STRUCTURAL FAILURES ON MOBILE MATERIALS HANDLING EQUIPMENT

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DISSERTATION SUMMARY

STRUCTURAL FAILURES ON MOBILE MATERIALS HANDLING EQUIPMENT

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Degree:	Master of Science (Structures)

Bulk materials handling systems are extensively used within the mining and minerals industry. Due to the nature of the mining environment, the support structures for these systems are often exposed to special and/or accidental loading conditions. This unfortunately leads to a fairly high incidence of structural damage or failure being experienced within the mining industry, notwithstanding design compliance with appropriate standards. Over the past few decades reputable mining companies have acknowledged the necessity for more conservative structural designs and this has led to the development of design rules for permanent structures which are used in conjunction with national and international design standards. The design of mobile continuous bulk handling equipment is governed internationally by the ISO 5049-1 (1994) Standard, except in Australia where AS 4324-1 (1995) is generally utilised.

The study investigates a number of catastrophic failures of mobile bulk materials handling (BMH) equipment to identify the typical root causes and their complex interaction in these disastrous events. A retrospective view is taken of the processes followed during the investigation of the main case study to develop a methodology for future failure investigations.

Brief case studies are made to demonstrate the shortcomings of the ISO 5049-1 (1994) Standard, which currently provides no rules or guidelines for machine protection systems. The aim of the study is ultimately to improve structural safety on future mobile BMH equipment designs, which does not necessarily imply a more conservative design approach, but rather that design loads and conditions be correctly assessed. The revision of ISO 5049-1 (1994) is subsequently proposed to provide specific rules and guidelines pertaining to machine protection systems. Other focus areas for consideration are also covered. It is furthermore recommended that the structural design engineer should play a more prominent role during the final acceptance of mobile BMH equipment and handover to the owner. A systems design approach integrating the respective engineering disciplines and based on a comprehensive risk assessment, is required.

SAMEVATTING VAN VERHANDELING

STRUKTURELE FALINGS BY MOBIELE MATERIAALHANTERINGSTOERUSTING

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Materiaalhanteringstelsels word op groot skaal in die mynbou en mineraal bedryf gebruik. Die ondersteuningstrukture van die genoemde stelsels word soms aan buitengewone belastings blootgestel, wat tiperend van die mynbou omgewing is. Ongelukkig kom strukturele falings redelik gereeld voor desnieteenstaande dat daar gewoonlik aan die betrokke ontwerpstandaarde voldoen word. Gedurende die laaste paar dekades het gevestigde myngroepe die behoefte vir strenger ontwerpsreëls geïdentifiseer, wat daartoe gelei het dat maatskappy spesifikasies ontwikkel is wat in oorleg met nasionale en internasionale ontwerpstandaarde gebruik word. Die ontwerp van mobiele materiaalhanteringstoerusting word egter deur die ISO 5049-1 (1994) Standaard gereguleer, behalwe in Australië waar die AS 4324-1 (1995) Standaard gewoonlik geld.
Die studie ondersoek ’n aantal voorvalle waar toerusting in duie gestort het, of erg beskadig is, om die leser se aandag te vestig op die identifikasie van die oorspronge van tiperende falings en hulle komplekse wisselwerking in die meeste rampspoedige strukturele insidente. Falings-ondersoek metodiek vir mobiele materiaalhanteringstoerusting word ontwikkel deur ’n terugskouing te neem van die prosesse wat gedurende die ondersoek van die hoof gevallestudie voltooi is.

Kort gevallestudies word gebruik om tekortkominge van die ISO 5049-1 (1994) Standaard te demonstreer. Laasgenoemde standaard bied tans geen vereistes ten opsigte van beskermingstelsels van toerusting nie. Die doel van die studie is uiteraard om strukturele veiligheid van mobiele materiaalhanteringstoerusting te bevorder, wat nie noodwendig ’n meer konserwatiewe ontwerpsbenadering vereis nie, maar eerder dat ontwerpsbelasting en toestande verstaan en korrek benader word.

Die hersiening van die ISO 5049-1 (1994) Standaard word voorgestel om voorsiening te maak vir reëls en riglyne ten opsigte van beskermingstelsels vir mobiele materiaalhanteringstoerusting. Addisionele fokus areas vir oorweging word ook aangespreek. Dit word aanbeveel dat die strukturele ingenieur voortaan ’n meer omvattende rol sal speel met die finale goedkeuring en gevolglike oorhandiging aan die eindverbruiker. Aandag word daarop gevestig dat toekomstige ontwerpe beter, oor die relevante ingenieurs-dissiplines heen, geïntegreerd moet wees, terwyl ’n ontwerpsbenadering gevolg moet word wat deur risikobepaling gedefinieer is.

ABSTRACT

Title: Structural Failures on Mobile Materials Handling Equipment

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Bulk materials handling (BMH) systems are extensively used within the mining and minerals industry where a fairly high incidence of structural failure is experienced, notwithstanding design compliance with appropriate standards. The study explores a number of catastrophic failures of mobile BMH equipment to identify the typical root causes and their complex interaction in most disastrous events. Some shortcomings of the ISO 5049-1 (1994) Standard, which is internationally used for the design of mobile BMH equipment are highlighted. Current design practice, as observed and noted from survey results, is discussed. The study aims to improve structural safety on future mobile BMH equipment designs. This necessitates an integrated systems design approach across engineering disciplines, based on a comprehensive risk assessment. The revision of ISO 5049-1 (1994) is proposed to provide specific rules and guidelines pertaining to machine protection systems. Additional aspects for consideration during the proposed revision are also discussed. It is further recommended that the structural design engineer fulfil a more prominent role during the final acceptance and handover of mobile BMH equipment. An investigation methodology is proposed to establish the root cause of mobile equipment failures.

DECLARATION

I, the undersigned, hereby declare that:

I understand what plagiarism is and I am aware of the University's policy in this regard;

The work contained in this thesis is my own original work;

I did not refer to work of current or previous students, lecture notes, handbooks or any other study material without proper referencing;

Where other people's work has been used this has been properly acknowledged and referenced;

I have not allowed anyone to copy any part of my thesis;

I have not previously, in its entirety or in part, submitted this thesis to any university for a degree.

MJ Schmidt

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a) This study is based on experience gained while employed by Anglo American PLC. Permission to use the material is gratefully acknowledged. The opinions expressed are those of the author and do not necessarily represent the policy of Anglo American PLC.

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c) Sandvik Mining and Construction.

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f) Dr Geoff Krige for input into the study and my career.

g) Prof Ben van Rensburg for valuable input as study leader.

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LIST OF SYMBOLS

GENERAL

a	Crack depth. [m]
A_{ne}	Effective net area. [m^2]
A_b	Bolt cross section area. [m^2]
D_m	Bolt minor diameter. [m]
D_n	Bolt nominal shank diameter. [m]
E	Modulus of elasticity. [Pa]
$F_{failure}$	Failure load - clevis, bolt. [N]
$F_{T\ failure}$	Failure load of clevis, tension. [N]
$F_{Ultimate}$	Ultimate cylinder tensile load. [N]
K_{IC}	Plane strain fracture toughness. [$MPa.m^{0.5}$]
LC	ISO 5049-1 Load Case.
n	Number of bolts in a bolted joint.
v_w	Wind speed. [m/s]
S_{ut}	Ultimate tensile strength. [Pa]
β	Stress intensity factor.
σ_1	First principal stress. [Pa]
σ_x	Stress component in x-direction. [Pa]
σ_y	Stress component in y-direction. [Pa]
$\mathrm{\sigma_y}$	Material yield strength. [Pa]
τ_{xy}	Shear stress component in xy-direction. [Pa]
ρ	Density of air in wind pressure calculation. [kg/m3]
$\mathrm{\rho}$	Bulk density of material. [kg/m3]

LIMIT STATES DESIGN

C_f	Force coefficient.
f_u	Ultimate tensile strength. [Pa]
k_p	A constant dependant on site altitude.
Q_m	Nominal material live load. [N]
q_z	Free stream velocity pressure of wind at height z. [N/m^2]
V_r	Factored shear resistance of a bolted joint. [N]
V_s	Wind speed. [m/s]
V_z	Characteristic wind speed at height z. [m/s]
W_n	Nominal wind load. [N]

ALLOWABLE STRESS

$R_{p\ 0.2}$	Yield point stress. [Pa]
V_r	Shear resistance based on ISO 5049-1 design load. [N]
σ_a	Allowable tensile strength. [Pa]
τ_a	Allowable shear strength. [Pa]

1 INTRODUCTION

1.1 BACKGROUND

A wide variety of structures are utilised within the mining industry to facilitate the transportation and general handling of bulk materials. Typical bulk handling systems include road tips, conveyors, storage facilities, beneficiation plants, stockyard and waste-handling equipment.

Although all structures are governed by the same design principles, mining structures are considered to be somewhat unusual in comparison to typical commercial and industrial structures because of occasional accidental loads and unforeseen overloading conditions. Uncertainties associated with the ore body mined, blast fragmentation and the inclusion of foreign bodies such as loader teeth, timber, or rock bolts present extraordinary challenges to the designer of support structures and mechanical equipment for mining. The logistics linked to high and variable production requirements often lead to the equipment doing heavier duty than was originally anticipated by the designer. Night shift operations are often marked by operator abuse and subsequent breakdowns. Figure 1.1 below shows an example of a severely overloaded mine hauling truck.

Figure 1.1: Overloaded mine hauling truck.

Unforeseen loads exerted on plant support structures during commissioning are illustrated in Figure 1.2 below.

Figure 1.2: Unforeseen loads exerted on support structures.

Unfortunately a fairly high incidence of structural damage or failure is experienced within the mining industry (42), notwithstanding compliance with appropriate design standards. The failures and incidents recorded within Anglo American Thermal Coal and the broader Anglo American mining operations are considered representative of the mining fraternity as a whole. Improved structural safety is in the interest of all employees and also ensures steady company earnings. Catastrophic failures may cause injuries or fatalities and will inevitably cause significant business interruption since bulk materials mines are usually operated on a continuous basis with scheduled maintenance intervals. The inability to meet contractual obligations to coal-fired power stations, for example, may lead to unexpected power outages.

The study explores the design requirements for mobile BMH and the special conditions encountered during the operation of this equipment in order to highlight the shortcomings of the ISO 5049-1 (1994) Standard (38).

1.2 STUDY MOTIVATION

ISO 5049-1 (1994) is commonly utilised throughout the industry (42) for the design of mobile equipment associated with the continuous handling of bulk materials. It, however, does not include an integrated systems design approach across engineering disciplines. Compliance with the said standard means that the designer has met the design obligation notwithstanding that the limitations of the standard are widely recognised (42). Where equipment damage or failure occurs, the potential dispute between the owner and the original equipment manufacturer (OEM) is not easily resolved when the latter party can prove that the equipment design met the requirements stipulated in the standard or client specification. It may also be extremely expensive for an owner to insist that an OEM provides equipment to more severe design requirements; a significant premium may be demanded due to uncertainties on the part of the OEM when bidding on equipment which goes beyond what is normally supplied.

Although highly skilled and experienced design engineers are usually involved with the delivery of mobile BMH equipment, recent failures of such equipment, designed in First World countries by reputable OEMs, support literature which states that the skills shortage crisis in the engineering industry is yet to be resolved (33, 40, 59). Failures cannot always be attributed to design issues. A wide range of factors could contribute to failures; these include material quality, manufacturing, commissioning, abuse, etc. The fast-track nature of most mining projects nevertheless puts pressure on OEMs to provide new designs with a minimum of engineering effort and this may be exacerbated by the scarcity of design engineering resources. New BMH suppliers, especially from Asia, are increasingly competing for market share with traditional suppliers, who are mainly from Europe. While more and more supplies come from countries with the ability to manufacture at a low cost, the core design competence, which may be limited to basic engineering, is usually kept within parent companies. The drive towards more cost-effective designs may result in less conservative designs which leave little tolerance for unexpected loading conditions or future upgrades. Furthermore, the lack of a proper systems design approach restricts the extent of integration between protection systems limits and structural or mechanical strength. The risk of failure is often not understood when controls are wilfully over-ridden or have not yet been commissioned.

1.3 STUDY AIM

The aim of the study is ultimately to improve the structural safety of future mobile BMH equipment designs, which does not necessarily imply a more conservative design approach, but rather that loading and conditions are accurately assessed. The revision of ISO 5049-1 (1994) (38) will be proposed to provide specific rules and guidelines regarding machine protection systems. Various other suggestions for consideration during the proposed revision will also be given.

A systems design approach, which is properly integrated between the respective engineering disciplines, and based on a comprehensive risk assessment, will be recommended. The need for the structural designer, or representative, to play a more prominent role during the final acceptance and handover of mobile BMH equipment to the owner will be emphasised.

1.4 STUDY OBJECTIVES

A number of selected case studies are discussed in order to demonstrate:

1. The identification of the typical root causes of failures, and the complex interaction of such causes, in most catastrophic events.
2. The complexity of typical collapses and the systematic investigation approach which is required to establish the root cause of failures.
3. Some limitations of the internationally recognised design standard, ISO 5049-1 (1994), which is most commonly used in the consulting industry worldwide.
4. A proposed methodology for failure investigation of mobile BMH equipment and recommendations to facilitate the identification of the root cause of failure.

1.5 SCOPE RESTRICTIONS

1. Although ISO 5049-1 (1994) is widely referenced, this study is not an attempt to resolve its shortcomings in totality. An alternative design approach to address the limitations is nevertheless proposed. Further study, with specialist input from a wider audience, is required to lay down revised rules for the design of mobile equipment for the continuous handling of bulk materials.
2. The review of literature in Chapter 2 is confined to a broad overview of the subject with the focus on loading conditions pertaining to the case studies only. Related aspects directly

relevant to the study are also covered. A detailed comparison of available design standards is reserved for future study.

