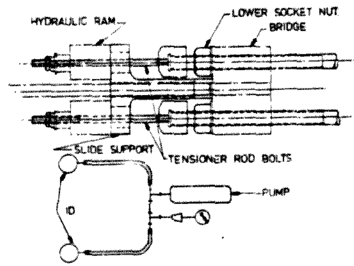


Figure 4.1 Hydraulic Tensioning Arrangement for Intermediate Suspension Pendants

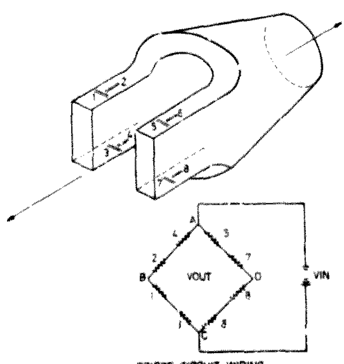


Figure 4.2 Schematic for Poisson Strain Gauge Bridge on a hot metaled socket

condition at which the strain gauges are calibrated to a zero stress level. It was obtained by supporting the boom on the ground with all pendants and boom raising ropes slack. All subsequent strain measurements were then taken in three well defined load conditions.

- bucket and rigging on the ground
- bucket empty, suspended below boom point box
- bucket full, suspended below boom point box

Following the raising of the boom and the pretensioning of the intermediate pendants, with both bucket and rigging on the ground as required by the manufacturer, the dragline was operated on normal excavation for a few days to "settle" the boom suspension system. Prior to any measurements, the intermediate pendant pretensions were again checked and readjusted to conform with the manufacturer's requirements. The strain gauges were then connected to the switch and balance unit and the zero stress position, as established with the boom on the ground, set for the strain gauges on the strain indicator.

The test sequence for the boom suspension gauges consists of:

1. Balance and calibrate the strain gauges with the bucket on the ground and with the spreader bar and dump block weight off the hoist ropes. The bucket should be positioned directly below the boom point sheaves equivalent to maximum reach. This position represents load case 1.

2. Hoist the empty bucket at maximum reach to a position just below the hoist trip-out point. This position represents load case 2.

3. Hoist a full bucket at maximum reach to a position just below the hoist trip-out point. This position represents load case 3.

4. On completion of the full bucket test, return to the starting point and set the bucket on the ground, making sure that the spreader bar and dump block weight are off the hoist ropes. Recheck the strain gauge calibration.

The tests were established with the dragline base supported under average ground conditions, with the supporting surface blasted, covered with one metre of shale and dozed level.

To establish the effect of a variation in intermediate pendant pretensions, on completion of the series of tests as detailed above, a right lower intermediate pendant pretension was reduced by 40 kN. The remaining intermediate pendants pretensions were not readjusted to their specified values, but allowed to compensate for the maladjustment to the right lower pretension. A second series of measurements was then undertaken using the previous test sequence.

This was followed by a further set of measurements to establish the effect of a variation to a right upper intermediate pendant, also subject to a reduced pretension of 40 kN. Before this adjustment was undertaken all intermediate pendant pretensions were re-established to the manufacturer's specified values and thereafter allowed to compensate for the reduced right upper loading.

The full series of test results for the Kleinkopje machine are summarised in Table 4.1. Table 4.2 shows the results and illustrates the extent of imbalance in maximum loads that can occur, because of errors in setting initial pretensions.

Rope Loadings (kN)

Rope Position	Bucket on Ground	Empty Bucket	Full Bucket
Preloads as specified			
Left Top	290	519	882
Left Bottom	438	719	1082
Right Top	248	471	810
Right Bottom	366	637	1006
Left Upper Inter.	260	320	379
Right Upper Inter.	260	320	395
Left Lower Inter.	285	285	302
Right Lower Inter.	285	285	285
Preload reduced on right lower intermediate			
Left Top	255	494	872
Left Bottom	402	689	1084
Right Top	225	433	801
Right Bottom	374	630	1010
Left Upper Inter.	212	285	395
Right Upper Inter.	250	231	363
Left Lower Inter.	290	299	302
Right Lower Inter.	244	244	244
Preload reduced on right upper intermediate			
Left Top	236	456	801
Left Bottom	395	666	1034
Right Top	205	412	738
Right Bottom	368	639	971
Left Upper Inter.	261	319	395
Right Upper Inter.	221	296	383
Left Lower Inter.	284	267	284
Right Lower Inter.	284	287	290

Table 4.1 **Field Test Results for both Upper Main and Intermediate Suspension Pendants as measured on the Kleinkopje, Bucyrus Erie 1570W Dragline**

Rope Loadings (kN)

Percentage Variation

Position		Load Case 1 Left	Load Case 1 Right			Load Case 2 Left	Load Case 2 Right			Load Case 3 Left	Load Case 3 Right	
Top		290	248			519	471			882	810	
	51,0			47,6	38,5			35,2	22,7			24,2
Bottom		438	366			719	637			1082	1006	
		728	614			1238	1108			1964	1816	
		18,6				11,7				8,2		
Preloads reduced on Right Lower Intermediate												
Top		255	225			494	433			872	810	
	57,6			66,2	35,2			45,5	24,3			26,1
Bottom		402	374			689	630			1084	1010	
		657	599			1183	1063			1956	1820	
		9,7				11,3				8,0		
Preloads reduced on Right Upper Intermediate												
Top		236	205			456	412			901	738	
	67,4			79,5	46,1			55,1	29,1			31,6
Bottom		395	368			666	639			1034	971	
		631	573			1122	1051			1835	1709	
		10,1				6,8				7,4		

Table 4.2 Extent of load imbalance in Upper Main Suspension Pendants due to a variation in setting of intermediate pendant preloads.

Rope Loadings (kN) | Percentage Variation

Position	Load Case 1 Left %	Load Case 1 Left	Load Case 1 Right	Load Case 1 Right %	Load Case 2 Left %	Load Case 2 Left	Load Case 2 Right	Load Case 2 Right %	Load Case 3 Left %	Load Case 3 Left	Load Case 3 Right	Load Case 3 Right %
Top	51,0	290	248	47,6	38,5	519	471	35,2	22,7	882	810	24,2
Bottom		438	366			719	637			1082	1006	
		728	614			1238	1108			1964	1816	
		18,6				11,7				8,2		
Preloads reduced on Right Lower Intermediate												
Top	57,6	255	225	66,2	35,2	494	433	45,5	24,3	872	810	26,1
Bottom		402	374			689	630			1084	1010	
		657	599			1183	1063			1956	1820	
		9,7				11,3				8,0		
Preloads reduced on Right Upper Intermediate												
Top	67,4	236	205	79,5	46,1	456	412	55,1	29,1	801	738	31,6
Bottom		395	368			666	639			1034	971	
		631	573			1122	1051			1835	1709	
		10,1				6,8				7,4		

Table 4.2 Extent of load imbalance in Upper Main Suspension Pendants due to a variation in setting of intermediate pendant preloads.

CHAPTER 5

5 STATIC ANALYSIS OF CABLE STRUCTURES

A cable assembly of this form belongs to the category of geometrically nonlinear structures, because they are structural mechanisms, the stiffness of which depends upon the stiffness in the cables and because the high tensile steel used in the wire permits strains which are approximately six times greater than those used in ordinary structural steel.

The non linear behaviour of this type of structure must be taken into account in analysis, since conventional linear methods will overestimate the displacements when the structure is stiffening, and underestimate the displacements when it is softening. A unifying approach to the analysis of both linear and non linear structures is to consider the determination of a system equilibrium equation and its solution by means of an iterative process, the position of equilibrium being reached when both sides of the equation are equal.

5.1 The Elastic Catenary

When freely hanging cables are connected to a common attachment, general linearised solutions developed to the solution of catenary problems are no longer adequate, and other methods of determining cluster response must be found. One possibility is to use the finite element method by dividing the cable up into several elements, writing approximate equilibrium and compatibility equations for each element, combining these into a global system for the complete pendant assembly, and so proceeding to determine its response. An alternative approach is, however, to use the exact analytical solutions for the elastic catenary developed by Irvine [(4)] in his book, Cable Structures, which in contrast to the multi-element techniques, represent the pendant by a single element. The potential saving in computer time makes this method more attractive for static response calculations.

Dragline suspension pendants resist applied load by changes in both tension and geometry. *A characteristic of relatively* flat suspended cables, is that small changes in cable length give rise to substantial changes in cable geometry.

The suspended cable shown in Figure 5.1 is suspended between two fixed points A and B which have Cartesian co-ordinates (0,0) and (l,h), respectively. Thus the span of the cable is l, and the relative vertical displacement of the ends is h. The unstrained length of cable is where L_o is not necessarily greater than $(l^2 + h^2)^{1/2}$, although it *obviously* cannot be much less if Hooke's law is not to be violated. A point in the cable has Lagrangian co-ordinate s in the unstrained profile (the length of cable from the origin to that point is s when the cable is unloaded). Under self-weight of W $(= mgL_o)$ this point moves to occupy its new position in the strained profile described by Cartesian co-ordinates x and y and Lagrangian co-ordinate p.

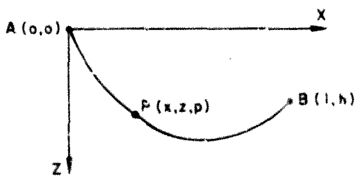


Figure 5.1 Co-ordinates for the Elastic Catenary (Irvine (4))

The geometric constraint to be satisfied is

$$\left(\frac{dx}{dp}\right)^2 + \left(\frac{dz}{dp}\right)^2 = 1$$

while reference to Figure 5.2 the balancing of horizontal and vertical forces yields

$$T \frac{dx}{dp} = H$$

$$T \frac{dz}{dp} = V - W \frac{s}{L_o}$$

where, owing to conservation of mass, the weight of that portion of the strained profile shown in the figure is simply W_s/L_o. The vertical reaction at the support is V and H is the constant horizontal component of cable tension.

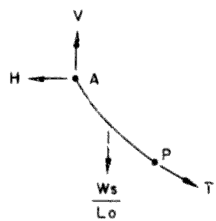


Figure 5.2 Forces on a segment of the strained cable profile (Irvine (4))

A constitutive relation that is a mathematically consistent expression of Hooke's Law is

$$T = EAo\left(\frac{dp}{ds} - 1\right)$$

where E is Young's modulus, A_o is the uniform cross-sectional area in the unstrained profile.

The end conditions at the cable supports A and B are

$x = 0 \quad z = 0 \quad p = 0 \quad s = 0$

$x = l \quad z = h \quad p = L \quad s = L_o$

where L is the length of cable in the strained profile.

Irvine now derives the parametric solution that describes the strained cable profile.

$$x(s) = \frac{Hs}{EA_o} + \frac{HL_o}{W}\left[\sinh^{-1}\left(\frac{V}{H}\right) - \sinh^{-1}\left(\frac{V - Ws/L_o}{H}\right)\right]$$

$$z(s) = \frac{Ws}{EA_o}\left(\frac{V}{W} - \frac{s}{2L_o}\right) + \frac{HL_o}{W}\left[\left(1 + \left(\frac{V}{H}\right)^2\right)^{1/2} - \left(1 + \left(\frac{V - Ws/L_o}{H}\right)^2\right)^{1/2}\right]$$

5.2 Pendant Stiffness Analysis

To determine the basic load deflection characteristic for dragline suspension pendants requires an understanding of their static and dynamic response. One way of thinking about this parameter is to consider it as resembling the stiffness of two springs in series. Consider a sagging cable in which the ends are being stretched apart. Some the resistance supplied is geometric, because the is being reduced. This stiffness is

$$\frac{12\ H}{l\left(\frac{mgl}{H}\right)^2}$$

The remaining stiffness is axial and qualified by a term like $\frac{EA}{l}$.

A combination of these two effects determines the response of the suspension pendant. Consider firstly, a pendant that hugs closely to the chord spanning the termination points and is therefore taut and straight. Such a cable must stretch to resist applied load, but this stretching is of the second order in the additional deflection, so that first order changes in tension are absent. On the other hand should there be appreciable sag in the catenary then the relative inextensibility typical of metal cable manifests itself. Additional tension can be generated to the f order, because the loaded pendant can opt a new profile that does not necessarily depend on changes in cable length for its existence. Somewhere between these two limits upper and lower main suspension pendants will be found to operate.

The resulting load deflection characteristic for an upper main suspension pendant is plotted in Figure 5.3 and is based on a nominal length of 62,176 m.

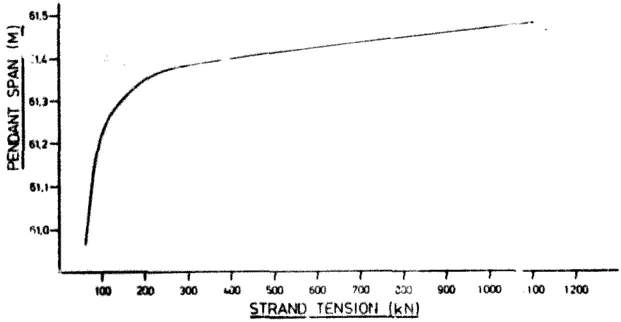


Figure 5.3 Load Deflection Characteristic of an Upper Main Suspension Pendant

5.3 Model Development

The cable support structure will be analysed as a single entity, and the analysis proceeds on the assumption that its top and bottom chords do not react independently. The relatively small weight of the cable has been included, so that the free hanging geometry is specified by the cable tensions, the position of the link and the span. The small longitudinal movements of the cable structure, associated with the vertical movement of the structure under load, must be allowed to occur freely. Even though the structural response is in general nonlinear, it is nonetheless elastic, so loading and unloading follow the same curve. Therefore the procedures are unaffected by the signs of the imposed increments.

Consider a cable support structure anchored in rigid supports, as shown in Figure 5.4. Suppose that under an applied vertical load the shear force at some cross section x along the span is S. Then from the vertical equilibrium as defined by Irvine, it is clear that

$$(H_u + h_u)\frac{d}{dx}(Z_u + w) + (H_l + h_l)\frac{d}{dx}(Z_l + w) = S$$

where H_u and H_l are the horizontal components of the tensions in the top and bottom chords, respectively. h_u and h_l are the additional horizontal components of cable tension owing to applied load, Z_u and Z_l are the initial profiles of the chords, and w is the additional vertical deflection.

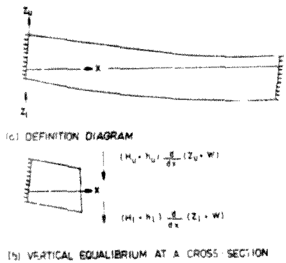


Figure 5.4 Equilibrium of a rigid support cable truss (Irvine 4)

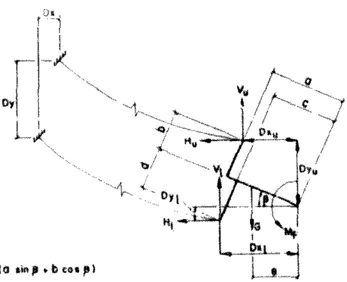
The internal equilibrium of such an unloaded structure can be expressed by

$$H_u\left(\frac{dz_u}{dx}\right) = H_l\left(\frac{dz_l}{dx}\right)$$

Although the dragline upper main suspension attachment at the boom point box may be considered to approximate a rigid support, the lower end of the pendants are connected at the mast head by means of a rotatable link. This therefore necessitates some modification to the general approach given above, in that the pendant profiles cannot be assumed to adopt uniform profiles even under equal length conditions.

Considering now the determination of an equilibrium equation for the dragline cable support structure over the desired range of dragline boom loading. For the mechanism shown in the accompanying Figure 5.5, the relationship between link geometry and motion can be found from consideration of the applicable geometric functions to be:

$$Dx_u = c\cos\beta - b\sin\beta$$
$$Dx_l = c\cos\beta + d\sin\beta$$
$$Dy_u = a\cos\beta - c\cos\beta$$
$$Dy_l = d\cos\beta - c\sin\beta$$



$$\Sigma M = M_F - V_u(a\cos\beta - b\sin\beta) + H_u(a\sin\beta + b\cos\beta)$$
$$- V_l(c\cos\beta + d\sin\beta) - H_l(d\cos\beta - c\sin\beta)$$
$$+ Ge\cos\beta = 0$$

Figure 5.5 Equilibrium of the Upper Main Suspension Pendant support system

If the chords are considered to be elastic catenaries, as developed in the previous section, the internal equilibrium in the initial profile is satisfied by the following dimensionless quantities

$$F * \mu * R = V_{u2} * Dx_u - H_u * Dy_u + V_{12} * Dx_1 + H_1 * Dy_1 - G * e * \cos \beta$$

where the horizontal and vertical components H and V of the tension at the upper end of the initial profile have been determined from the implicit equations

$$\text{span} = \frac{HL_o}{EA} + \frac{HL_o}{W}\left[\sinh^{-1}\left(\frac{V}{H}\right) - \sinh^{-1}\left(\frac{V-W}{H}\right)\right]$$

$$h = \frac{WL_o}{EA}\left(\frac{V}{W} - \frac{1}{2}\right) + \frac{HL_o}{W}\left[\left(1 + \left(\frac{V}{H}\right)^2\right)^{1/2} - \left(1 + \left(\frac{V-W}{H}\right)^2\right)^{1/2}\right]$$

(For calculation purposes the inverse hyperbolic sine has been replaced by its logarithmic representation, namely,

$$\sin^{-1}h = \ln\left(x + (1 + x^2)^{1/2}\right)$$

The functional constraints which have to be satisfied in the design solution are:

$$F = \sqrt{(H_u + H_1)^2 + (V_{u2} + V_{12} + G)^2}$$

where G is the vertical component of the mass of the link acting at its neutral axis and

$$V_{u2} = V_u - mgl$$
$$V_{12} = V_1 - mgl$$

where mgl is the self-weight of the cable from the origin of the upper support to the horizontal tangent of the catenary and the subscript refers to the lower cable support.

$$T_u = \sqrt{H_u^2 + V_u^2}$$

$$T_1 = \sqrt{H_1^2 + V_1^2}$$

where T represents the maximum cable tension acting at the upper support.

Regional constraints, which have to be satisfied in the design solution due to the fixed geometry of the upper support of the cables are:

Horizontal restraint

$$Dx = \text{span } 1 + Dx_1 - \text{span } u + Dx_u$$

Vertical restraint

$$Dy = h_u + Dy_u - h_1 - Dy_1$$

where Dx and Dy represent the specific location dimensions of the pendants at the boom point box.

5.4 Computer Solution

Every complex design problem has its own unique characteristic that make it a special challenge to traditional optimisation algorithms. It was therefore necessary to try a number of commercial software programmes to determine the best algorithm to suit this particular problem. Theoretically, the minimisation can take place along any defined decent vector. In this optimisation, the method of steepest descent was found to be practical.

The software programme "Eureka", availab from Borland [5], minimises functions by the steepest descent method. The solution path goes from the initial point to the constraint manifold and then lies on the constraint manifold until convergence is achieved. The manifold of points has a Riemannian metric, which is derived from the Hessian of the function being minimised. The constraints are a enforced by means of a user defined penalty function.

5.5 Convergency Criteria

A number of different convergency criteria may be used to terminate the iterative process. In the structure under investigation the method used is to evaluate each side of the solved equation individually and then compare the two sides; the difference between the two results should ideally be 0. However, in practice, the most accurate criteria on which to judge when the problem had converged to a sufficient degree of accuracy was when the changes in displacement of the horizon and vertical regional constraints were at a minimum. Fo. purpose of design, sufficient accuracy was assumed to be ach. , when the difference in equilibrium was reduced to between 0,01 percent and 0,1 percent.

5.6 Summary of Iterative Procedures

The main steps in the iterative process in determining the magnitude of pendant tensions of specified lengths to achieve structural equilibrium may be summarised as follows.

First before the start of the iteration scheme:

(a) Specify the intial problem constraints

- Friction Coeff
- Total Force on Link (Specific Load Case)
- Pendant lengths as determined by supplier.

(b) For the purposes of initialisation, estimate each pendant load and individually calculate the two components H and V of the tension at the upper end of the profile from the implicit equations.

$$\text{span} = \frac{HL_o}{EA} + \frac{HL_o}{W}\left[\sinh^{-1}\left(\frac{V}{H}\right) - \sinh^{-1}\left(\frac{V-W}{H}\right)\right]$$

$$h = \frac{WL_o}{EA}\left(\frac{V}{W} - \frac{1}{2}\right) + \frac{HL_o}{W}\left[\left(1 + \left(\frac{V}{H}\right)^2\right)^{1/2} - \left(1 + \left(\frac{V-W}{H}\right)^2\right)^{1/2}\right]$$

(c) Enter the derived H_u, H_1, V_u V_1 as initialisations.

(d) The nominal vertical distances between the pendant attachments h_u h_1 are entered as specific constraints.

The steps in the iterative procedure are then as follows

Step 1 Solve for the total system reactions and check if the problem has converged. If the displacement in the regional constraints is less than 0,1 percent stop the calculations and print results if not, proceed to Step 2.

Step 2 Update the initialisation of H_u, H_1, V_u V_1 as well as the individual pendant tensions from the calculated results of the previous calculation. Change the nominal vertical distances h_u h_1 from constraints to initialisations.

Step 3 Repeat the above iteration by returning to Step 1.

CHAPTER 6

6. RESULTS AND DISCUSSION

6.1 Intermediate Suspension Pendants

6.1.1 History

In 1977, soon after the introduction of Bucyrus Erie 1570 W draglines into the local coal mining industry, problems developed in the terminations of intermediate suspension pendants. The original manufacturers design of the termination failed from bending fatigue due to a "dogs-leg" design. As a result of this the design was changed to a system utilising straight bolts. Unfortunately these too suffered from bending fatigue problems in spite of improved materials. During 1985/86 an investigation [6] into these repeated failures came to the conclusion that significant bending stresses were present in the terminations, due to imperfect seating of the nuts of the preloading bolts on the socket bridge piece. As a result of this investigation, the termination design was refined to include parallel machined bearing faces and also ensured that the bolt holes were normal to the bearing faces.