3. The categorisation of collapses, or factors contributing to failures, is not addressed in the study.

1.6 DISSERTATION OVERVIEW

Chapter 2 – The background, intent and overview of the content of specifications and standards relevant to bulk materials handling structures are explored. The unusual loading conditions often encountered in the mining and minerals industry are highlighted.

Chapter 3 – A variety of incidents and failures are briefly explored to demonstrate some shortcomings of ISO 5049-1 (1994) in ensuring safe mobile BMH machines.

Chapter 4 – A systematic investigation methodology is proposed to determine the root cause of failures of mobile BMH equipment.

Chapter 5 – The details of the failure investigation presented as the main case study are explored.

Chapter 6 – Current design practice is discussed in conjunction with observations made during the case studies. The revision of ISO 5049-1 (1994) is proposed with key focus areas.

Chapter 7 – Conclusions based on the study objectives are discussed.

Appendix A – Terminology and abbreviations referred to in the study are provided.

Appendix B – Detailed wind loading calculations are provided for the waste spreader discussed in Chapter 3.

Appendix C – Survey responses from OEMs regarding design aspects and structural safety relating to mobile BMH machines are provided.

2 LITERATURE REVIEW

This chapter provides an overview of international, national and company specific standards and specifications currently available for the design of materials handling structures. The discussion focuses mainly on the rules for the determination of design loads, although items specifically relevant to the case studies addressed in the following chapters are also covered.

The following documents will be discussed:

1. AA 114/1 (2007) Design of steel structures – Anglo American Company Specification (1).
2. ISO 5049-1 (1994) Mobile equipment for continuous handling of bulk materials – Part 1 Rules for the design of steel structures (38).
3. FEM SECTION II (1992) 2 Rules for the design of mobile equipment for continuous handling of bulk materials, Document 2.131 / 2.132 (31).
4. DIN 22261 (2006) Excavators, spreaders and auxiliary equipment in opencast lignite mines (28).
5. AS 4324.1 (1995) Mobile equipment for continuous handling of bulk materials – General requirements for the design of steel structures (5).
6. AA 254/1 (2008) Stacking and reclaiming equipment, mechanical and structural – Anglo American Company Specification (2).
7. AA 248/2 (2010) Materials handling machines structural components specification – Anglo American Company Specification (3).

The design practice and approach followed by various leading mobile BMH equipment designers are not entirely consistent. It must be understood that leading companies specialising in the design, manufacture, erection and commissioning of mobile BMH equipment across the globe, utilise ISO 5049-1 (1994), FEM SECTION II (1992) or AS 4324.1 (1995) (or a combination of these) for loading criteria and designs. The author conducted a survey by means of a questionnaire amongst six reputable international OEMs that provide mobile BMH equipment to the mining and minerals industry in order to understand the design approaches commonly followed. Current design practice is discussed in view of the outcome of the survey. The discussion provides the basis for the design approach and recommendations proposed in Chapter 6.

2.1 AA 114/1

AA 114/1 (2007) Design of steel structures, is an Anglo American Company Specification (1) and is discussed in this section.

Unlike the other documents mentioned above, this specification is not confined to mobile equipment. Nevertheless, it is included in this chapter in order to demonstrate that Anglo American recognises that the mining industry presents unusual accidental and special loading conditions which are not adequately catered for by national design standards.

Overview of AA114/1
This specification details the requirements for the design of steel structures, and for steel components in structures framed in other materials, for underground and surface applications in mine shafts and plants. SANS 10160-1 (1989) (50) and SANS 10162-1 (2005) (51), form the basis of the specification. Limit states design is thus mandatory. Specific rules and requirements pertaining to the following items are spelled out:

- Design standards, specifications and related publications
- Design responsibility
- Quality management of design process
- Design calculations
- Design drawings and approval
- Materials
- Load factors and load combinations
- Design requirements and procedures
- Serviceability requirements
- Construction details.

2.1.1 LOADS

Nominal permanent and imposed loads are determined in accordance with SANS 10160-1 (1989), but additional clauses are stipulated to cater for mining-specific conditions. The following loading conditions will be covered in this study:

1. Imposed floor loads – It is required to assess these loads taking into account the intended use or occupancy of the structure. Specific minimum uniformly distributed floor design

loads are dictated. Of particular interest is the live load value of 2,5 kPa specified for conveyor gantries, which is an attempt to cater for unintended spillage loads. Figure 2.1 below shows a typical example of the spillage which is often encountered on conveyor gantries due to belt wander, overloaded belts or the sliding of wet material down a steeply inclined belt. Manual unloading of belts onto walkways following an electric trip of the drive, which cannot start with a loaded belt, is not unusual.

Figure 2.1: Conveyor walkway spillage.

2. Wind loads – It is required that the relevant terrain category is assessed in consultation with, and is approved by, the client and the owner. The terrain category adopted for inland terrains is not to be less severe than a category that falls midway between Category 2 and Category 3 as specified in SANS 10160-1 (1989).

3. *Abnormal loads or conditions* – Formal risk assessment is mandatory to establish whether abnormal loads or conditions should be considered in the design.

   Amongst several other items listed for consideration is the impact of vehicles and other moving objects. This does not imply that conveyor structures, as shown in the Figure 2.2 below, be designed for dozer impact loads, but rather that the need for an operating procedure to manage such a risk should have been a documented action item following on from the compulsory risk assessment.

Figure 2.2: Dozer activity on an over-filled stock pile.

A collapsed cable suspension bridge is shown in Figure 2.3 below. Severe corrosion due to the entrapment of moisture around the rope anchors caused the failure after a relatively short service life.

Figure 2.3: Suspension bridge failure due to corroded rope anchors.

4. Erection rigging load – The assessment of nominal loads acting on structures or structural elements specifically designed for erection rigging is to be done with the incorporation of an impact factor of 3,5.

   Rigging loads are non-routine lifts and are classified as safety critical. The impact factor caters for the dynamic effects associated with rigging operations. Figure 2.4 shows an example of major construction rigging activity at a mining operation site.

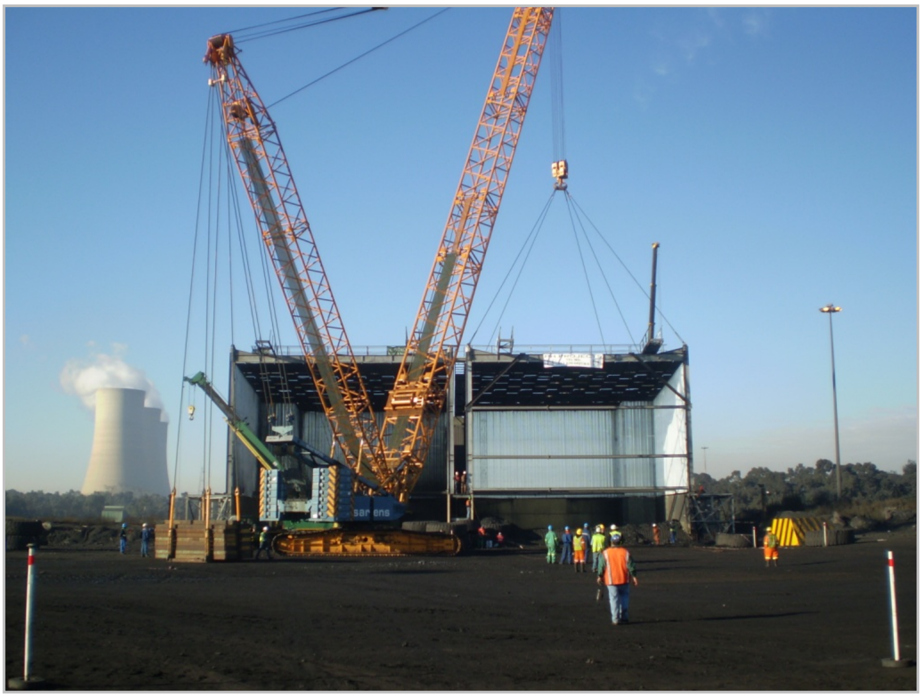


Figure 2.4: Major construction rigging, 80 ton single lift.

2.2 ISO 5049-1

ISO 5049-1 (1994) Mobile equipment for continuous handling of bulk materials – Part 1 Rules for the design of steel structures (38) is discussed below.

The original publication was done in 1980 and it was adopted from FEM 2.131 (31) as an ISO international standard under the reference ISO 5049-1 (1980) and was subsequently revised in 1994. It is based on an *allowable stress* design approach, but provides little guidance on design requirements for structural members.

Although ISO 5049-1 (1994) supposedly consists of Part 1: Rules for the design of steel structures, *and* Part 2: Rules for the design of machinery, *the latter standard was never published*. Part 1 of the standard establishes rules for determining the loads, types and combinations of loads (main, additional and special loads) which must be taken into account when designing steel structures for mobile continuous bulk handling equipment. It is applicable to rail-mounted mobile equipment for continuous handling of bulk materials, specifically:

- Stackers
- Ship loaders
- Reclaimers (fitted with bucket wheels or bucket chains)
- Combined stackers and reclaimers (fitted with bucket wheels or bucket chains)
- Continuous ship unloaders (fitted with bucket wheels or bucket chains).

Specific rules and requirements pertaining to the following items are spelled out:

- Loads
- Load cases
- Design of structural parts for general stress analysis
- Design of joints for general stress analysis
- Calculation of fatigue strength for structural members and joints
- Exceeding allowable stresses
- Fatigue
- Safety against overturning.

Loads are divided into three groups:
Main loads – permanent loads occurring under normal operating conditions.
Additional loads – loads that can occur intermittently during operation or while not operational.
Special loads – loads which should not occur during operation or while not operational, but cannot be excluded.

Main loads
Material loads on conveyors – where the belt load is not limited by automatic devices, flooded belt conditions must be considered in the design.
Encrustation – Loads due to dirt accumulation are taken as no less than 10 % of *the designed belt loading*.
Loads on gangways, stairs and platforms – A concentrated load of 3 kN under worst conditions.

Additional loads

Wind – During normal operation: V_z = 20 m/s unless otherwise specified because of local conditions.

Special loads

Lateral collision with the slope of the stock pile is only specified for bucket wheel reclaimers i.e. no requirements are specified for fixed-boom or slewing stackers.

2.3 FEM SECTION II

FEM SECTION II (1992) Rules for the design of mobile equipment for continuous handling of bulk materials, De La Federation Europeenne de la Manutention France, Document 2.131 / 2.132 (31) is discussed below. It consists of two parts:

FEM 2.131 – Rules for the design of mobile equipment for continuous handling of bulk handling equipment – Chapter 1 Structures; and

FEM 2.132 – Rules for the design of mobile equipment for continuous handling of bulk handling equipment – Chapter 2 Mechanisms.

The rules for the design of mobile equipment for continuous handling of bulk materials were developed by the technical committee of FEM SECTION II (1992) and are widely used internationally. FEM 2.131 was adopted as an ISO international standard under the reference ISO 5049-1 (1994).

Although initially published in 1978, it was revised in 1992 to include fatigue calculations for mechanisms, friction resistances to define drive mechanisms, braking devices and tables describing the notch effect of welded structures. It must be noted that an “Electrical” chapter was planned in a next edition *which was never published*.

The document forms the basis of ISO 5049-1 (1994) and is more comprehensive in that it covers allowable stress design principles in greater detail. Fatigue design is covered in depth while mechanisms, certain safety requirements, tests and tolerances are also addressed. The scope of the rules established includes the determination of loads and load combinations to be considered when designing handling appliances. It also includes the strength and stability conditions to be considered for various load combinations.

2.4 DIN 22261

DIN 22261 (2006) Excavators, spreaders and auxiliary equipment in opencast lignite mines (28), is included in the literature review for completeness of the study.

The standard was specifically written for machines working in large brown coal open-cut mines, including bucket wheel excavators (48), and consists of six parts:

- Part 1: Construction, commissioning and monitoring
- Part 2: Calculation principles
- Part 3: Welding connections, joint types, classification, test instruction
- Part 4: Hoisting winch brakes
- Part 5: Slewing brakes and overload protection devices
- Part 6: Examination of ropes and rope fittings.

Although the above-mentioned documents will not be discussed in great detail, the loading conditions deemed relevant to the study are included in the comparative matrix provided at the end of this chapter.

It must be noted that DIN 22261-2 (2006), Excavators, spreaders and auxiliary equipment in opencast lignite mines, Part 2: Calculation principles, is very similar to ISO 5049-1, although the former standard is far more comprehensive.

DIN 22261-2 (2006) recognises the need for increased walkway loading "in unfavourable conditions". A loading pressure of 1 kPa is subsequently specified to cater for material accumulation on these walkways.

Although DIN 22261-5 (2006) Excavators, spreaders and auxiliary equipment in opencast lignite mines, Part 5: Slewing brakes and overload protection devices, is not a very comprehensive document, the German Mining Standards Committee (FABERG) nevertheless recognised the need to lay down specific rules for overload protection devices, the only one of the available standards discussed in the study to do so.

2.5 AS 4324.1

AS 4324.1 (1995) Mobile equipment for continuous handling of bulk materials – General requirements for the design of steel structures (5), is discussed below.

Although not applicable to the case studies covered in the following chapters, the standard is nevertheless discussed because of its relevance to the general study topic.

This Australian National Standard was published in 1995 subsequent to a number of failures experienced in that country. Part 1 deals with steel structures which *should have been* followed by other parts addressing mechanical, electrical and other aspects, but *only Part 1 was ever published.*

A number of interesting remarks have been extracted from R Morgan's paper *Design of materials handling machines to AS4342* which was presented at the Australasian Engineering Conference of 2012 (48), and are discussed below:

- The standard "adopts a philosophy of not over relying on electrical protection devices for structural integrity".
- "AS 4324.1 has adopted material from DIN 22261 in addition to material from ISO 5049.1."
- "AS 4324.1 has been in use for over 16 years and major machine suppliers and design audit engineers operating in Australia are now familiar with the document and are making due allowance for the differences between AS 4324.1 and the International Standard ISO 5049.1. Since the introduction of AS 4324.1, the majority of new machines in Australia have been subject to a third party design audit. Its application in the purchase of bulk handling equipment for Australian ports and mines has generally resulted in robust and reliable machines which are expected to offer long-term benefits."

There are significant differences between the ISO 5049-1 (1994) and AS 4342.1 (1995). Although Morgan (48) claims that the use of ISO 5049-1 (1994) was discontinued in Australia subsequent to the publication of AS 4324.1 (1995), the author is aware of a number of mobile BMH machines provided in the past decade to Australian operations which were designed in accordance with the former standard. Machines designed to AS 4324.1 (1995) are generally heavier because a more conservative design approach is followed. Survey results, as provided in Appendix C, indicate that OEMs on average feel that machines are between 10 and 30 % heavier when designed to AS

4324.1 (1995) as opposed to ISO 5049 (1994). While machines designed inadequately are, of course, not acceptable, making machines heavier for no apparent reason is poor engineering practice. More expensive support infrastructure e.g. rails, soil preparation, support wheels, slew bearing etc. will be required for heavier machines and operational costs may also be higher. It is nevertheless remarkable that the mining and minerals fraternity in Australia realised almost two decades ago that the only available international standard was insufficient to facilitate the design of safe mobile BMH equipment, and yet no changes have been made, or even formally proposed, to ISO 5049-1 (1994).

The AS 4324.1 (1995) is currently under revision (30). According to the chairperson of the Australian Standards Committee ME43, Mr R Morgan (Personal communication 2013-10-30), it is envisaged that the revised standard will be published in May 2014 and will include:
Part 2 – Mechanisms
Part 3 – Electrical and controls
Part 4 – Manufacture, construction, commissioning, operation and inspection.

2.6 AA 254/1

AA 254/1 (2008) Stacking and reclaiming equipment, mechanical and structural (2), is an Anglo American Company Specification which details structural and mechanical requirements for stacking and reclaiming equipment supplied to the said company.

Specific requirements related to the design, environment, materials of construction, mechanical, structural, castings, forgings, corrosion, safety, quality assurance, test and inspection methods as well as packaging and marking are stipulated.

Mechanical components are to be designed in accordance with FEM 2.131/2.132 (31).

Structural designs are to be designed in accordance with ISO 5049-1 (1994) with specific modifications to loading clauses of which the most significant issues are:

- The operating design wind speed must be agreed with the client while the out-of-service design wind speed is to be done in accordance with SANS 10160-1 (1989) for a mean return period of 50 years and a minimum value of 40 m/s.
- End-on collision of the stacker boom with the pile must be considered.

2.7 AA 248/2

AA 248/2 (2010) Materials handling machines structural components specification (3), is an Anglo American Company Specification which details the requirements for the design, construction, monitoring and documentation of all structural components of materials handling machines supplied to Anglo American.

The specification was compiled by a technical committee, of which the author was a member, subsequent to a number of failures within the Anglo American Group. The document is far more comprehensive than the AA254/1 (2008) specification discussed above and also covers draglines, ship loaders and wagon tipplers in addition to stacking and reclaiming equipment.

The specification details the following technical requirements to be specified by the engineer:

- Duty and service
- Handled product characteristics
- Operational conditions.

Specific requirements relating to the following matters are also detailed:

- Design risk assessment
- Loading
- Analysis
- Design
- Construction
- Commissioning procedures
- Monitoring
- Quality assurance
- Documentation.

It must be noted that AA 248/2 (2010) is a fairly recent publication. To date, it has not been used for project execution.

2.8 OEM SURVEY

The author contacted leading international OEMs specialising in the design, fabrication and commissioning of mobile BMH equipment to partake in a high-level survey consisting of 15 questions focusing on design and structural safety directly related to mobile BMH equipment. All questionnaires were completed by high-profile design engineers or engineering managers directly involved with the successful delivery of up to 100 machines. Feedback responses were obtained from the companies listed in Table 2.1 below:

Table 2.1: Leading international companies interviewed.

OEM	**Remark**
FLSmidth	Global company
Huadian Heavy Industries, (HHI)	Leading Chinese company
ThyssenKrupp	Global company
Sandvik	Global company
Schade	Global company
Tenova-Takraf	Global company

Feedback responses on the questionnaires are provided in Appendix C. In view of the case studies which are discussed in the following chapters, it is important to note that most mobile BMH machines are designed by international companies in accordance with the rules and requirements specified in ISO 5049-1 (1994), except for in Australia where AS 4324-1 (1995) applies, unless otherwise agreed between the OEM and client. Survey responses nevertheless clearly show that the design approaches followed by various leading OEMs are not consistent. The survey outcome is discussed in more detail in Chapter 6 of the study.

2.9 COMPARISON OF LOADING CASES

A brief summary of loading cases for stacking devices directly related to case studies 1 and 3 and discussed in Chapter 3, is included below in Table 2.2 to highlight similarities and differences.

Table 2.2: Comparative summary of selected stacker loading cases.

	Walkway loading	Wind load - In operation	Wind load – Not in operation	Encrustation
AA 114/1 (2007)	2,5 kPa acting on full vertical projected area.	SANS 10160-1 (1989) Not less severe than midway between Category 2 and Category 3.	N/A	Allowed for in 2,5 kPa imposed load
ISO 5049-1 (1994)	3 kN pointload at worst position	20 m/s (but consider local conditions)	Above ground height dependant. (as per Table 3)	Not less than 10 % of conveyed belt load for mines
FEM 2.2131 (1980)	3 kN pointload at worst position 1,5 kN/m for access 1 kPa for "large platforms" at "unfavourable conditions"	20 m/s maximum (unless otherwise stated)	Above ground height dependant. (as per Table T.2. – 2.3.6)	Not less than 10 % of conveyed belt load Unless otherwise specified by client
DIN 22261-2 (2006)	3 kN pointload at worst position. 1 kPA at "unfavourable conditions"	20 m/s unless a higher wind speed is specified)	35,8 m/s (unless higher specified)	Presumed 10 % of conveyed belt load (without special verification)
AS 4324-1 (1995)	3 kN pointload at "any point"	20 m/s (but consider local conditions.)	AS 1170.2 in most adverse direction but not less severe than 20 m/s	Not less than 10 % of conveyed belt increase allowance for sticky material.
AA 254/1 (2008)	As per ISO 5049-1 (1994) and SAISC Red Book.	To be agreed with client	SANS 10160-1 (1989) No less than 40 m/s	As per ISO 5049-1 (1994)
AA 248/1 (2010)	Not less than 3 kPa	Specific geographical operating conditions shall be defined	Risk based. If FNF < 1,0 Hz dynamic analysis required FNF = Fundamental Natural Frequency.	Allowed for in 3 kPa imposed load

A comparison of digging forces as dictated by ISO 5049-1 (1994) and AS 4324-1 (1995) is shown below in Table 2.3. This comparison will be referred to in Chapter 3 when the failure of a portal reclaimer is discussed in Case Study 3.

Table 2.3: Simplified comparison of digging resistance.

Load condition	ISO 5049-1 (1994) Clause extract	AS 4324.1 Clause extract
NDR	3.1.4.1: $\text{NDR} = \dfrac{\text{Motor power}}{\text{reducer efficiency x chain speed}}$	3.3.7: $\text{NDR} = \dfrac{\text{Motor power}}{\text{reducer efficiency x chain speed}} \text{ x } 1{,}1$ (But consider load limiting device if fitted)
NLR	3.1.4.2: NLR = 0,3 x normal digging resistance	3.3.8: The greater of: 1) NLR = 0,3 x digging resistance for full drive power 2) NLR = 1,1 x limited torque
ADR	3.2.4: ADR = motor starting torque for empty chain	3.3.4 The greater of: 1) $ADR =$ 1,1 x limited torque or 2) $ADR =$ 1,1 x greatest overload protection value or 3) $ADR =$ 1,5 x full load motor torque
ALDR	3.2.4: $ALDR =$ 0,3 x abnormal digging resistance	3.4.5: Derived from the greater of: 1) ALDR = 1,1 x limited torque or 2) ALDR = 1,1 x greatest overload protection value or 3) ALDR = Maximum torque of lateral drive motor

Where:
NDR = Normal digging resistance
NLR = Normal lateral resistance
ADR = Abnormal digging resistance
ALDR = Abnormal lateral digging resistance

2.10 SUMMARY

This chapter explored the relevant standards used by reputable international OEMs specialising in the design, supply and commissioning of mobile BMH machines. The key focus area was on loading requirements, to provide a platform for the case studies which will be discussed in the following chapters. These selected case studies are all related to mobile BMH machines. A company specific design specification, AA 114/1 (2007), utilised for the design of permanent structures, was nevertheless used as a basis to motivate the unusual loading conditions which exist within the mining and minerals industry. Additional relevant company specifications, specifically associated with the design of mobile BMH machines, were included in the discussion.

It is important to note that the recognised standards i.e. FEM SECTION II (1992), ISO 5094-1 (1994) and AS 4324-1 (1995), used for the design of mobile BMH equipment, focus largely on structural design and only some mechanical aspects. Additional chapters or parts, which would address mechanical, electrical and other aspects, were originally planned for the above-mentioned standards, but never published. Only DIN 22261-2 (2006) and AS 4324-1 (1995) have some reference to machine protection systems. The latter standard is currently under revision.

Survey results, as provided in Appendix C, indicate that the design approaches followed by various leading OEMs are not consistent.

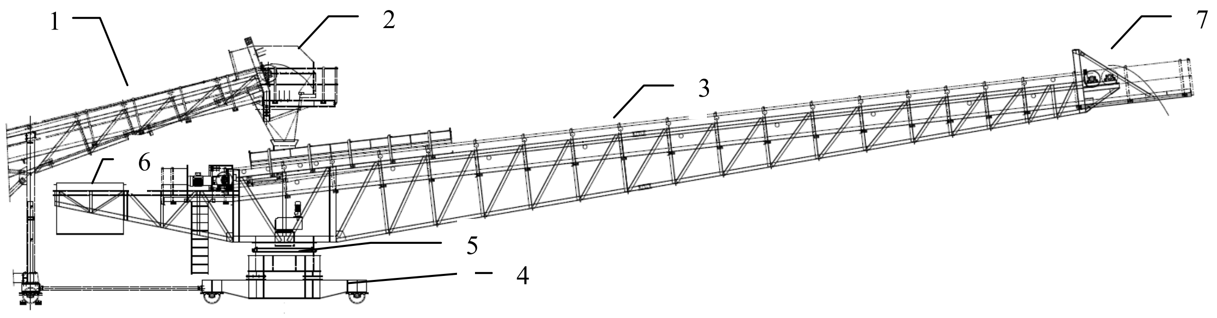
The revision of ISO 5049 (1994) will be proposed in the Chapter 6 where the findings and conclusions from the selected case studies and survey results are revisited.

3 CASE STUDIES

This chapter covers a number of concise case studies in which the original designs of the mobile BMH machines involved were deemed to comply with ISO 5049-1 (1994), but nevertheless failed catastrophically. Only key issues directly related to the root cause of failure will be discussed. Brief descriptions of the BMH equipment discussed in the studies are provided. General design terminology, as well as terms specifically used within the BMH industry, is provided in Appendix A.

3.1 WASTE SPREADER COLLAPSE

Waste spreader design and utilisation may vary significantly. The spreader discussed in this section of the study was used to create a waste dump by discharging waste material in a planned footprint configuration by moving a series of semi-fixed feeding belt conveyors. The main machine components associated with a typical waste spreader are illustrated in Figure 3.1 below.



1 -	Fixed tripper	5 -	Revolving frame
2 -	Tripper discharge chute	6 -	Ballast mass
3 -	Spreader boom	7 -	Spreader head end / discharge
4 -	Bogie carriage		

Figure 3.1: Waste spreader, annotated.

The machine can be described as follows:

- The material feed system consists of a series of conveyors which transport waste material from the beneficiation plant to a conveyor which is equipped with a fixed tripper and is located on the dump.
- The tripper facilitates the feeding of material onto the spreader boom where it is ultimately discharged at the head end of the machine to form the waste dump.

- The spreader is located at the top of the dump, approximately 60 m above the normal ground level in the case study discussed.
- The spreader boom is essentially a balanced beam which utilises a ballast mass to counteract its own weight and the material load.
- The boom is mounted on top of a bogie carriage and can be slewed sideways to spread the discharged material to develop the waste dump. The sideways slew motion of the boom is facilitated by a motorised revolving frame mounted on top of a bogie carriage.
- The spreader and tripper car, which are fixed to the spreader carriage, are occasionally moved forward along rails to advance the waste dump footprint.
- Belt extensions are periodically done behind the tripper car when the spreader is advanced.

3.1.1 BACKGROUND

Shortly after the commissioning of the spreader, high wind loads associated with a thunder storm in the region caused the failure. The buckling of the boom's bottom chord initiated the collapse (26). The machine was designed in accordance with ISO 5049-1 (1994). The author had already commenced with the design audit, on request of the client, when the collapse occurred. Figure 3.2 below shows the collapsed machine on top of the dump, while Figure 3.3 provides a close-up view of buckled structural members.