This revised termination greatly reduced the incidence of failures of the terminations of intermediate suspension pendants, but has not in all cases produced an adequate service life. The same investigation into preloading bolt failures also identified axial forces in the bolts greater than the design loads. This was, however, not pursued and the source of the overloading and its consequent maldistribution of loads allowed to continue.

6.1.2 Pattern of Load Variation

Typical chart records for the inner and outer bolts of the left hand lower intermediate are shown in Figure 6.1. These measurements were obtained from the investigation into the failure of the preload bolts referred to under Section 6.1.1 on a second machine. They are however believed to be representative of the pattern of load variations present during normal machine operation.

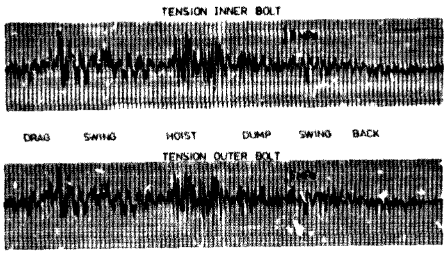


Figure 6.1 Pattern of Lower Intermediate Pendant Load Variations (Fry (6))

They illustrate that during a swing cycle the load variations are fairly repeatable. During digging, vibration of the boom (induced either by impact as the bucket is released or retrieved, or by drag load variations) causes oscillations in rope load about the mean static tension level. Average pendant load increases as the bucket is hoisted and the dragline swings, maximum loads being achieved during the simultaneous swing/hoist cycle. Generally the maximum rope load occurring for any cycle will correspond to the peak level reached during the boom induced oscillations towards the end of the hoist cycle.

Additional load components are induced as a result of lateral vibration of the ropes at a frequency of about 2,5 Hz. Mean axial load in the rope is significantly affected by this vibration.

6.1.5 Load Measurements of Intermediate Suspension Pendants

Measurements of static rope loads were undertaken with a view to establishing the influence of intermediate static pretension and its interaction on the redundant pendant system.

The manufacturer's computer analysis used in sizing the boom chords and suspension pendants, only utilises a single member to simulate the two right-hand upper main pendants and a single member to simulate the left hand pair. Therefore this approach does not allow for the calculation of a differential load in direct comparison with the measured values. With the simple pin connected design no consideration was given to any differential in loading occurring in the programme output.

Recognising these deficiencies the manufacturer's design analysis produced the following results for direct comparison with measured values from the Kleinkopje dragline. The measurement technique utilised both strain gauged sockets and pressure readings from the intermediate pendant prestressing equipment, as discussed in Section 4.2.

Load Case 1

Pendant Location	Calculated Loading (kN)	Measured Loading (kN)	Variation (kN)
Upper Mains	260	335 (averaged)	+75,0
Lower Intermediate	285	285 (preload)	–
Upper Intermediate	[illegible]	260 (preload)	–

Load Case 2

Pendant Location	Calculated Loading (kN)	Measured Loading (kN)	Variation (kN)
Upper Mains	536	586 (averaged)	+50,0
Lower Intermediate	327	320	-7,0
Upper Intermediate	292	285	-7,0

Table 6.1 Comparison between the dragline manufacturer's calculated loads and the loads for the upper main suspension as measured on the Kleinkopje dragline.

6.1.4 Effect of Variations in Pendant Static Preloads

To analyse and assess the accuracy of the pendant load measuring for the specific variations in setting initial tension, as applied to the intermediate pendants of the Kleinkopje dragline, and their consequential effects on the upper main pendants required the following analysis.

Consider a diagram of the boom supported at its pin, lower intermediates, upper intermediates and upper mains, as illustrated in Figure 6.2. The sum of the products of each pendant load and its moment arm about the boom foot pin must remain constant for each of the distinct load conditions (bucket on ground, empty bucket and full bucket). Therefore the effects of variations in intermediate pendant preloads can be assessed against those calculated for individual load cases at their specified preloads.

Effective strand tensions in respect of the fleet angle of the pendants from the plane, perpendicular to the boom feet, are as follows for the Kleinkopje dragline geometry. Lower intermediate (8,285 deg) is equivalent to 98,956 percent of measured tension effective in resisting front end moment, upper intermediate (5,575 deg) is equivalent to 99,527 percent and the upper mains at 0 deg are 100 percent effective. The calculated moment arms of the pendant lines about the boom foot pins are: lower intermediate 32,004 m, upper intermediate 42,266 m and upper main 40,465 m.

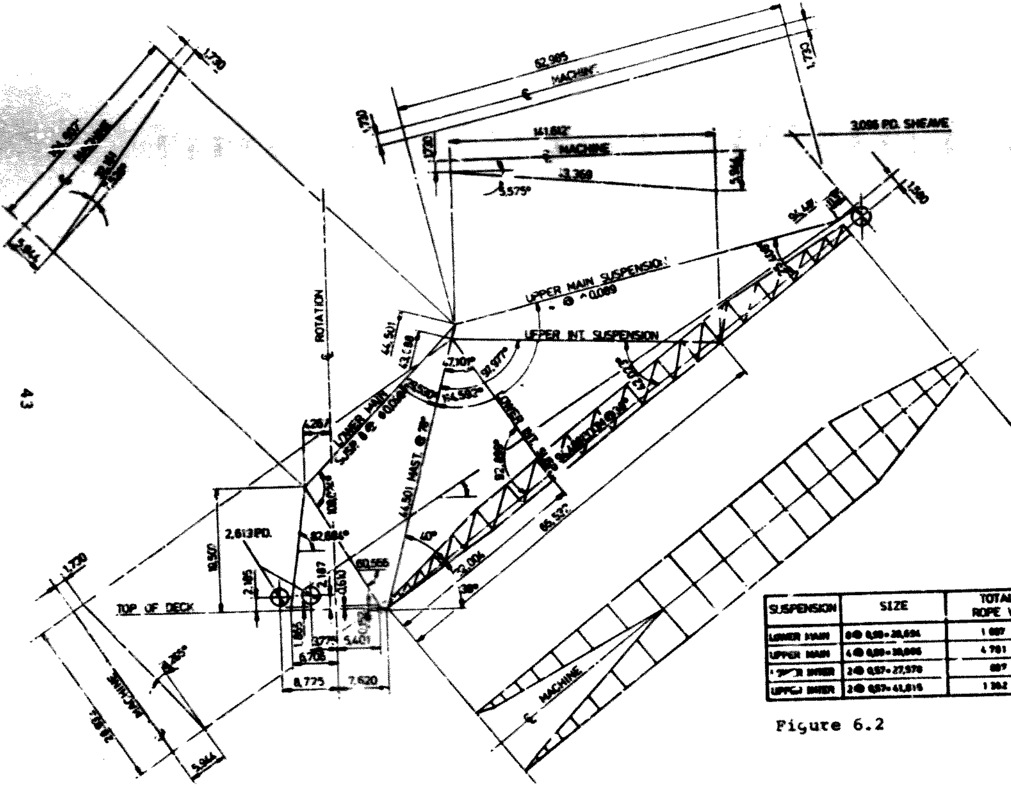
In this analysis the reference moment about the boom foot pins has been determined from the pendant tensions, derived from the dragline manufacturers computer model of the boom. For example, considering load case 1.

Pendant Location	No. of Pendants	Design Loading kN.	Moment Arm m	Percentage Effective	Moment kNm
Upper Main	4	* 260	* 40,465	* 1,0	42084
Lower Inter.	2	* 285	* 32,004	* 0,989	18052
Upper Inter.	2	* 260	* 42,266	* 0,995	21874
			Reference Moment		82010

This can be assessed against a moment derived from the pendant tensions established during the field tests. Consider the reduction in preload of the right lower intermediate.

Pendant Location	No. of Pendants	Design Loading kN.	Moment Arm m	Percentage Effective	Moment kNm
Left Top	1	* 255	* 40,465	* 1,0	10319
Left Bottom	1	* 402	* 40,465	* 1,0	16267
Right Top	1	* 225	* 40,465	* 1,0	9105
Right Bottom	1	* 374	* 40,465	8 1,0	15134
Upper Mains					50825
Left	1	* 212	* 32,004	* 0,989	6714
Right	1	* 250	* 32,004	* 0,989	7917
Lower Inter.					14631
Left	1	* 290	* 42,266	* 0,995	12199
Right	1	* 244	* 42,266	* 0,995	10264
Upper Inter.					22463
			Measured Moment		87919

This represents a 93 percent correlation in measured load accuracy with that of the manufacturer's computer model.



SUSPENSION	SIZE	TOTAL ROPE WT.	TOTAL SOCKET WT.
LOWER MAIN	8@ 0,50 × 28,654	1 887	7 180
UPPER MAIN	4@ 0,50 × 38,086	4 781	[illegible]
LOWER INTER	2@ 0,57 × 27,570	[illegible]	740
UPPER INTER	2@ 0,57 × 41,815	1 302	743

Figure 6.2

Measured load accuracy, determined in this manner for both load cases 1 and 2 in respect of the 40 kN reduction to upper and lower intermediate pendants as well as at the specified preloads is listed in Table 6.2.

Load Case 1

Preloading Condition	Reference Moment kNm	Measured Moment kNm	Correlation
As Specified	82010	94230	0,87
Right Lower Reduced	82010	87919	0,93
Right Upper Reduced	82010	86984	0,94

Load Case 2

Preloading Condition	Reference Moment kNm	Measured Moment kNm	Correlation
As Specified	132 036	139 905	0,94
Right Lower Reduced	132 036	133 994	0,99
Right Upper Reduced	132 036	131 410	0,99

Table 6.2 Upper Main Suspension Load Accuracy Analysis

Table 6.3 shows the results from the manufacturers' boom model for a deviation of 7,0 kN from specified preloads in both intermediate pendants. It illustrates the imbalance in maximum loads which can occur, in respect of load case 1, because of errors in setting initial tensions in intermediate pendants.

Decreasing Lower Intermediate Tension (Load Case 1)

Pendant Location	Design Loading (kN)	Calculated Loading (kN)	Variation (kN)
Upper Mains	260	254,4	-5,6
Lower Intermediate	285	277,4 (reduced)	-7,6
Upper Intermediate	260	276,1	+16,1

Decreasing Upper Intermediate Tension (Load Case 1)

Pendant Location	Design Loading (kN)	Calculated Loading (kN)	Variation (kN)
Upper Mains	260	260,36	+0,36
Lower Intermediate	285	294,4	+9,[illegible]
Upper Intermediate	260	252,4 (reduced)	-7,6

Table 6.3 Resultant load share interaction of dragline upper main suspension produced by a variation in each intermediate pendant as determined by a model of boom geometry. ([illegible]aper (10))

6.1.5 Load-deflection Characteristics of Intermediate Pendants

Computations were undertaken on the load-deflection characteristics of the intermediate pendant systems for two basic reasons. The first was to determine the combination of effects which determine the response of the pendant. For example, should there be appreciable sag in the strand following preloading, the relative inextensibility of the metallic strand is manifest and additional tension can be generated by reduction of the span/sag ratio, which does not necessarily depend on changes in strand length. The second requirement was to determine which factors contributed to the observed non-uniform load sharing between left and right-hand sides of the boom suspension system.