Figure 3.2: Waste spreader failure.

Figure 3.3: Close-up of buckled truss bottom chords.

Although the overall structural design was not robust, drastically reduced design criteria are considered to be the root cause of the failure. This case study therefore focuses on matters related to the design loading aspects only.

3.1.2 KEY FINDINGS FROM THE FAILURE INVESTIGATION

The need to design in accordance with local wind conditions was not fully appreciated by the designer. Design criteria provided in ISO 5049-1 (1994) were presumably used to motivate a less stringent design allowance for wind loading based on economic considerations. The design allowance made for encrustation loading was done in accordance with the guideline requirements of ISO 5049-1 (1994). Site conditions show that these allowances were insufficient.

A comparison of the consultant's design criteria, the requirements of ISO 5049-1 (1994) and the loading as derived from SANS 10160-1 (1989) is shown in Table 3.1 below (26).

Table 3.1: Design loading comparison.

	OEMs Design Criteria	**ISO 5049-1 (1994)**	**SANS 10160-1 (1989)**
Terrain category	-	-	Adopt Terrain 2 - conservative
Wind load - In operation	30 m/s (Owners representative granted a concession)	20 m/s (but consider local conditions.)	V_z = 47 m/s Terrain 2 Class B @ 60 m
Conveyor walkway load	3 kN @ worst position	3 kN @ worst position	-
Encrustation	10 % of theoretical effective load	Not less than 10 % of theoretical effective belt load	-

Wind loading

From Table 3.1 above, it is obvious that the wind speed under operational conditions was severely underestimated. Clause 3.3.6 of ISO 5049-1 (1994) nevertheless makes provision for the wind load of non-operating machines i.e. shutting down of machines under storm conditions. Under the mentioned clause a wind speed of 42 m/s would be applicable considering the working height of approximately 60 m above normal ground level.

For operational conditions Clause 3.2.1 of ISO 5049-1 (1994) states "That during handling, a wind speed of v_w = 20 m/s (72 km/h) shall be assumed, unless otherwise specified because of local conditions."

In order to fully grasp the effect of wind loading, the equations governing the resultant wind force on latticed structures as provided in SANS 10160-1 (1989) must be considered. Wind loading calculations, as provided in Appendix B, show that the resultant wind force *F,* was underestimated by a factor of 2,5.

It must be noted that the wind loading was initially correctly assessed according to SANS 10160-1 (1989) procedure i.e. for a wind speed of 47 m/s. A concession was nevertheless granted to design the structure to withstand a wind of 30 m/s as per the design consultant's modified design criteria. This change was allowed by the project clerk of works, who was a carpenter by trade, and had no idea of the implications of allowing the concession request (42).

The severity of the storm at the time of the collapse was not assessed since no anemometer readings were available at the site. It is therefore not possible to determine if the wind loading exerted on the machine at the time of collapse exceeded the requirements of SANS 10160-1 (1989).

Encrustation and other contributing factors

The encrustation allowance of 10 % of the material load on the belt, in the opinion of the author, is deemed completely insufficient for the realistic operational conditions generally associated with this type of equipment. Although belt scrapers are commonly used at the head end of the returning belt to minimise material carry back, certain sticky and wet materials are problematic and difficult to remove. Belt scrapers require continuous maintenance and adjustment to be effective. The de-training of a belt conveyor can occur for various reasons. De-training will result in material spillages on support structures and walkways. Figure 3.4 below illustrates the severity of local encrustation on the structural members of the spreader boom after a short period of operation. The buckled bottom chord members of the boom, close to the revolving frame, can also be observed.

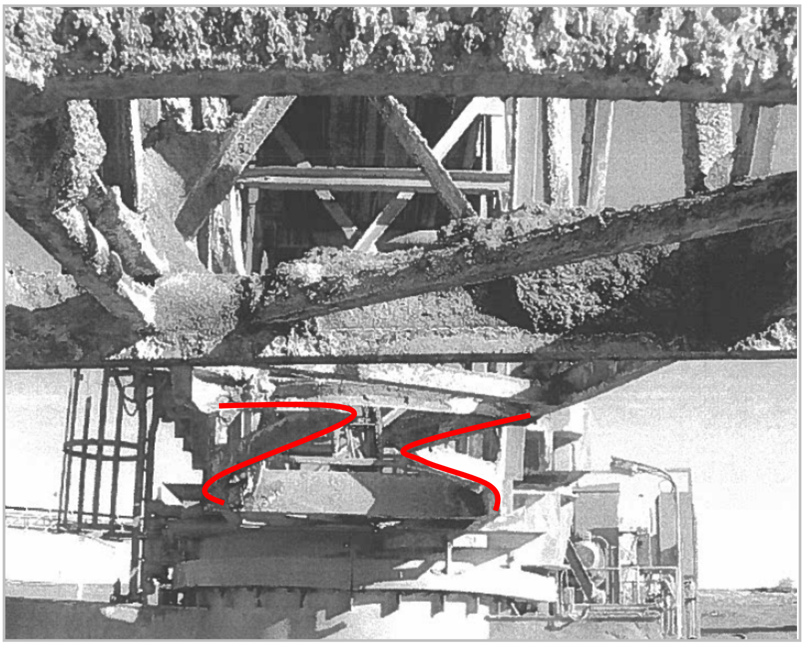


Figure 3.4: Localised encrustation, buckled bottom chords highlighted.

The design of stairs platforms and gangways must cater for a concentrated load of 3 kN applied under the worst conditions as described under Clause 3.1.8 of ISO 5049-1 (1994), in addition to other loading conditions defined in the said standard. The clause quoted above caters for local loading effects only.

The load application and duty of the spreader boom structure are comparable to a conventional conveyor gantry typically encountered within the mining environment. Clause 4.9.2 of the AA 114/1 (2007) (1) requires that a live load value of 2,5 kPa be applied across the vertically projected area of the gantry structure. Although it could be argued that this value is conservative for a mobile machine, it would nevertheless ensure carefree operation if a scheduled cleaning program is put in place. It must be understood that the application of this proposed design value would have a significant effect, not only on the overall strength and stiffness of the structure, but also on the stability of the machine. It must be noted that significant material carry back was present below the return belt of the waste spreader at the time of collapse. The author is of the opinion that this carry back could conservatively be estimated to be no less than 1 kPa acting on the vertical projected area of the spreader boom.

Very specific stability validations are required under Clause 9 of ISO 5049-1 (1994) to ensure that overturning cannot occur. Overturning moments must be resisted by the bogie wheels and slew ring. Although the overall machine overturning moment can be reduced by an appropriately selected ballast mast, the bottom chord truss members of the boom would be more severely loaded. As already stated, the boom collapsed as a consequence of the buckling of bottom chord members close to the revolving frame of the machine. The design audit revealed that boom elements were designed without considering bending moments as a consequence of member eccentricity (26). This omission contributed to excessive structural flexibility and although it was the bottom chord failure that initiated the collapse, this aspect nevertheless played a part. Figure 3.5 below illustrates the incorrect design assumption made.

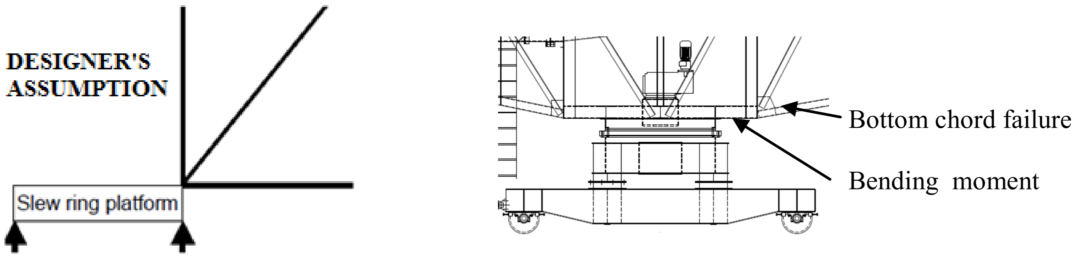


Figure 3.5: Bending introduced in structural frame.

Although the proposed live load of 2,5 kPa could arguably be reduced by 50 %, for example, it would nevertheless have a significant effect on the overall structural design. To demonstrate this point, a basic calculation follows:

$$\begin{aligned}\text{Projected Area} &= \text{Boom Length x Boom Width} \\ &= 25 \text{ m x } 2{,}3 \text{ m} \\ &= 57{,}5 \text{ m}^2\end{aligned} \qquad (1)$$

For an applied live load of 1,25 kPa, as discussed above, the vertical load, disregarding the live load factor for a limit states design approach, can be approximated as:

$$\begin{aligned}Q_m &= \text{Projected Area x Live Load} \\ &= 57{,}5 \text{ m}^2 \text{ x } 1{,}25 \text{ kPa} \\ &= 80 \text{ kN}\end{aligned} \qquad (2)$$

Where:

Q_m = The imposed material load [N/m]

The application and resisting moment forces are depicted in Figure 3.6 below.

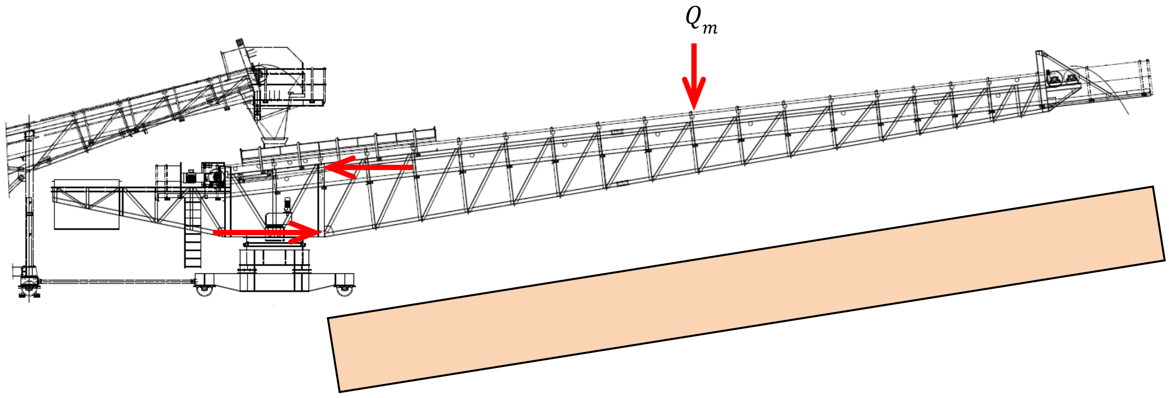


Figure 3.6: Projected boom area, applied load and resisting moment forces.

Although it can be expected that the underestimation of the encrustation loading may have contributed to the failure, it is not suggested that an appropriate allowance would necessarily have prevented the collapse. It would, nevertheless, have ensured a much more robust design. The stiffness of the boom structure was found to be inadequate. Remedial work was subsequently required (26). This requirement was based on serviceability criteria. However, should the flexible boom structure experience uplift under wind gusting, a more severe impact load is to be expected (53) under the acting gravitational force when the gust subsides. The bottom chord of the boom would therefore experience a dynamic compression load far greater than steady state conditions when the bottom chord only supports the combined dead load of the structure and encrustation load. Although the wind speed was not recorded at the time of the collapse, the general wind direction was such that it would indeed have generated an uplift force on the spreader boom (45). The excessive flexibility of the boom is therefore considered to be a contributing factor to the collapse.

From the wind loading calculations, as provided in Appendix B, for a wind speed of 47 m/s, the uplift pressure exerted on the boom, as discussed above, would be approximately 1,15 kPa which is of similar magnitude to the encrustation load already calculated. If the force coefficient which is associated with wind loading is considered, hypothetically the boom experienced a dynamically applied load in excess of 80 kN when the gust subsided.

The audit report (26) shows that the load-carrying capacity of the original structure did not meet the requirements of realistic design criteria. The boom was found to be inadequate to withstand lateral wind forces. The collapse of the boom under severe dynamic loading conditions was therefore to be expected.

3.1.3 CONCLUSION

The minimum guideline figure for encrustation, as provided in ISO 5049-1 (1994), was utilised for the design of a waste spreader without evaluating whether the allowance was realistic for the conditions under which the machine would be operating.

The design criteria for wind loading as prescribed by ISO 5049-1 (1994) was not adhered to. Although the initial wind loading assessment was correctly done, a concession request was approved for the relaxation of the design wind loading based on economic considerations.

The contributing effects of insufficient allowance for encrustation, inaccurate wind loading, incorrect design assumptions for boundary conditions of critical structural members and the effect of inappropriate overall boom stiffness were discussed.