The analysis took into account elastic rope stretch, span geometry and difference in elevation of the pendant attachments. It developed the variation in the horizontal distance between pendant attachment points for various pendant loads. The resulting characteristic is plotted in Figure 6.3 for lower intermediate and Figure 6.4 for the upper intermediate pendants. This has been based on the existing 57 mm dia 146 wire bridge strand with an effective modulus of $171,7 \times 10^6$ kPa and nominal lengths of upper intermediate 41,815 m and lower intermediate 27,578 m.

6.1.6 Factors affecting the asymetrical distribution of load in the dragline boom support system

An analysis of the measured pendant tensions in association with the load deflection characteristics has been used to examine some of the factors, which could possibly affect the degree of load sharing between the left and right-hand boom suspension assemblies.

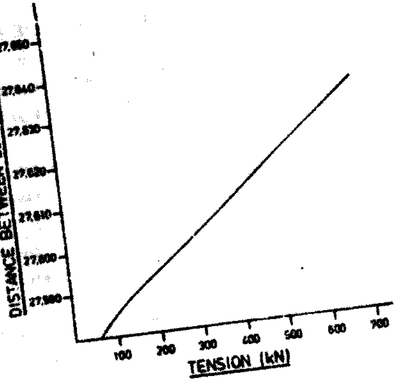


Figure 6.3 Load-Deflection Characteristics of Lower Intermediate Pendants

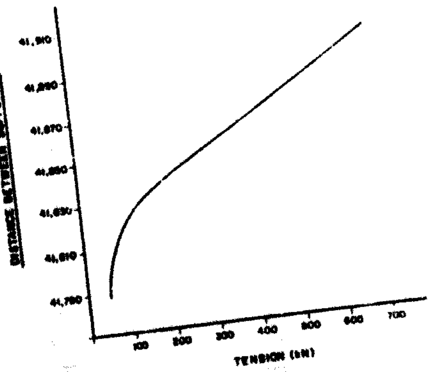


Figure 6.4 Load-Deflection Characteristics of Upper Intermediate Pendants

The asymetrical distribution of load, confirmed by the tests on both the upper main suspension as well as the upper and lower intermediate suspensions, is considered the result of one or more of the following pendant variations.

1. The difference in the distance between pendant attachment points, where the pendant lengths are identical.

2. The effect of discrepancies in the preloading of intermediate pendants.

3. The effect of differences in overall stiffness of the pendants, as a result of some variation in either the geometric and or axial response of the elastic catenary pendant.

4. A difference in unstrained strand lengths, but with the same total distance between attachments. The compensation in strand length being accommodated by the standard take-up incorporated in the pendant preload termination.

The results of the calculations for conditions 2, 3 and 4 with typical assumed variations in the parameters under consideration, are superimposed on Figures 6.5 and 6.6. For both upper and lower intermediate pendants the overall stiffness, following the application of the preload tension, results in a taut flat strand. Under this condition all additional tension is generated solely by the axial extension of the strand, as the sag has been reduced to a very small value, therefore no additional resistance can be obtained by reductions in the catenary profile. In other words the overall stiffness of the pendant is equal to its elastic stiffness. In the example illustrated in Figure 6.6 a reduction in preload of 40 kN is insufficient to reduce the operating point of the load-deflection characteristic into the lower, non-linear regime of the curve. Errors produced in setting preload tensions therefore maintain a constant value throughout the tension range.

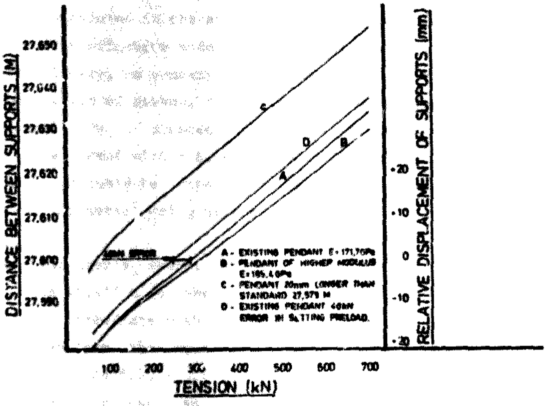


Figure 6.5 **Effects of Errors in setting lower intermediate pendant preload**

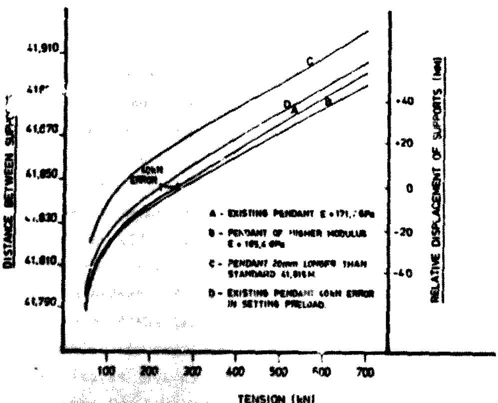


Figure 6.6 **Effects of Errors in setting upper intermediate pendant preload**

A factor considered in the stiffness analysis was the effect of a difference in effective modulus between the pendants. Although this is unlikely, as pendant sets are manufactured from the same original length of strand, collieries' have on occasion replaced one pendant due to component failure. The analysis indicated that, for a strand with a higher modulus of about 8 percent, the maximum loads could be expected to be about 6 percent greater, all other parameters being maintained the same.

The manufacturer's specifications for intermediate pendant lengths are 41.815 m for the upper intermediate and 27,578 m for the lower intermediate with a length tolerance of $\pm$ 0,025 m. As can be seen from the load-deflection characteristic shown in Figures 6.5 and 6.6, a difference of 20 mm in length between a pair of pendants can result in a relative displacement of supports of 8 mm for lower intermediates and 12 mm in the case of the upper strand. This variation in length will have the effect of modifying the respective support moment arms of the boom about the boom foot pin with a consequential adjustment in their product.

The fact that the boom is not torsionally rigid can be expected to alleviate the results of the length variation, with some side effects. Dragline booms are supported in a manner, which attempts to eliminate or at least minimise the effect of deadweight bending. This should be achieved in the dragline by means of the intermediate pendants and the of upper main pendants, which effectively segment the boom into shorter members, theoretically producing straight boom chords, and minimising the effect of deadweight and supporting the bending components. Should the lengths of the intermediate ropes vary, the relationship of tensions in the boom supporting strands becomes complicated due to the effect of boom deflection and resultant change in boom centroid. This can be illustrated by perusal of Table 4.1, which details the effect of a modest change in one intermediate preload, with an equivalent length variation of only 5 mm, resulting in a mass redistribution of load.

It is believed that this mechanism is mainly responsible for the differences in maximum load observed in the left and right pairs of upper main pendants. The other main factor in the discrepancy is due to inadequate intermediate preload application and measurement, with resultant imbalances in the pendants.

6.2 Upper Main Suspension Pendants

6.2.1 History

Perhaps the most critical of all the dragline structures is the tension path supporting the boom. The load is carried from the boom point box to the mast head by two pairs of suspension strands, the upper ends of which are solidly connected to the structure of the boom point box and the lower ends by means of bronze bushed, steel fabricated equalizer links to the mast head. Each pair of strands and their associated link carry one half of the tension path load. Figure 6.7 shows the fabricated link incorporating the bronze bush. During April 1980, a lower suspension pendant link failed at its connection to the A frame after five years of operation. Alarmed by this condition all other links were removed and inspected, and an upper main suspension link was also found to be in danger of imminent failure.

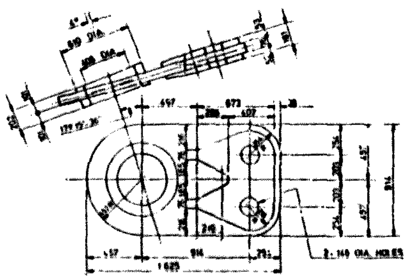


Figure 6.7 Upper Main Suspension Fabricated Link

As a consequence of these failures a design review was undertaken which established that the existing design had an inadequate fatigue life and high operational stress levels. The upper main pendants, their terminations and the links are difficult to inspect "in situ" and removal for inspection is a time consuming operation. Although the manufacturer undertook a partial redesign of the upper main links, as a consequence of the inspection difficulties, it was decided to replace these components at 4-6 year intervals dependent on operational conditions.

The periodic replacement of upper main pendants and their associated links on Bucyrus Erie 1570W draglines are major maintenance items. The purchase cost of the components is by no means negligible, and to this must be added the cost of lost production due to the extension in maintenance time to effect their replacement. Consequently, investigations have been undertaken to extend their operational duty by identifying the factors which cause the abnormally short lives of the components. As mentioned under Section 1.3 the initial work undertaken identified the variations in pendant tension, but was unable to find any correlation between rope length and tension [7]. As a consequence of this inability, it was postulated that the cause of the error in load sharing between the upper and lower pendants connected to the same link was due to the seizure of the equalising link mechanism, which was unable to rotate to a position of equilibrium to compensate for length variations. It was reasoned that at a certain position, dependent on the coefficient of friction in the bearing, the link ceases to rotate with the bottom pendant spanning a greater distance than the top rope.

6.2.2 Upper Main Pendant Load Sharing

The difference in load sharing in upper main pendants, confirmed in this series of tests and influenced by the setting of intermediate pendant preloads, is in principle caused by the same factors introduced in the analysis of intermediate pendants. These are listed as follows:-

1. The difference in the distance between pendant attachment points, where the pendnat lengths are identical.

2. The effect of difference in overall stiffness.

3. The difference in unstrained strand lengths, but with the same total distance between the attachments.

4. An assymetrical distribution of load in either or both upper and lower intermediate suspensions brought about by variations in preload.

5. Inadequate link rotation as a result of seizure induced by a high coefficient of friction.

6.2.3 *Determination of Equilibrium Conditions for Upper Main Suspension System

Preliminary analysis of the measurements of pendant tension obtained from the Kleinkopje dragline test indicated a definite relationship between pendant length and tension. This was in contrast to all previous work, which was unable to determine any such relationship, and as a consequence, postulated that the major cause of the inadequate load sharing was due to the seizure of the link, as a result of an insufficient compensating moment to overcome friction torque.

As a result, the computer analysis detailed under Section 5.3 has been developed to determine the conditions of equilibrium of the total upper main suspension system. To establish the conditions for equilibrium in the system requires the simultaneous determination of the horizontal and vertical components of both the upper and lower pendant catenaries and the results of their reaction on the link producing the following static balance:

$$F * \mu * R = V_u * Dx_u - H_u * Dy_u + V_l * Dx_l + H_l * Dy_l - G * e * \cos\beta$$

Where:

F = Resultant Tension of Link

μ = Coefficient of Friction

R = Radius of Link Pin

V_u & V_l = Vertical components of upper and lower pendants

H_u & H_l = Horizontal components of upper and lower pendants

e = Length of moment arm

β = Angle of rotation of Link

In order to quantify the error in load sharing measured on the Kleinkopje dragline, the computer programme was used to analyse the extent of the rotation of the link under equilibrium conditions in the three load conditions:

- bucket and rigging on the ground
- bucket empty, suspended below boom point box
- bucket full, suspended below boom point box

The actual length of each rope, in this case identical, and the measured tension in the upper pendant are used as programme inputs and the remaining lower pendant tension calculated by the programme. The relationship between maximum pendant load inferred from the computer analysis and maximum pendant load derived from strain measurements is shown in Table 6.4. The geometric nonlinearity of the pendant response is clearly noticeable, as the displacement of the pendant attachment locations is markedly varied by the application of load. This nonlinearity is solely geometric and its origin may be traced to the equation given in Section 5.1, namely that of geometric constraint.

$$\left(\frac{dx}{dp}\right)^2 + \left(\frac{dz}{dp}\right)^2 = 1$$

Description	Load Case		
	1	2	3
	Tension (kN)	Tension (kN)	Tension (kN)
Defined load in upper Pendant	248	471	810
Calcul ted load in lower P[illegible]ant	325	578	924
Link Equilibrium Angle (degrees)	6,67	7,46	7,60

Table 6.4 Suspension system equilibrium conditions for identical length upper main pendants (62,174 m)

The horizontal and vertical components of pendant tension are a function of the strained cable profile and the resultant of these components obviously varies in accordance with applied load. This facet of pendant behaviour can be illustrated by considering the change in the angle of attachment of the pendant, that results between the horizontal and a tangent to the catenary as it approaches the link attachment. Equilibrium position of the system is therefore the resultant position of the link as determined by the horizontal and vertical components of both the upper and lower pendants.

The resultant angle of the link for identical lengths of pendants varies from 6,67 degrees to 7,60 degrees for load cases 1 and 3 respectively. This result is achieved with a difference in positional equilibrium of less than 0,016 mm, thus indicating the accuracy of the calculations. Consideration of the information leads to the conclusion that the resultant angle of pendant tension must always be less than the design condition of a 9,092 degree angle of the chord between the upper and lower main pendant supports, as this is asymptote to the tension curve as illustrated in Figure 6.8.

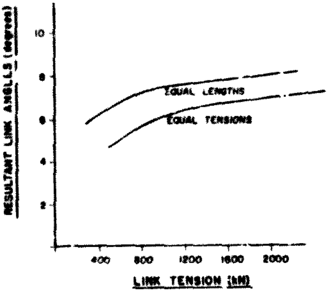


Figure 6.8 Load-Link Equalisation Angle Characteristic for Upper Main Suspension

From the above, it is evident that the differences in pendant tension measured on the Kleinkopje dragline must be due primarily to the differences in distance created by the link as it rotates to a position dictated by pendant tension and catenary. At the design loads a pair of identical length pendants will result in a difference in tension of up to 114 kN. The conclusion is reached that the link design does not fulfil its intended purpose to compensate for variations in pendant length and thus equalise tensions.

6.2.4 Magnitude of Link Error Factor

In order to quantify the magnitude of the error in end point locations, which would result in identical pendant tensions, computations were undertaken using the same basic computer model by means of a revised set of input data. The requirement was to establirh the difference in length between a pair of upper main pendants, which would result in identical tensions under equilibrium conditions.

The total suspended load per side for each load condition, established in the tests on the Kleinkopje dragline, was used to determine the individual, equal, pendant tensions in the programme. The upper pendant of the pair at the manufacturer's design length, as well as the derived coefficient of friction from the first exercise, were selected as the remaining specified inputs. The computer model was then used to determine the lower pendant length commensurate with the specified values.

Table 6.5 shows the results for all load conditions and establishes that the magnitude of error due to link rotation is between 28 mm and 16 mm for load cases 1 and 3 respectively.

Description	Load Case					
	1		2		3	
	Tension kN	Length M	Tension kN	Length M	Tension kN	Length M
Upper Pendant	307	62,174	544	62,174	908	62,174
Lower Pendent	307	62,146	544	62,154	908	62,158
Link Equilibrium Angle (degrees)	5,11		6,27		6,80	

Table 6.5 Difference in Pendant lengths necessary to result in equal pendant tensions under equilibrium conditions

The results show that the difference in length between a pair of pendants, to maintain equal tension, does not remain constant throughout the operating range.

6.2.5 Consideration of Friction Coefficient

It is difficult to define those factors that govern the action of frictional forces, which effect the 1ii mechanism. Several types of friction may play a part and they are described briefly before discussing them in detail.

(a) Dry: static

This type of friction results when the contracting surfaces are free of contaminating films - oxide, moisture, etc.

(b) Boundary: static

This condition of friction results from heavy loadings at slow rotational speed even when a lubricant is present. The surfaces are not in intimate contact, but are separated by a film lubricant.

(c) Semifluid or mixed friction

This kind of friction results when there are incomplete or partial fluid films. The situation develops when the rubbing surfaces are separated partly by viscous films and partly by areas of boundary lubrication.

These factors represent part of the complex picture of surfaces in contact. Fuller [8] in his book, Theory and Practice of Lubrication for Engineers, discussed how specimens that were clean and free of oily contamination, but were exposed to atmospheric air, showed a coefficient of static friction of 0,39. When the specimens were lubricated, the coefficient of static friction fell as low as 0,11. In general the factors that influence dry friction also directly affect boundary friction.

The relationship for dry/boundary friction on the friction force can be considered as follows:

1. Is directly proportional to load.
2. Is independent of the gross area of the contacting surfaces.
3. Depends on the nature of the sliding surfaces.
4. Is influenced by other variables, such as surface roughness, vibration, etc.

Walking draglines are self-propelled machines powered by large onboard motor-generator sets. These units together with other equipment, such as air compressors and ventilation fans, produce vibrational disturbances, even while the dragline is stationary. This is aggravated by vibration of the boom (induced either by impact as the bucket is released or retrieved, or by drag rope variations) and is considered to influence the coefficient of friction. The effect of this can be illustrated by the normal practice of tapping pressure gauges and barometers to decrease the friction of the links and increase the sensitivity and accuracy of the instrument reading.

However, when smooth surfaces are held in tight contact and subjected to slight vibratory motion, some degree of fretting corrosion is possible. This can be minimised by the presence of a lubricant. The bronze on steel configuration of the journal bearing is lubricated by the manual application of grease at some specified maintenance interval and is thus not infallible. Bushing cleanliness also cannot be taken for granted, due to the dusty operating environment of the dragline.

As a result of these factors the estimation of a static friction coefficient, suitable for this application from the typical values of published friction coefficients, is considered unreliable. In this investigation, in order to probe further into the influence of the link on the equilibrium conditions of the pendant assembly, a low value of friction coefficient has been used. This has the effect of minimising the friction force on the link, and in terms of the degree of pendant load sharing, can be considered to be a conservative approach.

The effect of variations in the friction coefficient of the link bushing and their influence in the degree of pendant load sharing is illustrated in Figure 6.9. The analysis was determined by using the computer model in respect of the boom only condition (load case 1) for both equal length and equal tension upper main suspension pendants. It confirms that an increase in the friction coefficient results in an increasing linear response in the range of the pendant load sharing.

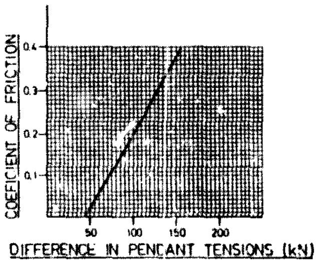


Figure 6.9 Friction - Tension characteristic for the Link Assembly with identical length Upper Main Suspension Pendants for Load Case 1.

The analysis further indicates that a friction coefficient of 0,26 is necessary to result in corresponding values of pendant tension between the calculated tensions, as determined by the computer model, and the tensions, as measured on the Kleinkopje dragline. It is therefore concluded that the type of friction existing between the contact faces of the link bearing, during the duration of the measurements, was generally dry, or possibly with some surfaces in intimate contact, and the remainder separated by an area of boundary lubrication.

7 CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions from Investigation

A number of conclusions and recommendations have been formulated from the study of the dragline boom suspension. Foremost is that substantial imbalances in load sharing occur between the top and bottom pendants on each side of the dragline under static and dynamic load conditions. This is due to the failure of the equalising link to fulfil its intended purpose of equalising pendant tension due to the inherent effects of the basic link geometry.

The search for a solution, which would satisfy the constraints identified in the investigation, but at the same time yield acceptable improvements in dragline design, required consideration of the following goals.

- Limit variations in pendant tension.
- Reduce the effects of variations in the friction coefficient.
- Ease inspection requirements.
- Produce a simple robust design.
- Improve dragline operational availability.
- Reduce costs.

All of the local Bucyrus Erie draglines, and a substantial number in the United States, use the same basic link geometry, both on the upper and lower main suspension systems.

As mentioned in Section 6.2.1, following a failure of a link its periodic replacement with the associated pendants was made mandatory on a 4-6 year cycle. Therefore it is practical to redesign these components, incorporating the improvements as elucidated above.

Because of the non-linear behaviour of the suspension system, simple modifications to the link will not result in an effective length/tension equalisation method. Figure 7.1 and Table 7.1 illustrate the results of one exercise for a modified link under the influence of equal length pendants. Although this is clearly an improvement on the results obtained in the original configuration, see Table 6.3, it does not address the effect of a variation in the friction coefficient. The replacement of the bronze bushing with a rolling element bearing is not considered practical, because of the dragline's inherent vibration. Any alternative low friction bushing material does not have the necessary load carrying capacity.

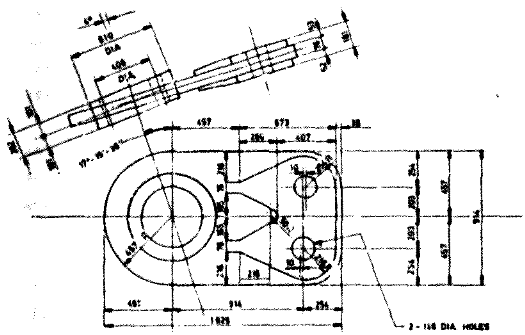


Figure 7.1 Upper Main Suspension Fabricated Modified Link

Description	Load Case		
	1 Tension (kN)	2 Tension (kN)	3 Tension (kN)
Calculated load upper Pendant	295	554	896
Calculated load lower Pendant	319	554	920
Link Equilibrium Angle (degrees)	5,16	6,27	6,64

Table 7.1 Suspension system equilibrium conditions with modified link for identical length upper main pendants (62,174 m)

Present requirements on length tolerance for a set of upper main suspension pendants is 62,176m ± 0,006 measured under a load of 1197 kN. The range in lengths as supplied by the pendant manufacturer for the Kleinkopje dragline differ by only 1mm. Therefore it would seem feasible to seek an improvement in the permitted manufacturing tolerance. If the pendant lengths are supplied in pairs to a tolerance of say ± 2mm, it is only necessary to specify a tolerance between pairs of ± 6mm as the low torsional rigidity of the boom can be expected to alleviate the difference.

The solid attachment of suspension pendants at both ends manufactured to the above specification, will result in an adequate degree of load sharing under all load conditions. The effect of a ± 2mm design tolerance would result in a maximum load imbalance of only 45 kN, an error of less than 4,5 percent at maximum load, as compared to the over 200 kN difference measured with the existing arrangement. It is predicted that by this reduction in the range of pendant tensions, a significant improvement can be expected in the bounce response of the upper main suspension, following the dumping of a full drag bucket. This in turn would result in a dramatic drop in the stress range near the pendant socket, with a consequent improvement in pendant life.