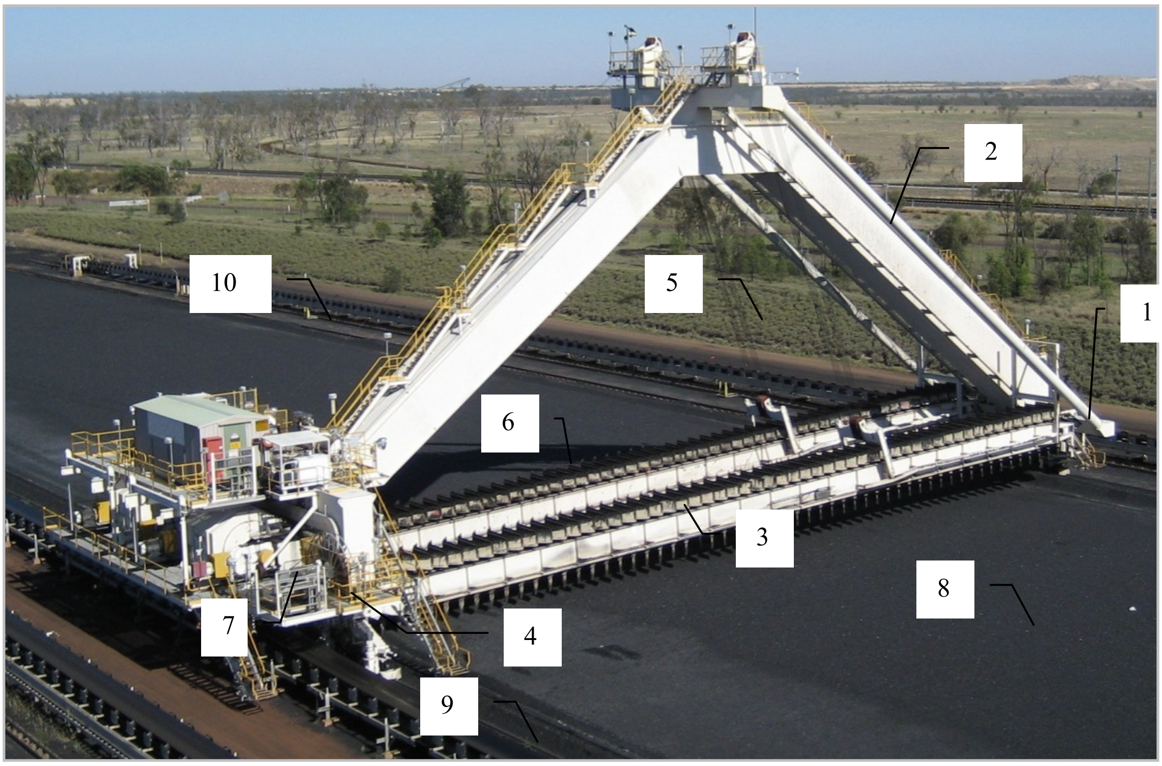
It was highlighted that ISO 5049-1 (1994) provides no guidelines regarding serviceability criteria for mobile BMH equipment.

The root cause of failure is deemed to be overloading conditions exerted on a machine structure of insufficient stiffness, due to a combination of design errors and the underestimation of design loading.

3.2 PORTAL RECLAIMER COLLAPSE

The collapse of a portal reclaimer is discussed in this section of the study. The focus is on matters relating to the machine protection systems and accidental loads associated with the failure, and earlier events that preceded the failure. The author had no direct involvement with this investigation, but analysed relevant information pertaining to the failure (24).

Different variations of portal reclaimers are encountered within the materials handling industry. A representative machine in the field with annotation of the main parts is illustrated in Figure 3.7 below.



1 -	End carriage	6 -	Scraper chain
2 -	Portal structure	7 -	Ramp trough
3 -	Twin boom	8 -	Stockpile bed
4 -	Pivot	9 -	Yard conveyor belt, reclaiming
5 -	Rope system	10 -	Rails for long travel

Figure 3.7: Typical portal reclaimer, annotated.

The main machine components of a typical portal reclaimer are described below:

- End carriages are located on rails positioned on both sides of the stockpile.

- The end carriages are equipped with a combination of driven and non-driven wheels which facilitate the travelling motion along the rails.
- The A-framed portal structure spans the carriages and facilitates boom raising via a winch rope system.
- The boom houses scraper chains onto which scraper buckets are fixed. The chains pass around a drive and tail sprockets.
- The boom is provided with a pivot to allow the scrapers to be positioned on, and reclaim from, the side face of the stockpile.
- A winch is utilised to raise and lower the free end of the boom, while the other end is fixed to the pivot.
- The chain drive is mounted above the end carriage on the discharge side.
- The material which is reclaimed from the chain scraper system is ultimately discharged onto the yard belt conveyor which runs parallel to the stockpile bed.

3.2.1 BACKGROUND

Prior to failure, the machine had been in production use for several months, although the final commissioning of the collision protection system had not been concluded. At the time of the collapse, the designed reclamation rate was being exceeded by approximately 30 %. The stockpile proximity probes appeared to not be working, which led to unexpectedly high digging forces leading to the failure as shown below in Figure 3.8.

Figure 3.8: Failure of the bogie on a portal reclaimer.

3.2.2 KEY FINDINGS FROM THE FAILURE INVESTIGATION

The key findings of the investigation done by others, but which are relevant to the study topic, are discussed below.

Protection systems interrelated with structural design requirements – The lateral resistance of the machine was insufficient to withstand the forces generated within the structure when excessive digging was experienced. Proximity probes, which detect the stockpile height, are fitted to ensure that the digging depth of rake buckets is maintained within the prescribed limits. These devices did not function properly or had not yet been commissioned and so were switched off, resulting in excessive digging forces (24, 42).

Electric drive motors are equipped with protection relays to limit the electric current which can be drawn during operation, i.e. the applied system torque can be limited. Industry practice suggests that the overload protection is set to a value of 5 to 10 % above the peak system design load (7). The protection study report, compiled subsequent to the failure, indicated that the motor protection relay setting on the reclamation drives was at a default value of 2 instead of 1,05 (24). The mechanical design for the scraper drive system dictated an installed motor power requirement of 154 kW, which implies that the next size up of 160 kW was specified. During procurement, 185 kW motors were supplied due to unavailability of the 160 kW motors. This decision was made without consultation with the relevant design engineers (24).

Upon investigation, it was also found that the fluid couplings installed between the drive motors and reducers were rated at service factors which allowed for the delivery of a reclamation drive torque which was only marginally below the maximum electric motor torque. Torque transfer through fluid couplings can be limited according to the design requirement by reducing the percentage of oil fill, which is normal practice. The commissioning data revealed that the machine was commissioned with 18 litres of oil in the fluid coupling under discussion. Subsequent to the failure it was determined that only 15,8 litres were required. In a typical fluid coupling, any oil over the proper fill level, will lead to a significant increase in torque transfer capacity (24).

A simplified comparison of the requirements relating to digging forces as prescribed by ISO 5049-1 (1994) and AS 4324.1 (1995) was summarised in Chapter 2, Table 2.3. It must be noted that AS 4324.1 (1995) facilitates a more detailed design approach and guides the structural engineer in an ordered manner to a thorough understanding of loading conditions in a selection of equipment.

The author is of the opinion that the requirements of ISO 5049-1 (1994) were not satisfied, considering that the required motor protection settings were not adhered to and that oversized motors were supplied. The machine could not withstand the motor starting torque as prescribed for the abnormal digging resistance criteria. The effect of characteristic motor start-up torque curves is illustrated below.

Depending on start-up torque control, the motor torque during start-up could be higher than 250 % of the operating torque of the motor, depending on the motor type selection. From Figure 3.9 below, start-up curve B is representative of the typical selection for a scraper drive. It would therefore be possible to achieve 200 % of the motor full-load torque as shown below, as opposed to the 105 to 110 % (if protection relays were correctly commissioned).

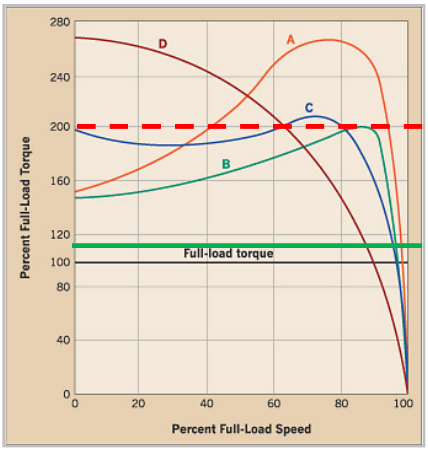


Figure 3.9: Characteristic start-up curves for different electric motors (6).

Accidental loads – The machine had collided with the stockpile a month prior to the failure as discussed above. It is worth noting that the original design did not make provision for loads associated with the collision of the machine against the stockpile or another machine. The technical support team for the incident investigation and recovery recommended that the machine structures be strengthened to withstand the full speed impact load associated with the above-mentioned collision scenarios (24).

Multi-disciplinary design integration – The lack of proper design integration between mechanical, structural, electrical, and control and instrumentation engineering disciplines was revealed during the investigation (24). It is essential that the structural engineer understands the effect and magnitude of forces that could be exerted on machine structures under abnormal conditions and equipment selections. The mechanical and, likewise, the electrical design engineer, must understand how the selection and commissioning of equipment, such as fluid couplings and electric motors, could have an adverse effect on structural design parameters. Interaction between the control and instrumentation, and the structural and mechanical designers to ensure that alarm levels and limits are correctly designed and commissioned cannot be overemphasised. This aspect is discussed in more detail in Chapter 6.

3.2.3 CONCLUSION

The author's conclusion, derived from analysing documentation relevant to the failure of the portal reclaimer, is that the malfunctioning and incorrect commissioning of machine protection systems resulted in accidental loads which were much higher than were originally anticipated in the structural design.

Various compounding factors contributed to the failure:

- The proximity probes, which are a key element of the primary protection system, did not function as intended.
- The electronic overload protection system of the scraper chain drive motor was not commissioned correctly. Oversized electric motors were procured without consultation with the design engineers.
- The fluid coupling of the scraper chain drive was overfilled resulting in higher than expected torque transfer capacity.

The design rules provided by AS 4342-1 (1995) facilitate a more robust overall machine design than ISO 5049-1 (1994). Only aspects related to digging forces were discussed.

The level of design integration between the various engineering disciplines is a concern and should be addressed through a structured process. Final acceptance and approval of the machine by the structural design engineer, or a representative who understands the structural limitations of the equipment, is crucial.

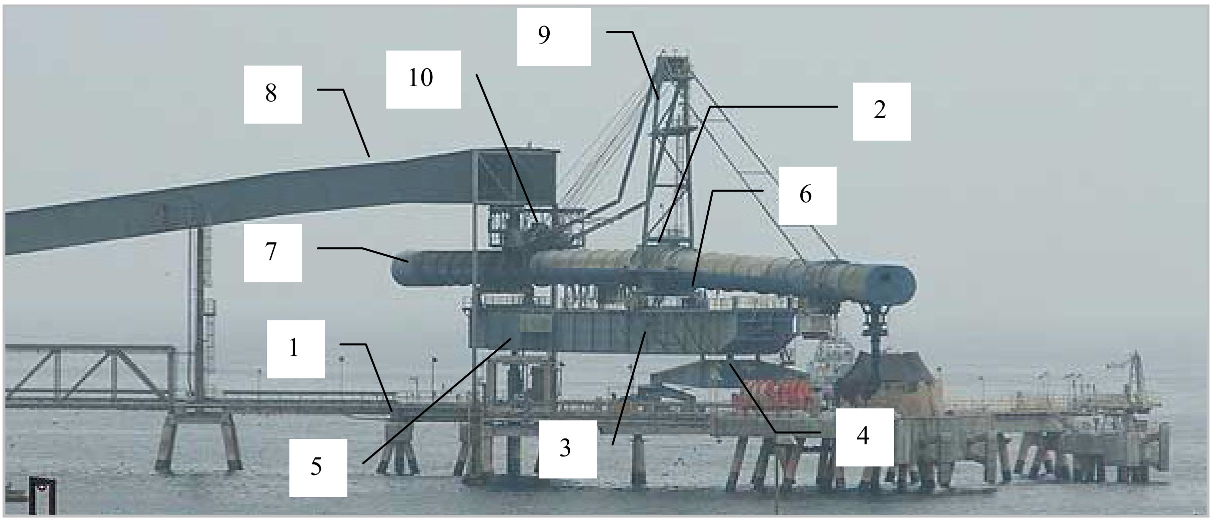
This collapse highlighted the importance of understanding the additional risks associated with the production use of a machine which has not been fully commissioned, and in which protection systems may be inoperative and where stockpile volumes have not yet been fully calibrated. The operation of machines which have not been fully commissioned must be prohibited, regardless of production pressures.

3.3 PARTIAL COLLAPSE OF A SHIP LOADER

The partial collapse of a ship loader, which led to the death of three workers, is discussed in this section of the study. The focus is on matters relating to special design requirements and the correct assessment of future risks pertaining to change control and loading. The background against which ISO 5049-1 (1994) was developed is also briefly explored.

The author had no direct involvement with the investigation, but analysed relevant information pertaining to the failure (23).

The main components of the ship loader installation are highlighted in Figure 3.10 below.



1 -	Jetty structure	6 -	Bogies – shuttle section
2 -	Shuttle structure	7 -	Tubular boom enclosure & discharge conveyor
3 -	Support girder	8 -	Feed conveyor
4 -	Equaliser beam and bogies	9 -	Mast
5 -	Pivot bearing	10 -	Winch and rope system

Figure 3.10: Ship loader configuration, annotated.

A general description of the ship loader configuration is provided below:

- The ship loader is supported by a jetty structure and includes a shuttle structure and a pair of support girders.
- The ship loader girders are supported at the outer end, where the ship is loaded, on a set of equaliser beams and bogies which move along a rail mounted on top of a curved beam.

- The rear end of the girders is supported by a pivot bearing which is mounted on a vertical pile which passes through the deck of the jetty into the seabed.
- The shuttle section of the ship loader moves along rails above the main girders and is fitted with bogies. These bogies allow the shuttle to be extended to reach over the holds of a moored vessel and to be retracted to the maintenance position.
- The shuttle comprises a tubular boom which houses a conveyor. The boom connection is pivoted.
- A mast is fitted above the pivot point and is connected to the outboard end of the boom with fixed stays. Luffing of the boom is facilitated by means of a winch and rope system.
- A swivelling discharge chute is attached beneath the outboard end of the boom and is lowered into the hold of the ship as the boom is luffed down.
- Inside the shuttle and boom is a conveyor belt which runs from the tail end compartment of the shuttle to the front end of the boom.
- The ship loader is fed via a series of conveyors.

3.3.1 BACKGROUND

During the project design phase, the need for a light-weight ship loader structure was identified primarily due to severe prevailing seismic conditions. The marine support structures had to be light enough to be carried by slender piles which would be able to move laterally under the influence of an earthquake. The tubular boom concept was subsequently developed since it could satisfy environmental and structural design considerations. The sudden collapse occurred after the ship loader had operated successfully for over a decade. The partially collapsed ship loader is shown in Figure 3.11 below.

Figure 3.11: Ship loader, partially collapsed.

3.3.2 KEY FINDINGS FROM THE INVESTIGATION

The machine was designed in accordance with ISO 5049-1 (1994) which stipulates an allowance for encrustation loading of not less than 10% of the material load being transferred. The design consultancy nevertheless recognised the need to increase the design allowance for encrustation substantially and subsequent to lengthy discussions, it was concluded that a vacuuming system appeared to be the most appropriate system for maintaining the permitted spillage levels. An agreed design criteria pertaining to encrustation loading was adopted which would cater for 3,5 times more than the minimum as defined by ISO 5049-1 (1994). The project was successfully delivered. However, over a period of time, the vacuuming system proved to be unreliable and was eventually decommissioned. After a successful service life of 14 years, the tube structure finally failed under brazier buckling subsequent to the severe spillage overload. Table 3.2 shows the comparative encrustation loading as a percentage of the material load conveyed in the system.

Table 3.2: Encrustation loading comparison.

Loading	ISO 5049-1 (1994)	Agreed design criteria	Approximate loading at the time of failure
Encrustation allowance	Not less than 10 % of theoretical effective belt load – 20 kg/m (minimum)	35 % of material load conveyed – 70 kg/m	700 % of material load conveyed – 1400 kg/m

3.3.3 DISCUSSION

It appeared that the mine site personnel were not aware of the high level of risk posed by decommissioning of the vacuum cleaning system. Knowledge regarding the sensitivity of the design and hence the requirement to maintain spillage levels to an absolute minimum in order to maintain the integrity of the structure, was lost to operations personnel present at the time of the collapse.

The following direct quote from the investigation report (23) is very relevant to this study topic: "The encrustation allowance of ISO 5049-1 (1994) clearly envisaged an open structure where most of the spillage could fall through to the ground or to the water and only provided allowance for the material which was expected to become encrusted on the members of the open structure. It was not written with enclosed structures in mind and did not provide any guidance to the designer to make additional allowance for spillage which could not fall away from the machine due to the enclosure or the presence of spill trays. Over decades of use of this code, a number of weaknesses have come to be recognised and, in the case of Australia, led to the development of an Australian Standard AS

4324.1 for such machines where a number of issues, including the issue of the potential additional spillage collection, were addressed." It could nevertheless be argued that a responsible person carrying out routine inspections should have identified the excessive spillage below the conveyor as a potential threat.

When evaluating the spillage loading at the time of the collapse, it is evident that the structure had more than sufficient redundancy to withstand normal (and perhaps abnormal) spillage conditions exceeding the design parameters. The author is of the opinion that the original design was sound and that the root cause of the failure should instead be classified under change control. Had the risk been properly assessed in consultation with a structural engineer prior to the decommissioning of the vacuuming system, the cleaning system would most likely have been reinstated or substituted with manual labour.

3.3.4 CONCLUSION

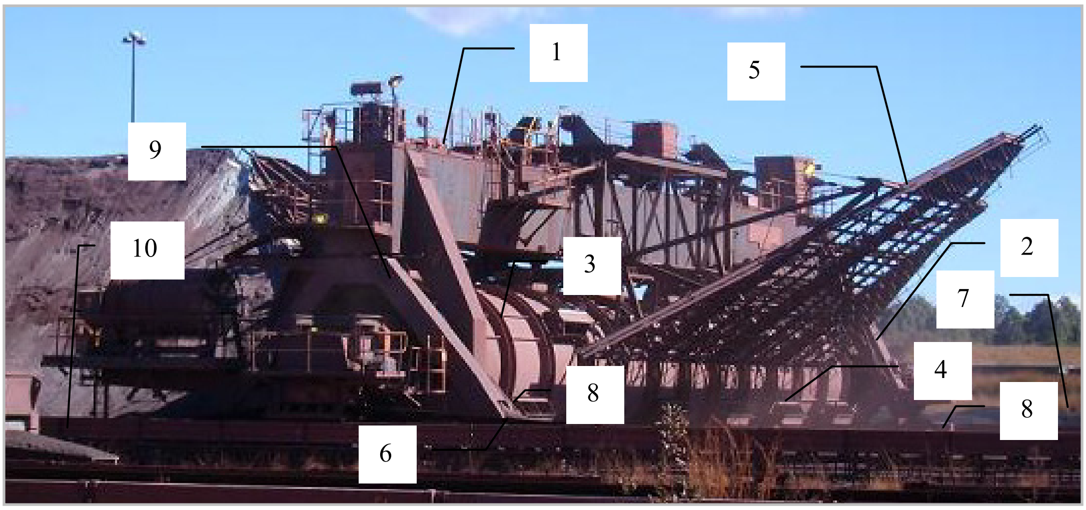
This case study emphasised the need for a risk-based design approach, especially when embarking on unique mobile BMH machine designs. Mitigating measures resulting from the risk assessment may have led to the recognition of the requirement for clear signage along the shuttle system graphically demonstrating that the spillage loading had to be managed to maintain the structural integrity thereof. The magnitude of spillage as noted during the time of the collapse shows that the structure had significant redundancy and could comfortably withstand normal (and perhaps abnormal) spillage loads.

The importance of change management in operational processes, pertaining to structural integrity, was highlighted. Individual responsibility and vigilance remains a key aspect of safety.

3.4 STRUCTURAL DAMAGE TO A DRUM RECLAIMER

This section considers an incident, or perhaps a recurring incident, whereby significant structural damage was caused to the superstructure of a drum reclaimer because of inadequate interlocks incorporated into the control system.

Although several variations of these machines are provided by different OEMs within the mining and minerals industry, many similarities can be observed in the basic elements and the overall machine construction of drum reclaimers. The main components of a typical machine are illustrated in Figure 3.12 below.



1 -	Bridge	6 -	Bogie
2 -	Fixed legs, portal	7 -	Stockpile bed
3 -	Drum	8 -	Yard conveyor
4 -	Reclamation buckets	9 -	Swivel legs
5 -	Harrow	10 -	Rails for long travel

Figure 3.12: Typical drum reclaimer, annotated.

A general description of the machine and key parts follows:

- A bridge, spanning the stockpile bed, is supported on a set of fixed legs and an adjacent set of swivelled legs.
- Both sets of legs are fixed to equalised rail-mounted bogies.
- The bogies are motorised to allow longitudinal travel along the stockpile bed.
- A drum is supported on rollers located on the fixed legs.

- The drum is rotated with a drive mechanism and is equipped with a set of buckets on its circumference.
- The drum houses a cross conveyor which receives the material reclaimed by the buckets and discharges it onto the yard conveyor via chute work.
- A reciprocating harrow allows material to run down the stockpile reclaim face in a controlled way.
- Material is reclaimed as the machine travels at a controlled speed in the direction of the stockpile. As the drum rotates, the buckets are filled and discharge their contents into the cross conveyor feed chute.
- Material is ultimately discharged onto a yard conveyor belt running along the stockpile.

3.4.1 BACKGROUND

Although no failure occurred as such, significant damage was done to the support legs of the drum reclaimer when the control system of one of the long travel drives malfunctioned, resulting in a skewing action which imposed excessive loading which was not considered in the original design. The machine had been in service for decades. The future maintenance and alignment of systems and mechanisms associated with the long travelling motion as well as the drum are expected to be problematic (25). The author had no direct involvement with the investigation, but analysed relevant information from reports and had personal communication with the consulting engineer who carried out the inspections on the machines (44). This brief case study was included to demonstrate the importance of an integrated systems design approach, based on a risk assessment.

3.4.2 DISCUSSION

The machine's main structure is supported by four sets of bogie wheels at each end. Independent drive systems are located on both sides of the machine. The overall machine control system was originally configured without interlocks between the two systems and was operated like that for decades. When the control system for the drives on the one end malfunctioned, the drives on the opposite end continued with the long travelling sequence until the drives tripped on overload as a consequence of the skewing of the machine. Severe local damage and permanent deformation was caused to the boxed plate structural section of the fixed legs. The fixed legs had already been replaced twice. Figure 3.13 shows a side elevation of the machine with the fixed leg highlighted.

Figure 3.13: Side view of drum reclaimer, fixed legs highlighted.

Skew control can be achieved by comparing signals from incremental encoders on both sides of the machine (47). Skew should only occur if one side of the machine cannot travel for accidental reasons, e.g. an obstacle on the rails, and if this happens a signal must trigger the immediate shut down of the machine.

The author is of the opinion that the incident highlights the fact that designs were not sufficiently integrated across the various engineering disciplines and that “what if?” scenarios were not adequately assessed during the detail design phase. ISO 5049-1 (1994) does not provide any mandatory requirements or guidelines to address aspects pertaining to machine protection.

3.4.3 CONCLUSION

The control systems associated with the long travel of the machine were not fail-safe. Abnormal loads, not anticipated in the original structural design, were subsequently exerted on major structural members. The equipment was nevertheless operated successfully for many years prior to the skewing incident. Insufficient design integration existed between the structural, mechanical, electrical, and control and instrumentation engineering disciplines during the detail design phase of the original project. The damage could have been avoided by the incorporation of additional protection instrumentation for negligible additional capital cost. Machine protection systems are not addressed in ISO 5049-1 (1994).

3.5 SUMMARY

This chapter covered a number of typical case studies associated with failures of mobile BMH equipment. It provides a basis for the main case study and failure investigation covered in Chapter 5.

Common themes featured in the case studies include:

- The root cause of catastrophic failures on mobile BMH equipment can seldom be ascribed to a single aspect or item of non-compliance with a design standard or established engineering practice. A combination of factors usually contributes to catastrophic events.
- The lack of proper design integration and the need to have an interdependent design approach across engineering disciplines was highlighted.
- Design compliance with ISO 5049-1 (1994) does not necessarily constitute a safe mobile BMH machine design.
- The requirement for a risk-based design approach was pointed out.

The following aspects pertaining to the ISO 5049-1 (1994) Standard were highlighted:

- Serviceability criteria for mobile BMH structures are not addressed.
- No rules or guidance is provided for machine protection systems.
- The standard focuses predominantly on matters pertaining to structural design and provides no rules or guidance to facilitate an integrated design approach across engineering disciplines.

4 FAILURE INVESTIGATION METHODOLOGY

This chapter provides a retrospective view of the structured process followed to reveal the root cause of failure for the main case study discussed in Chapter 5.

A framework for the investigation of failures is proposed through the identification of universal elements and sub-systems typically encountered when dealing with mobile BMH machines. Case studies covered in the preceding chapters are revisited to illustrate the relevance of the proposed guideline elements.

A model for typical investigation teams and key role functions is proposed – Failure investigations and the subsequent reinstatement of the structures and systems generally require a significant team effort. The author's role, responsibility and interaction with various parties involved with the failure investigation, as discussed in the main case study, is provided.

4.1 START OF INVESTIGATION

Operations management personnel will normally initiate the failure investigation by providing a brief which may include preceding incidents or difficulties experienced prior to the collapse. Depending on the specific nature of the failure, operations may or may not have views on the cause of the failure. Specific concerns on certain immediate safety risks may also be expressed. Insurance partners must be consulted as soon as practically possible. Appointing a specialist forensic investigator in consultation with the insurance partner may be justified, but if this is not done, an investigation leader must be assigned as a minimum requirement. It should be noted that due to scarcity of specialised skills, it may be a challenge to immediately secure the services of a suitable forensic investigator.

4.2 OPERATOR REPORTS AND INTERVIEWS

Where applicable, signed statements must be obtained from operators or other personnel working within the immediate area where the collapse occurred. Clues may be provided by what they noticed, felt or heard at the time of failure. Witness reports are deemed to be a time-based input since operators and other witnesses may quickly forget important details if interviews are delayed.

Where it is deemed necessary to conduct interviews with operators, it is essential that the investigation leader heads up these sessions with representation from the OEM and operations management to ensure that unbiased accounts are recorded.

4.3 SITE INSPECTION

A thorough site inspection would logically be the first step in the investigation process. Besides looking for clues to understand the cause of failure, an important outcome of the initial site inspection is to ensure that the area is safe for access or to devise a plan to make failed structures or mechanisms safe. High resolution digital photos must be taken from various angles before moving or removing any structures or parts thereof.

Follow-up site inspections are highly recommended, especially as the investigation develops. Focused observations can be made on site to validate hypothetical assumptions used to develop failure postulations. Key evidence may not be obvious from the first observations.

4.4 PRODUCTION ISSUES

Temporary by-pass systems may have to be designed and constructed to minimise production losses for the interim, until the machine has been reinstated.

4.5 IDENTIFICATION OF FAILED MEMBERS

When production pressures or logistics require that the failure site needs to be cleared, it may be required to mark members which are key to the investigation outcome for identification at a later stage. Since the primary cause of the failure is usually not known early on in the investigation, the clearance of the failure site must only be permitted if it adversely affects production activities and after consultation with appropriate technical investigation personnel. Critical evidence and clues may be lost once the original location and state of members have been disturbed after the collapse.

4.6 REVIEW OF CONTRACTUAL DOCUMENTATION

The validity of design parameters originally specified must be checked against current operational parameters. These may include throughput rates, travelling speeds, material bulk densities handled, etc. It may be required to confirm the validity of the machine's mission profile (36) i.e. the anticipated duty and working parameters.