In the event of the pendant manufacturer not being able to meet the recommended length specification on a regular basis, a minor modification to the fixed attachment could alleviate the problem. Pendant lengths are determined prior to installation as a result of the verification exercise undertaken by the manufacturer in the tensioning frame. The verification was described in Section 3.3, Manufacture of Pendants. A locking type of eccentric bush can be designed into the fixed attachment point for one of the pendants to compensate for minor length variations.

A second conclusion reached is the inadequacy of the setting of intermediate pendant preloads. The manufacturer's procedure for the static tensioning of intermediate preloads requires the simultaneous operation of four sets of independant hydraulic tensioning jacks, using pressure gauges with non-direct reading scales. Small imbalances in the setting of these critical preloads for both left and right and upper and lower pendants can result in significant shifts in the reactions of the components of the boom suspension system.

It is therefore recommended that the jacking cylinders for left and right hand pendants be hydraulically interconnected, so that both pendant tensions can be set simultaneously. The resulting hydraulic pressurising equipment for both the upper and lower intermediate pendants should be so positioned as to evaluate their combined effects.

Finally, it is considered that the stiffness of the main and intermediate suspension pendants should be matched, so that increases in load are proportionally shared between them, so that the reactions due to the centrifugal forces are correctly distributed.

7.2 Recommendations for Future Investigation

The pressure gauges provided for the tensioning procedure are considered incapable of being used to set the required preloads to a sufficient degree of accuracy due to a number of factors. Some confusion results from the necessity to double the pressure gauge reading to obtain the pendant preload. Furthermore, in the event of overpressurisation, the gauge reading was found inaccurate due to seizure resulting from friction of the preloading bolts on the socket bridge piece. It is recommended that a strain gauge method be developed for determining and monitoring pendant loads during tensioning.

In view of the importance of correct tensioning of intermediate pendant preloads, the effect of variations in pendant length and their significance should be determined. The present specification calls for the pendant to be manufactured to a tolerance of ± 25mm and as discussed in Section 6.1.6, due to the low level of torsional rigidity of the boom, the consequences of the liberal tolerance should be examined.

Finally it is recommended that the conclusions reached in this investigation and the data obtained should be used to refine the boom finite model and their consequential effect on structural integrity analysed.

APPENDIX A

SPECIFICATION OF INSTRUMENTS USED

SPECIFICATION OF INSTRUMENTS USED

1. Portable Strain Indicator Model P-3500

Manufacturer	MEASUREMENTS GROUP, INC. USA
Range	$\pm$ 19 999 $\mu\epsilon$
Accuracy	$\pm$ 3 $\mu\epsilon$
Sensitivity	$\pm$ 1 $\mu\epsilon$
Bridge Excitation	2,0 Vdc
Calibration	Shunt calibration across 120 Ω and 350 Ω dummy gauges to simulate 5000 $\mu\epsilon$
Analog Output	Linear $\pm$ 2,50 V max
Power Supply	Internal battery pack
Size and Weight	228 * 152 * 152 mm 2,9 Kg

Description

The model P-3500 Strain Indicator is a portable battery powered instrument for use in stress analysis testing and for use with strain-gauged based transducers. It incorporates a highly stable DC amplifier regulated bridge excitation supply and settable gauge factor controls.

Static measurements are read directly from the LCD readout with 1 $\mu\epsilon$ resolution. An analog output with a - 3dB bandwidth of 4 kHz is provided to drive an external oscilloscope or recorder for dynamic measurements. The instrumen will accept full-, half-, or quarter- bridge strain gauge inputs.

Bridge excitation is 2 Vdc, resulting in low gauge power and negligible drift due to gauge self-heating. Gauge factor is precisely settable by a front panel 10-turn potentiometer to a resolution of 0,001 unit, and is displayed on the LCD readout. when the gauge factor push button is depressed.

.. Switch and Balance Unit Model SB-10

Manufacturer	MEASUREMENTS GROUP, INC. USA
Circuits	10 Channels
Inputs	Will accept quarter-, half-, or full bridge circuits in any combination
Balance Range	$\pm$ 2000 $\mu\epsilon$
Switching Repeatability	Better than 1 $\mu\epsilon$ for gauge resistance of 120 Ω or higher
Size and Weight	230 * 150 * 150 mm. 2,5 Kg

Description

The SB-10 Switch and Balance Unit provides a means for strain measurement when more than one strain gauge is involved. It is designed for use primarily with the P-3500 Strain Indicator and features gold-plated push/clamp binding posts to allow fast and reliable connections of input circuits. Individual ten-turn locking potentiometers with turns counting dials are used for fine balance adjustment. The channel selector switch has negligible switch resistance.

3. Strain Gauge Type FCA-6-1'

Manufacturer	TOKYO SOKKI KENYUJO CO., LTD JAPAN
Material	Cu-Ni Foil
Temperature Range	0-80°C
Gauge Length	6 mm

Gauge Width	2,7 mm
Base	23 * 21 mm
Nominal Resistance	120 $\pm$ 0,5 Ω
Gauge Factor	2,1

Description

The gauge has a two element 90° "tee" stacked rosette configuration and has been designed to eliminate a moisture proof coating. One metre long lead out wires are attached and the whole area of the gauge and junction of the lead wires overcoated with a transparent flexible epoxy resin of approx. 1mm thickness.

A cyano-acrylate adhesive was used to bond the gauge to the pendant socket.

APPENDIX B

CALIBRATION OF SOCKETS

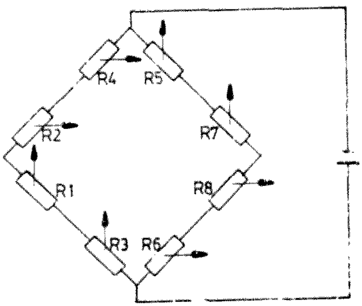
APPENDIX E

CALIBRATION OF SOCKETS

In order to evaluate the degree of load equalisation on dragline upper main suspension pendants and the effect variations in intermediate pendant preloads have on them, the pendant loads were measured by attaching strain gauges to their sockets.

The strain gauges were arranged and attached to measure axial strain and to compensate for bending effects as illustrated in Figure 4.2

Determination of Bridge Factor



$$i_{4213} = \frac{E}{R_4 + R_2 + R_1 + R_3}$$

$$= \frac{E}{2(R + R) + 2(R - 0{,}3\ R)} = \frac{E}{4R + 1{,}4\ R}$$

$$V_{42} = i_{4231}\ R_{42}$$

$$= \frac{E}{4R + 1{,}4\ R}\ 2(R + R) = \frac{2ER + 2E\ R}{4R + 1{,}4\ R}$$

$$i_{5786} = \frac{E}{R_5 + R_7 + R_8 + R_6}$$

$$= \frac{E}{2(R - 0{,}3 \quad R) + 2(R + \quad R)} = \frac{E}{4R + 1{,}4 \quad R}$$

$$V_{57} = i_{5786} \; R_{57}$$

$$= \frac{E}{4R + 1{,}4 \quad R} \; 2(R = 0{,}3 \quad R) = \frac{2ER - 0{,}6E \quad R}{4R + 1{,}4 \quad R}$$

$$V = V_{57} - V_{42}$$

$$= \frac{2ER - 0{,}6E \quad R}{4R + 1{,}4 \quad R} - \frac{2ER + 2E \quad R}{4R + \; {,}4}$$

$$= \frac{2ER - 0{,}6E \quad - 2ER - 2E \quad R}{4R + 1{,}4 \quad R}$$

$$= \pm \frac{2{,}6 \quad R}{4R + 1{,}4 \quad R}$$

For very small values of R
$4R + 1{,}4 \quad R = 2R$

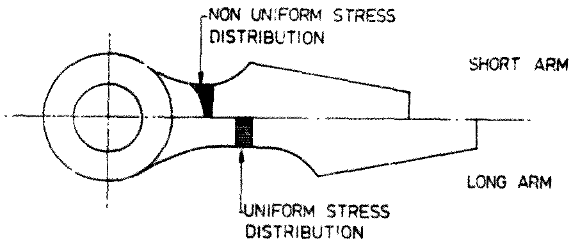
So that $\frac{2{,}6E \quad R}{4R}$

Thus the full bridge connection has the effect of making the bridge sensitivity 2,6 times the sensitivity of a single active gauge.

Single gauge $= \pm \frac{E \quad R}{4R}$

Due to a lack of spare dragline sockets it has not been possible to verify the calculated load. However, in a previous investigation undertaken by D.R. Fry [7] slightly larger sockets by the same manufacturer were calibrated in the Maggie Hand Test rig. This investigation found that there was a significant difference between the load/strain calibration and the load from the stress * area calculation. The calculation over-estimates the load by as much as 74 percent. The design of the sockets is such that the use of nominal stresses to calculate could and obviously does, lead to errors. The author stated that this was due to stress concentration and gradients, as clearly this depends on the geometry and shape of the socket. Designs, which have long lugs between the eyes and the socket basket, will give a closer correlation between calculated and actual load, than designs which have short lugs without a reasonable length of uniform cross section. The figure shows schematic sockets and illustrates why the strain measured could lead to inaccurate over-estimates of load.

The dimension of the lugs used in the calibration exercise are 208 * 77 mm against the 184 * 67 mm of the Kleinkopje sockets. In view of this similarity and the socket geometry, it was decided to factor calculated pendant loads on the basis of the calibration exercise to compensate for tension gradient errors. This resulted in a derived gauge factor of 4,49. It must also be noted that the requirements of pendant tension of this investigation are basically of a comparative nature and thus are considered to be relevant despite any tension calibration irregularities.



KLEINKOPJE BUCYRUS ERIE 1570W DRAGLINE UPPER MAIN SUSPENSION PENDANT LOAD MEASUREMENTS AT THE MANUFACTURERS SPECIFIED INTERMEDIATE PENDANT PRELOADS

Bridge Microstrain

	Socket	Bucket on Ground,	Empty Bucket,	Full Bucket
1.	Left top	250	449	763
2.	Left Bottom	379	622	935
3.	Right Top	214	407	701
4.	Right Bottom	320	551	870

Pendant Load (kN)

		Calc.	Derived	Calc.	Derived	Calc.	Derived
1.	Left Top	499	290	896	519	1522	882
2.	Left Bottom	756	438	1241	719	1865	1082
3.	Right Top	427	248	812	471	1398	810
4.	Right Bottom	638	366	1099	537	1736	1006

Difference in Pendant Tension (kN)

Between 1 & 2	257	148	345	200	343	200
Between 3 & 4	211	118	287	166	338	196

Note

The calculated pendant load is derived from a bridge factor of 2,6 and the derived pendant load is determined from the previous calibration results.