4.7 FAILURE MODES

The hypothesis of possible failure modes can be developed once the available clues and data have been assessed. Through team discussion, it may be possible to eliminate flawed failure modes that were initially identified. The most obvious failure mode or modes must be evaluated and backed up or eliminated by available evidence and analytical work. The above may require a number of iterations to arrive at a plausible failure mode. It is important to prove, as far as practically possible, how the failure occurred so that recurrence of the failure can be avoided.

4.8 REVIEW OF DESIGN REPORTS

It must be established whether design audits were carried out during the implementation of the original project. These reports may highlight design aspects which were of concern to the third-party reviewer but which were, perhaps, for whatever reason, not addressed.

4.9 SCADA RECORDINGS

Vital data pertaining to the operation of major equipment is usually logged within the control system software, and may include incidents such as system alarms, electrical trips, chute blockage, etc. Depending on the specific machine considered, trend data typically recorded may include:

- relative position of the machine
- long travel speed
- electrical current drawn by long travel drives
- boom luffing inclination
- production throughput rate

- boom slewing angle
- electrical current drawn by boom conveyor belt
- various hydraulic pressures
- wind speed.

A copy of trend data must be made available to the relevant members of the investigation team and extracted from the system with representation from the OEM and operations management present. In addition to the trend data, a snapshot copy of the SCADA control program utilised at the time of collapse must be stored electronically for investigation purposes. Although the review and analysis of the control program can be done at a later stage, the capturing of the software version in use at the time of the collapse is considered a time-based investigation element which must be done before changes can be done by any party. Representatives from the owner and OEM should be present during this activity to cover both parties and to eliminate potential disputes.

Depending on the nature of the failure, it may be required to revisit the control philosophy document and evaluate whether the practical implementation thereof was indeed sound.

4.10 STRUCTURAL DESIGN AUDIT

Once the prevailing loading conditions at the time of the collapse are understood, the failure postulations may indicate that design aspects of certain critical parts of the structure need to be reviewed. If defects are found, a complete design audit is advisable. It is the author's experience that, should non-compliance to design standards be identified in the design of members directly associated with the collapse, a high probability of non-compliance is to be expected in other aspects of the design.

Design compliance with the applicable standards must be validated. It must nevertheless be understood that deviation from the relevant design standard does not necessarily explain why the member failed. The ultimate load-bearing capacity of members or parts in question must be understood before formulating conclusions.

4.11 METALLURGICAL INVESTIGATION

General observations or the outcome of the structural design audit may highlight the need for a metallurgical examination of specific members or fasteners.

It must be noted that fracture planes of fasteners and structural members may rapidly corrode if left exposed to the elements. Critical evidence may be jeopardised if specialised examination is no longer possible because of deteriorated surfaces. This indicates that there may sometimes (or even often) be important timelines that must be met in order to complete a valid investigation.

It is essential to have written consensus from the OEM and the operation's management that samples prepared for metallurgical examination are representative and that the testing laboratory is deemed suitable for the scope of work anticipated. If this is not done, the results may be disputed at a later stage. It may then not be possible to repeat test work due to the unavailability of test specimens.

4.12 AD HOC TESTING

Various types of examination or testing may be required, depending on the nature of the failure. Thickness measurements may be required to confirm that structural members are in line with what was specified in the certified design. Where corrosion, abrasion or wear is relevant, it may be appropriate to verify the extent of material loss incurred. Non-destructive testing, or, in extreme cases, destructive laboratory testing may be required to prove the competence of critical components, e.g. connections.

4.13 COMMISSIONING AND FABRICATION DOCUMENTATION

Data books used to record the performance testing and final acceptance of the machine or equipment must be reviewed unless deemed unrelated to the failure. Of specific importance is the documentation relating to the control systems and specific acceptance of alarm levels, limits and pre-determined electrical trip-out settings.

The author found that often, due to the fast-track nature of mining and minerals projects, equipment is not commissioned at peak load capacity. Although there could be many reasons for this, high-capacity systems require a significant amount of raw material to be operated continuously for, say, an eight-hour period at peak capacity to satisfy performance criteria. The amount of raw material is often not available early on in the life of the new operation since capacity is generally steadily ramped up towards the target production figures over a period of time, which could exceed a year. When the required raw material is eventually available, the commissioning engineers have usually moved on to other projects. The operational personnel may not be aware of outstanding matters and continue to operate already functional equipment, gradually increasing throughput capacity. It is therefore essential that outstanding performance testing is documented and addressed by the operations team when appropriate. This is considered a project management function which should specifically be reported on during project handover.

Fabrication data books, used for quality assurance purposes, may need to be consulted where inappropriate manufacturing practices or material quality is suspected.

4.14 OPERATOR ABUSE

Depending on the nature of the collapse, it may be required to investigate the possibility of incorrect or inappropriate operational practices. Machine protection systems must nevertheless be adequate to protect the machine under normal and foreseen abnormal operational conditions. What seems to be common sense to the machine designer is not necessarily at all clear to the machine operator. It may be required to evaluate the competence of operators and to review the adequacy of training manuals.

4.15 AD HOC AUDITING OF SUB-SYSTEMS

It may be required to conduct independent audits on sub-systems supporting the functioning of the machine. Detailed findings must be presented in formal reports. Sub-systems may include hydraulic, instrumentation and machine protection systems. This investigation activity is also deemed as time dependent, and must be done in conjunction with representation from the owner and OEM to eliminate disputes at a later stage in the investigation or during litigation.

4.16 ADDITIONAL INFORMATION

Routine maintenance and inspection reports may prove to be a valuable source of information to provide insight into the cause or contributing factors that led to the collapse.

4.17 SPECIALIST CONSULTATION

An independent consultation and review of the investigation may be valuable, especially where the outcome of the investigation is not conclusive. A fresh set of eyes and unbiased opinion from very experienced specialists may proof to be invaluable.

4.18 IDENTICAL OR SIMILAR EQUIPMENT

Depending on the nature of the collapse, it may be required to modify design or control programs on similar equipment used within the organisation or operation. The detailed inspection of similar equipment in view of the collapse is imperative.

4.19 LEGAL

Where design or construction defects can be proven, legal action may ultimately be considered unless an agreement can be reached with the OEM.

Litigation will usually be very costly and involves a drawn-out process. The nature of the legal process is counterproductive and will usually result in damaged relationships. It must be understood that the judge of the court, (who is non-technical) will find it very difficult to determine which party was in the wrong, especially when dealing with a very complex failure. Where the failure occurred because of various contributing factors and different parties are to blame, the probability of a fair outcome is considered to be low. Litigation should therefore only be undertaken as a last resort and where a strong case can be presented to the court. It must however be kept in mind that, regardless of the merits of the case, litigation involves risk without any guarantee of a favourable outcome.

4.20 TIMELINE

It is usually required that investigations are concluded as soon as possible so that settlements can be negotiated where applicable. The nature of the root cause of failure may also have an impact on the amount paid out by the insurer. Although all investigation activities do not require the same urgency, some of the items indicated above are time dependent if the best investigation outcome is to be obtained. The schedule provided below in Figure 4.1 provides an indication of the relative timing of investigation activities.

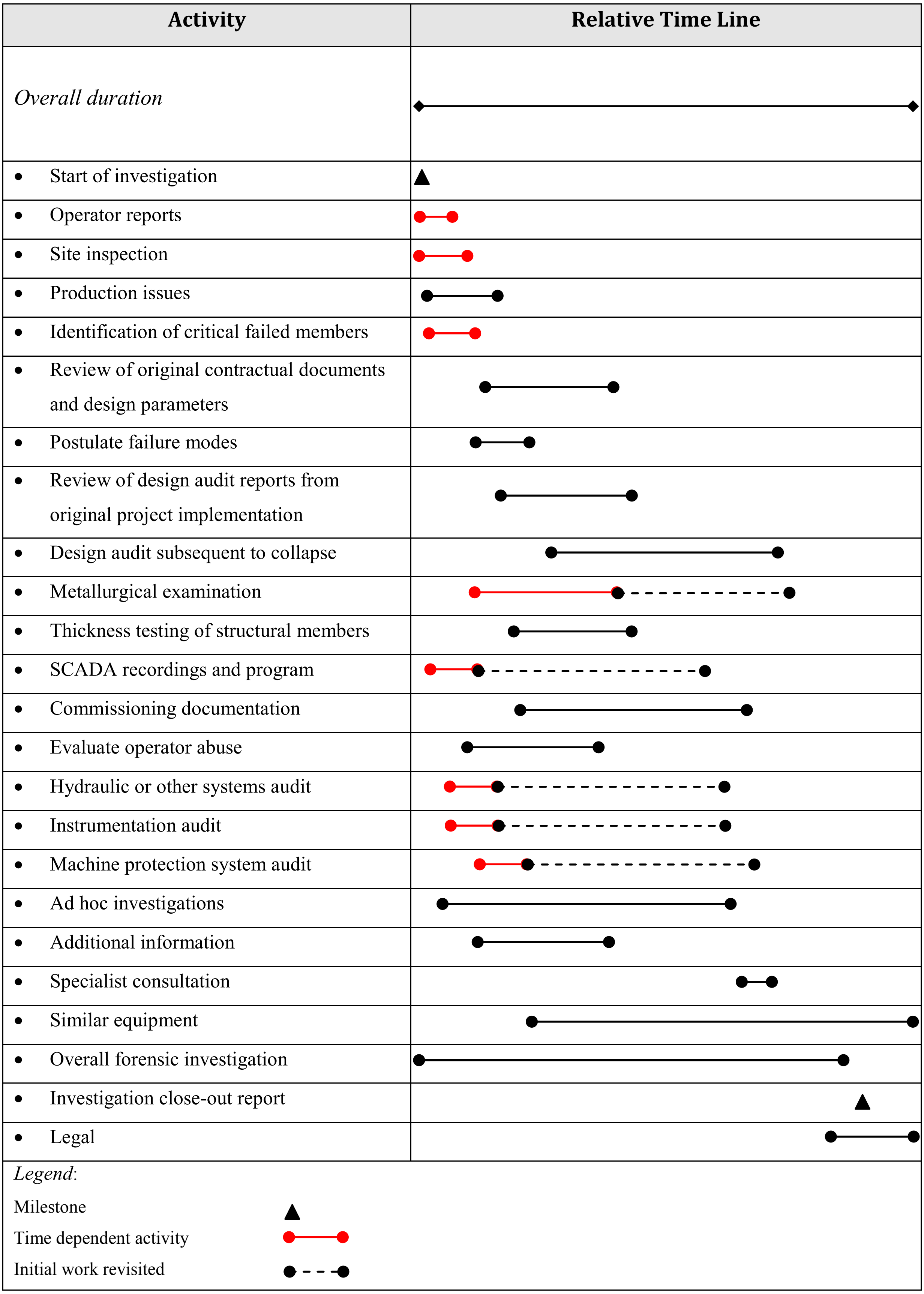


Figure 4.1: Indicative investigation timelines.

4.21 REVIEW OF CASE STUDIES

Chapter 4 was developed by taking a retrospective view of the main case study discussed in the following chapter. In this section the brief case studies already discussed were reviewed in order to demonstrate which investigation elements are most likely to be employed during failure investigations. Table 4.1 lists the summary of expected investigation activities and links to discussion points in preceding sections of this chapter.

Table 4.1: Summary of investigation activities for brief case studies.

Activity	Spreader	Portal Reclaimer	Ship loader	Drum Reclaimer
Start of investigation	√	√	√	√
Operator reports	√	√	√	√
Site inspection	√	√	√	√
Production issues	√	√	√	√
Identification of critical failed members	√	√	√	√
Review of original contractual documents and design parameters	√	√	√	√
Postulate failure modes	x	x	√	x
Review of design audit reports from original project implementation	x	x	x	x
Design audit subsequent to collapse	√	√	√	√
Metallurgical examination	x	x	√	x
Thickness testing (member size compliance or corrosion)	√	x	√	x
SCADA recordings	x	√	√	√
Commissioning documentation	x	√	x	x
Fabrication data books	x	x	x	x
Evaluate operator abuse	√	√	√	√
Hydraulic or other systems audit	x	x	x	x
Instrumentation audit	x	√	√	√
Machine protection system audit	x	√	x	√
Ad hoc investigations	√	√	√	√
Additional information	√	√	√	√
Specialist consultation	√	√	√	√
Similar equipment	x	√	x	√
Overall forensic investigation	x	√	√	√
Compilation of final close-out report	√	√	√	√
Legal	√	√	√	√

4.22 INVESTIGATION TEAM

Investigation teams will invariably not be made up of the same elements. Key roles, which the author deems essential for a successful failure investigation, i.e. to establish the root cause of the collapse, are nevertheless proposed as a guideline. Although investigation activities are bound to overlap between roles, the author is of the opinion that predefined roles will enhance the accountability of individuals and assist in delivering a successful investigation. From an owner's perspective, a typical investigation team should, as a minimum, consist of the roles illustrated below in Figure 4.2.

Figure 4.2: Model for a typical investigation team.

It is essential that the owner appoints a forensic investigator right at the beginning of the investigation. Although it may take some time to find a suitable consultant, it is envisaged that the role be filled by a senior manager from the operation, or a suitable candidate from the investigation team until an appointment can be made. Collaboration between all members is essential for success and informal sharing of knowledge and findings is encouraged. Formal communication through the appointed forensic investigator is proposed. In most cases, especially where a collapse of a recently commissioned machine occurs, the OEM will carry out a self-governing investigation in parallel with the owner's investigation. The level of experience of team members with regard to previous investigations will not be the same, hence the prominence of team members in certain activities may vary. The skill set of the appointed forensic investigator will determine to what level certain specialised functions will be handled by discipline-specific experts. A guideline for key functions of team members is nonetheless proposed in Table 4.2 below. Activities once again link to discussion points in the preceding sections of this chapter.