APPENDIX C

EXAMPLE RESULTS OF COMPUTER ANALYSIS INCLUDING ERROR ANA SIS

```
************************************************************************
Eureka: The Solver, Version 1.0
Friday March 31, 1989, 2:59 pm.
Name of input file: A:\RCPESBT1
************************************************************************


;Upper main suspension right hand load case 1
;NOTE       ** Check for model validity **

;Cable equations for elastic catenary

;Lower rope
hl=(24.208*lengthl/698420)*(Vl/24.208-.5)+(Hl*lengthl/24.208)_
*((1+(Vl/Hl)^2)^.5-(1+((Vl-24.208)/Hl)^2)^.5)
spanl=(Hl*lengthl/698420)+((Hl*lengthl/24.208)*(ln(Vl/Hl+((Vl/Hl)_
^2+1)^.5)-ln((Vl-24.208)/Hl+(((Vl-24.208)/Hl)^2+1)^.5)))

;Upper rope
hu=(24.208*lengthu/698420)*(Vu/24.208-.5)+(Hu*lengthu/24.208)_
*((1+(Vu/Hu)^2)^.5-(1+((Vu-24.208)/Hu)^2)^.5)
spanu=(Hu*lengthu/698420)+((Hu*lengthu/24.208)*(ln(Vu/Hu+((Vu/Hu)_
^2+1)^.5)-ln((Vu-24.208)/Hu+(((Vu-24.208)/Hu)^2+1)^.5)))

;Equations for lower link
 F*M*.203=((Vu2*(.9144*cos(E)-.2023*sin(E)))-(Hu*(.9144*sin(E)_
+.2023*cos(E)))+(Vl2*(.9144*cos(E)+.2023*sin(E)))+(Hl*(.2023*cos(E)
-.9144*sin(E)))-(5.0*.6*cos(E)))

;Equations of equalisation
.3997=((hu+(.9144*sin(E)+.2023*cos(E)))-(hl-(.2023*cos(E)-.9144*sin
.064=((spanl+(.9144*cos(E)+.2023*sin(E)))-(spanu+(.9144*cos(E)_
-.2023*sin(E))))

F=((Hu+Hl)^2+(Vu2+Vl2+5.0)^2)^.5
Vu2=Vu-24.208
Vl2=Vl-24.208
Tu=(Hu^2+Vu^2)^.5
Tl=(Hl^2+Vl^2)^.5

;Constraints                ;NOTE Length in metres
                            ;     Load in kilonewtons

lengthu=62.174
lengthl=62.174
F=606.87673
M=.12

;Initialisations
Tu:=248
Tl:=360
spanu:=61.3700
spanl:=61.3935
Hu:=242.6002
```

```
******************************************************************
Eureka: The Solver, Version 1.0                          Page 2
Friday March 31, 1989, 2:59 pm.
Name of input file: A:\ROPESPT1
******************************************************************

Hl:=359.1886
Vu:=51.4684
Vl:=70.3626
hu>9.952
hl>9.952

******************************************************************

Solution:

 Variables     Values

 E           =      .11906626

 F           =      606.87673

 hl          =      9.9540354

 Hl          =      341.09968

 Hu          =      260.68878

 hu          =      9.9519998

 lengthl     =      62.174000

 lengthu     =      62.174000

 M           =      .12000000

 spanl       =      61.390629

 spanu       =      61.374689

 Tl          =      347.70139

 Tu          =      266.30528

 Vl          =      67.433383

 Vl2         =      43.225383

 Vu          =      54.404646

 Vu2         =      30.196646
```

```
*********************************************************************
Eureka: The Solver, Version 1.0                          Page 3
Friday March 31, 1989, 2:59 pm.
Name of input file: A:\ROPESBT1
*********************************************************************


Maximum error is 1.2380434e-06


Evaluation of formulas:

 Formulas            Values

 hl                  =     9.9540354
 (24.208*leng... =     9.9540354
   difference =      .00000000


 spanl               =     61.390629
 (Hl*lengthl/... =     61.390629
   difference =      .00000000


 hu                  =     9.9519998
 (24.208*leng... =     9.9519998
   difference =      .00000000


 spanu               =     61.374689
 (Hu*lengthu/... =     61.374689
   difference =      .00000000


 F*M*.203            =     14.783517
 ((Vu2*(.9144... =     14.783518
   difference (error) = -1.2380434e-06


 .3997               =     .39970000
 ((hu+(.9144*... =     .39969982
   difference (error) = 1.8452512e-07


 .064                =    .064000000
 ((spanl+(.91... =    .064000061
   difference (error) = -6.1084172e-08


 F                   =     606.87673
 ((Hu+Hl)^2+(... =     606.87673
   difference =      .00000000
```

```
Vu2               =     30.196646
Vu-24.208         =     30.196646
  difference =       .00000000

V12               =     43.225383
V1-24.208         =     43.225383
  difference =       .00000000

Tu                =     266.30528
(Hu^2+Vu^2)^... =     266.30528
  difference =       .00000000

Tl                =     347.70139
(Hl^2+Vl^2)^... =     347.70139
  difference =       .00000000

lengthu           =     62.174000
62.174            =     62.174000
  difference =       .00000000

lengthl           =     62.174000
62.174            =     62.174000
  difference =       .00000000

F                 =     606.87673
606.87673         =     606.87673
  difference =       .00000000

M                 =     .12000000
.12               =     .12000000
  difference =       .00000000

Tu                =     266.30528
Tl                =     347.70139
spanu             =     61.374689
spanl             =     61.390629
Hu                =     260.68878
Hl                =     341.09968
Vu                =     54.404646
```

```
****************************************************************
Eureka: The Solver, Version 1.0                          Page 5
Friday March 31, 1989, 2:59 pm.
Name of input file: A:\ROPESBT1
****************************************************************

 V1              =     67.433383

 hu              =     9.9519998
 9.952           =     9.9520000
   difference (error) = -1.8489033e-07


 hl              =     9.9540354
 9.952           =     9.9520000
   difference =    .0020354247


Maximum error is 1.2380434e-06

****************************************************************
```

REFERENCES

REFERENCES

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ADDENDUM TO THE PROJECT REPORT

BUCYRUS ERIE 1570W DRAGLINE -
ANALYSIS OF THE UPPER MAIN
SUSPENSION SYSTEM.

SUBMITTED IN RESPONSE TO THE
EXTERNAL EXAMINER'S REPORT.

N. BROWN

INTRODUCTION

Investigations of this nature, are forced to progress of necessity by prevailing circumstances. Seldom is it possible to develop a test procedure in a totally planned and logical sequence, where, for instance, the calibration of the output of each strain gauge bridge can be undertaken following its manufacture.

Because of their extreme size, draglines present unique problems in respect of access to components. Installation of strain gauges on the pendants must be co-ordinated with many other repair procedures when the boom is on ground level. Gauges must be well protected and leads strung to an accessible point prior to the raising of the boom. A further consideration in dragline component testing is machine availibity. Draglines are primary overburden stripping machines and any test represents lost production for the colliery, with every attempt being made to minimize down time.

Draglines are also not located in a laboratory, and all test equipment and personnel must travel to the site in order to perform the test. These considerations combined, produce expensive one time test data.

Perhaps a few statistics can amplify this position. Walking draglines are the largest, land based, mobile machines ever built. The Bucyrus Erie 1570 W Dragline weighs 3 600 t and represents a capital cost of R 70 000 000. It operates on a 24 hour day, 7 day week basis and stops on average for only four days in the year.

Maintenance is undertaken on a fortnightly cycle usually of 10-12 hours. During these periods bucket and running rope replacement (bucket hoist and drag ropes) is often undertaken. Each suspension pendant costs R 90 000 and suspension link R 85 000, these are replaced at periods not normally exceeding five years. To replace these components takes 10 days, necessitating the lowering of the boom and mast at a total cost of over R 2 000 000.

GENERAL TESTING PROCEDURE RESTRAINTS.

Although Amcoal at the time of the investigation operated five 1570W model draglines, each suspension pendant is unique. This is due to variations in the length of boom and its operating angle and as a consequence, group spare availability is not practical. As it is not possible to forecast exactly the life of these components, they are manufactured at a convenient time ensuring delivery to provide insurance cover for the remaining life of the operating pendants. Due to the cost of these components the timing of the order for the replacement set is variable and is dependent on an assessment of the condition of the operating pendants. The Kleinkopje pendants were available on the colliery six months before their installation and were ordered nine months before delivery. No requirement for a strain gauge analysis was foreseen at this early stage, therefore calibration following manufacture was never considered.

At the time of the investigation the local manufacture of pendants was still in its infancy. Similarly, the local manufacture of sockets was in progress. As no sockets were available at the time, the importation of a set of sockets was necessary to meet the required delivery. These sockets were supplied by a different manufacturer and differed in physical dimensions from those previously installed on group draglines. Sockets from discarded pendants from both Bethlehem Steel and US Steel were available, but following Amcoal's policy, the use of secondhand sockets was not permissible. As all the new Esco sockets were used on the Kleinkopje installation, no calibration exercise following this route was possible.

The examiner has suggested post discard calibration of the pendants, but as these components are still in service and will remain so for a number of years this alternative is not practical to achieve results under normal operational circumstances.

Access to the pendant sockets for the installation of strain gauges was only obtained following their installation on the lowered boom. It was then possible to grind the cast surfaces of the sockets to a suitable finish at the location of the strain gauges. The cross sectional dimensions of the lugs were measured to establish the centre lines for the correct location of the gauges. Thereafter, the gauges were carefully bonded to these established datums. Cross-sectional areas did not vary by more 1 percent in the lugs fitted with strain gauges.

CORRELATION OF ANALYTICAL STRESS WITH EXPERIMENTAL TEST RESULTS.

A requirement to establish academic levels of accuracy on the conversion of measured strain to force is considered inappropriate. This is commensurate with the level of accuracy claimed in the report of within 10-15 percent of absolute values.

The repeatability of measurements taken on mobile equipment operating on uneven blasted surfaces is to some extent uncertain and under these conditions an accuracy to within 10 per cent of absolute values can be considered to be good.

Practical methods have the ability to establish pendant loadings which can be used to assess engineering structures and produce meaningful improvements commensurate with the available time and cost.

Preliminary work on this investigation involved the determination of the resultant moment about the boom foot pins for the number of loading options. Following the determination of the measured strain, it was immediately obvious that pendant tension derived from a stress x area calculation was significantly inaccurate, when compared with the relevant resultant moment calculation. This can be illustrated as follows:

Position the bucket on the ground with the spreader bar and dump block weight off the hoist ropes. The bucket should be positioned directly below the boom point sheaves equivalent to maximum reach. Record a set a strain gauge readings. This position is designated as load case 1.

A further set of strain measurements is established with the empty bucket hoisted at maximum reach to a position just below the hoist trip-out point. This position represents load case 2.

The differences in calculated pendant tensions between load case 1 and load case 2 at the stress concentration factor used in the report produced the following resultant moment about the boom-foot pins.

PENDANT LOCATION	DIFFERENCE (kN)	MOMENT ARM (m)	MOMENT (kNm)
Left Top	229	40,465	9 266
Left Bottom	281	40,465	11 371
Right Top	223	40,465	9 024
Right Bottom	271	40,465	10 996
	Upper Main Pendant Sub Total		40 627
Left	60	42,266 x 0,995	2 492
Right	60	42,466 x 0,995	2 492
	Upper Intermediate Sub Total		4 984
Left	0	32,004 x 0,989	-
Right	0	32,004 x 0,989	-
	Lower Intermediate Sub Total		-
		Total	45 611

This resultant can now be compared with that due to the empty bucket.

Mass of fully rigged bucket as equipped during the test sequence	72 859 Kg
Less the reduction due to hoist rope (43,56Kg/m)	(5 477)Kg
	67 382 Kg

Moment Arm of Load	75,68 m

Resultant Moment due to bucket load

$$67\ 382 \times 75{,}68 \times 9{,}81 = 50\ 025\text{kNm}$$

Effective correlation $\dfrac{45\ 611}{50\ 025} = 0{,}91$

If this moment calcuation had been used as a method of calibrating the strain/force relationship for the tested socket (on the assumption that each socket behaved in the same way) then the stress concentraction factor would have been 1,72 x 0,91 = 1,57.

This is one of the many comparative exercise undertaken to establish overall accuracy. Further examples are listed in Section 6.1.4 based on information obtained from the dragline manufacturer's engineering department.

WIRE ROPE/STRAND PROPERTIES.

The examiner appears to put undue emphasis on the torque behaviour of bridge strand rope. These, as the name implies, are commonly used in suspension application such as suspension bridges, track ropes on aerial cableways and guide ropes in mine shafts were any appreciable amount of spin is undesirable.

Upper main suspension pendants are manufactured from a single strand consisting of 243 wires laid up (51/45/39/33/27/21/15/9/3). The single strand is constructed of successive layers of wire left and right hand lay with the objective of minimising the tendency of the strand to produce any appreciable torque under axial loadings. The term right-hand lay in the specification refers by convention to the lay of the outermost wires.

Little technical information has been published on the torque generating characteristics of bridge strand rope in comparison with the axial force involved. However, wire rope manufacturers consider it to be insignificant. Haggie Steel Ropes, consider the following torque factors to apply.

Rope Construction	Torque Factor in mm mm of Rope Diameter
Triangular Strand	0,165
Langs Lay	0,157
Multi-Strand Non Spin	0,019
Bridge Strand	0,006213

It is difficult for me to see how the examiner's comments on a combination of right hand and left hand lay on each pair of upper suspension ropes could have any validity as the torque transmitted by bridge strand ropes is negligible.

Further as the intermediate pendants are also manufactured from bridge strand rope his comment in this regard would also appear to have no substance.

THEORETICAL ANALYSIS.

I fear that because the examiner had "neither the time nor the inclination" to verify the simulation of the upper main pendant system by means of the theoretical model, he also ignored the major contribution to the report.

This model was used to derive the source of the poor load sharing displayed by the upper main suspension pendants. A theoretical model by its nature does not require that absolute values of tension be determined to guarantee its validity.

CONCLUSIONS AND RECOMMENDATIONS.

Attached [illegible] similar [illegible] of the standard link shown in Figure 6.7 and the modified link illustrated in Figure 7.1. In the modified link the pendant socket attachment locations are offset about the original centre line by 10mm.

Finally, some clarification on the recommendation on the solid attachment of suspension pendants. The effect of a fixed location of pendants has been simulated on one dragline by hydraulically jacking the "equalising" link to a position where the pendant tensions were equalised with the bucket on the ground. Following the application of load the pendants behaved linearly and the incremental loads were equally shared between the pendants.

The advantages to be gained by improving the specified length tolerance and fixing the location of upper main suspension pendants are considerable. A component, which has caused problems over the years [illegible] to be replaced every [illegible] years, can be eliminated. [illegible] life can be expected due to reduced stress range. [illegible]ngs of R 750 000 every five years, by obviating the need for the replacement of "equalising" links, would be attractive under any circumstances.

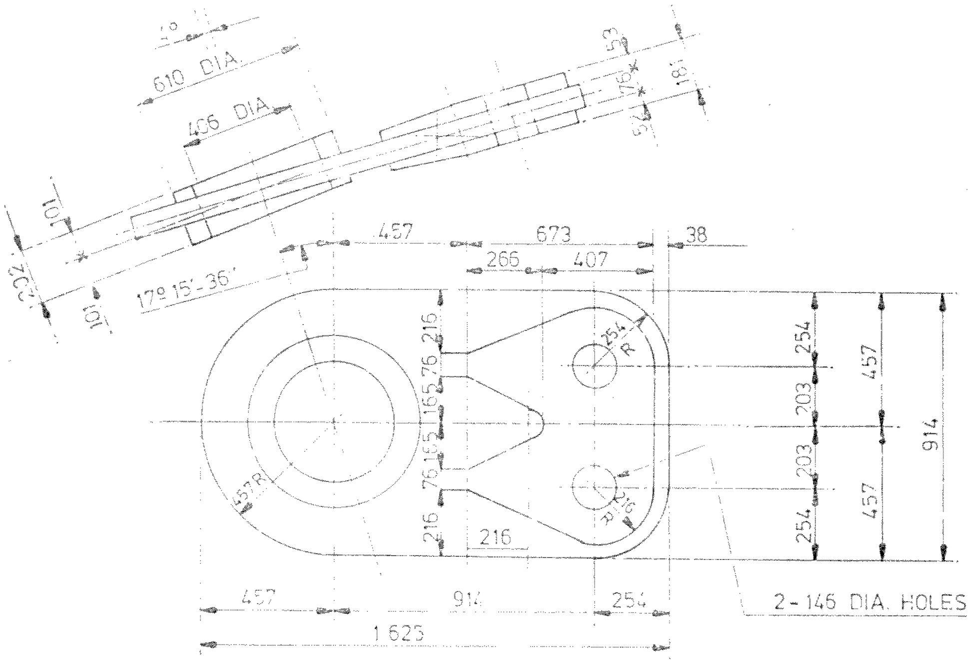

Figure 6.7 Upper Main Suspension Fabricated Link

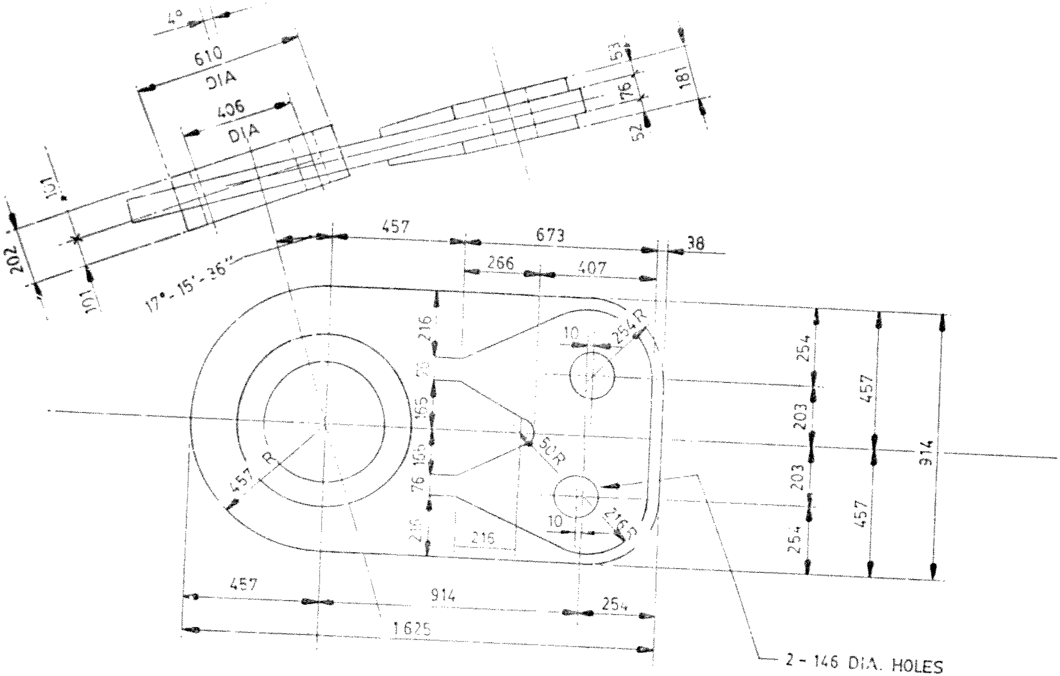

Figure 7.1 Upper Main Suspension Modified Link

outer wire diameters (see 5.2.1) the increase in load obtained from the formula should be increased by 50%.

The additional load induced in a 6x30(12/12/6Δ)/F rope due to bending is shown (fig. 23) as a percentage of the breaking force for various ratios of sheave to rope diameter.

4.5 PERCENTAGE RESERVE STRENGTH

Unless affected by corrosion, the inner wires of a rope are usually intact after many outer wires have broken through wear or fatigue. The ratio of metallic area of inner wires only to total metallic area of the whole rope, multiplied by 100 gives the percentage reserve strength.

Although a somewhat theoretical figure, it is of interest when choosing the best construction for an application where rope safety and economy are both of paramount importance.

The following table gives percentage reserve strengths for commonly used rope constructions.

Rope Construction	Percentage Reserve Strength
6x7(6/1)/F	14%
6x14(8/6Δ)/F	19%
6x28(10/12/6Δ)/F	27%
6x19(9/9/1)/F	31%
6x25(12/6F+6/1)/F	42%
18 Strand Non-spin 12x7(6/1)/6x7(6/1)/F	43%
6x36(14/7+7/7/1)/F	48%
14 Strand Non-spin 8x6/6x10(7/3Δ)/WMC	50%
15 Strand Non-spin 9x8/6x29(11/12/6Δ)/WMC	59%

4.6 TORQUE IN ROPES

When a rope is loaded under tension a torque is produced which will cause the rope end to tend to rotate. Six strand Lang's lay ropes must be operated with both ends restrained from rotating otherwise the rope will unlay and become unstable.

Non-spin ropes are designed to adjust to this torque and reach a balance so that this type of rope can be used with ends free to rotate. The amount of readjustment necessary for a non-spin rope is dependent on its design, and Haggie Rand non-spin sinking ropes are specifically designed to minimise turn due to a change in load. This makes them particularly suitable for deep shaft sinking operations.

The torque generated when loading a rope in the as manufactured condition can be estimated from the following formula:

$$T = C\,d\,P$$

Where

T = Torque (newton metres)
P = Tensile load on rope (newtons)
C = Torque factor (mm per mm of rope diameter)
d = Rope diameter (metres)

Representative estimates of the torque factor for ropes in the as manufactured condition are given below.

Rope Construction	Torque Factor in mm per mm of rope diameter
6x19(9/9/1)/F Lang's Lay	0,157
6x19(9/9/1)/F Ordinary Lay	0,098
6x25(12/6F+6/1)/F Lang's Lay	0,143
6x25(12/6F+6/1)/F Ordinary Lay	0,089
6x30(12/12/6Δ)/F Lang's Lay	0,165
15 Strand Non-spin 9x8/6x29(11/12/6Δ)/WMC	0,019
18 Strand "Fishback" 12x10 (8/2)/6x29(11/12/6Δ)/WMC	0,016
15 Strand "Fishback" 9x10(8/2)/6x14(8x6Δ)/WMC	0,062