Table 4.2: Guideline for key responsibilities.

Activity	Owner	Forensic Investigator	Lead Mechanical / Structural	Lead Electrical & Instrumentation
Initiate investigation kick-off	√			
Notify Insurer	√			
Operator reports		√		
Facilitates site inspection	√			
Facilitates production issues	√			
Identification of critical failed members		√	√	
Review of original contractual documents and design parameters		√	√	
Postulate failure modes		√	√	
Review of design audit reports from original project implementation			√	
Facilitate structural design audit subsequent to collapse			√	
Facilitate metallurgical examination			√	
Facilitate thickness testing (member size compliance or corrosion)			√	
Interpret SCADA recordings		√	√	√
Review commissioning documentation		√	√	√
Review fabrication data books			√	
Evaluate operator abuse	√	√	√	√
Facilitate hydraulic or other systems audit			√	
Facilitate electrical / instrumentation audit				√
Machine protection system audit		√	√	√
Facilitate ad hoc investigations		√		
Additional information	√			
Facilitate specialist consultation		√		
Similar equipment	√			
Compilation of final close-out report		√		
Legal	√			

4.23 RESPONSIBILITY MATRIX

Depending on the complexity of the failure investigation, a responsibility matrix may be developed to clearly define the roles and responsibilities of the investigation team members. The matrix shown in Table 4.3 below was compiled to demonstrate the author's input and responsibility during and after the investigation of the case study failure discussed in Chapter 5. The activities listed link to the discussion points above.

Table 4.3: Responsibility matrix for investigation activities.

Activity	Input	Oversight	Responsible
Start of investigation	√		
Operator reports		√	
Site inspection	√		
Production issues	√	√	
Identification of critical failed members		√	
Review of original contractual documents and design parameters			√
Postulate failure modes		√	√
Review of design audit reports		x	√
Structural design audit		√	√
Metallurgical investigation		√	√
Destructive testing		√	
SCADA recordings			√
Commissioning documentation			√
Operator abuse			√
Hydraulic system audit		√	
Instrumentation audit		√	
Machine protection system audit		√	
Ad hoc investigation - fasteners			√
Additional information		√	
Specialist consultation	√		
Similar equipment		√	
Overall forensic investigation. (interpretation of findings from task-team investigations)			√
Compilation of final close-out report			√
Legal	√		

Legend:
Input – Activities where the author's input was provided
Oversight – Work was carried out by others under supervision of the author
Responsible – Activities for which the author was primarily responsible

4.24 SUMMARY

A methodology for the successful investigation of catastrophic failures associated with mobile BMH equipment was proposed. An outline model defining a typical investigation team was provided and key functions for the major role players were defined.

The relevance of the proposed methodology was demonstrated by evaluating probable investigation activities associated with the case studies covered in Chapter 3.

The author’s role in the main case study, which will be discussed in Chapter 5, was illustrated by means of a responsibility matrix.

5 CATASTROPHIC COLLAPSE OF A SLEWING STACKER

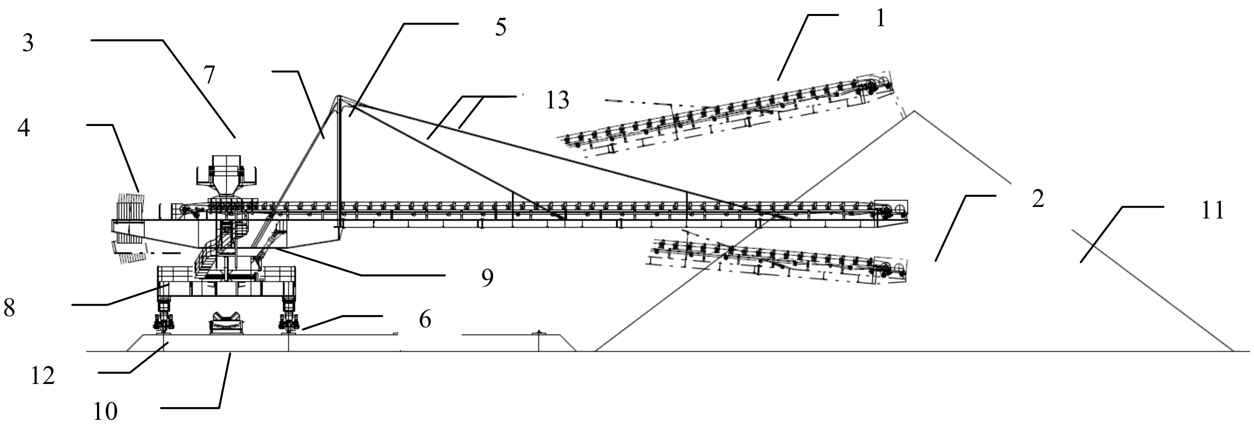
5.1 OVERVIEW

This chapter follows the investigation methodology outline provided in Chapter 4. The details of the main failure case study are discussed systematically in order to obtain the root cause of failure. It must be understood that multiple sub-investigations were carried out in parallel to the main investigation. It was only possible to arrive at a plausible conclusion at the end of the overall investigation once the key findings from various task teams could be interpreted as a whole. Detailed investigative work done, but which did not contribute to the final outcome, has been omitted from the study. Only key findings are discussed in the relevant sections.

General design terminology, as well as terms specifically used in the mining and minerals industry, are provided in Appendix A.

5.2 MACHINE DESCRIPTION

The construction of a typical slewing stacker is illustrated in Figure 5.1 below.



1 - Stacker boom, conveyor, raised
2 - Stacker boom, lowered position
3 - Tripper car
4 - Ballast mass
5 - Pylon
6 - Bogie
7 - Tie beam(s)
8 - Under carriage
9 - Luffing cylinder
10 - Yard conveyor
11 - Stockpile
12 - Rails for long travel
13 - Tie rods

Figure 5.1: Slewing stacker, annotated.

A general description of the slewing stacker and main components follows:

- The tripper car straddles the yard conveyor belt and is interlinked with the stacker carriage.
- The stacker and tripper car are rail mounted and travel as an assembly along the stockpile bed as stacking operations are carried out.
- The stacker boom, supporting the boom conveyor belt, is essentially a balanced beam which utilises ballast mass to counteract the weight of the boom structure and the material load. A tie-rod and tie-beam configuration is utilised in conjunction with a pylon structure to reduce internal moments within the boom superstructure and to reduce boom deflection. Note that tie-beams are connected at each side of the pylon i.e. to the eastern and western side with the boom in normal stacking position. Only a two-dimensional diagram was provided in Figure 5.1 above.
- The stability of the machine is determined by the location of the centre of gravity of the boom, which moves longitudinally along the boom as the material loading varies. The optimum ballast mass is a trade-off consideration between a material overload scenario and the weight of the boom without any material loading present.
- The stacker boom can be luffed up or down by means of a hydraulic cylinder in a controlled manner to reduce material degradation and dust generation when the stockpile is being built.
- The tripper facilitates the feed of bulk material from the yard conveyor belt onto the stacker boom belt. Bulk material is ultimately discharged at the head end of the machine onto the longitudinal stockpile.
- Boom belts are prone to slippage when high throughput capacities are handled, especially when the boom is luffed at a positive inclination. The drive power that can be transferred is a function of the coefficient of friction between the conveyor belt and the drive pulley. Sufficient take-up tension is required within the boom conveyor belt to facilitate the transfer of drive power. When boom belt slippage occurs, the belt fill factor will increase if the feed tonnage from the yard conveyor belt is maintained. Belt slippage can be detected by monitoring the belt speed with suitable instrumentation.
- Sideward slew motion of the boom is facilitated by a motorised revolving frame with a thrust bearing arrangement.
- The revolving frame is mounted on top of an undercarriage which is supported by balanced bogie wheel sets.
- Although the specific stacking procedure is generally determined by process specific requirements, material is essentially stacked onto longitudinal piles as the machine travels along the stockyard.

- An automatic stacking operation is preferable, but manual operation is possible.
- Material is recovered from the stockpile bed by means of a reclaimer and associated conveyor belt systems.

5.3 BACKGROUND

The machine was successfully operated for approximately a year before collapsing completely. An incident in which the boom conveyor belt was overloaded preceded the failure. A tie-back system, with a combination of tie rods and beams, was incorporated in the design to reduce the overall mass of the boom structure and to limit static deflection. Refer to annotations 13 and 7 in Figure 5.1 above. The failure of a critical tie-beam connection initiated the collapse of the boom and ultimately ruined the entire machine. The extent of the damage can be seen in Figures 5.2 and 5.3 below.

Figure 5.2: Collapsed stacker viewed from the north.

Figure 5.3: Collapsed stacker viewed from the south.

5.4 TIMELINES

It must be recognised that the nature of failure investigations are such that various matters need to be investigated concurrently. An attempt is nevertheless made to present this chapter in a logical and chronological order as far as practically possible. A typical outline investigation program schedule was provided in Chapter 4 as part of the investigation methodology and should be referred to in order to gain a better understanding of the timelines associated with respective investigation activities.

5.5 START OF INVESTIGATION

Immediately after the failure, an investigation team was assembled with representatives from the owner, operator, the OEM, and third-party consultants. The author played a leading role, taking responsibility for the oversight of all aspects related to mechanical and structural engineering. Refer to Section 4.23 Table 4.3 for details.

5.6 OPERATOR REPORTS

Based on the eye-witness report of a shift operator who was standing at the tip of the boom, the most likely sequence of collapse was as follows:

1. Overload of the stacker was detected and the boom belt stopped.
2. Coal had to be cleaned off the conveyor to reduce the load in order to initiate the restart.
3. The boom conveyor was restarted and ran for a short period, then stopped.
4. The control room was requested to restart the belt again.

5. The operator perceived an upward movement of the boom before the collapse.

It must also be noted that:

1. A faulty inclinometer (device which detects the luffing angle of the stacker boom) was detected and had to be replaced several hours prior to the collapse.
2. There was a malfunctioning of the luffing hydraulic system which manifested as a vertical oscillation motion of the stacker boom tip. Uncontrolled raising and lowering of the boom was observed without human intervention.

5.7 IDENTIFICATION OF FAILED MEMBERS

Comprehensive plans were devised to clear the site and to construct temporary works to minimise production losses. The specific details are irrelevant to the nature of the study, but it is sufficient to state that the collapsed machine had to be removed to allow production activities to commence.

High-resolution images of the collapsed machine and site surroundings were taken from various angles. From team discussions, it was anticipated that the failed hydraulic cylinder clevis, shown below in Figure 5.4, and the tie-beam connection, shown in Figure 5.5, could possibly provide critical clues which would lead to understanding the root cause of the failure.

Figure 5.4: Failed cylinder clevis half.

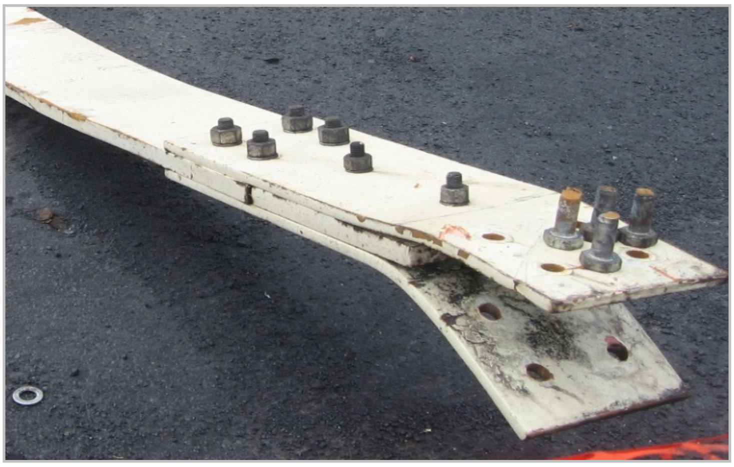


Figure 5.5: Failed tie-beam connection, western side of machine in normal stacking position.

5.8 FAILURE HYPOTHESIS

Subsequent to detailed site inspections, preliminary calculations and team discussions, two main failure mechanisms were postulated as depicted in Figure 5.6 below.

1. Clevis failure of the luffing cylinder
2. Tie-beam connection failure

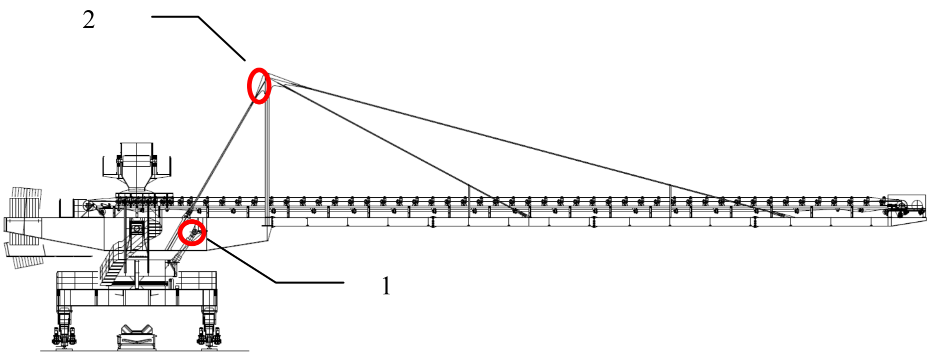


Figure 5.6: Critical components associated with failure hypothesis.

5.8.1 CLEVIS FAILURE OF THE LUFFING CYLINDER

An overview of the first failure hypothesis which focuses on the failure of the clevis of the hydraulic luffing cylinder is provided below.

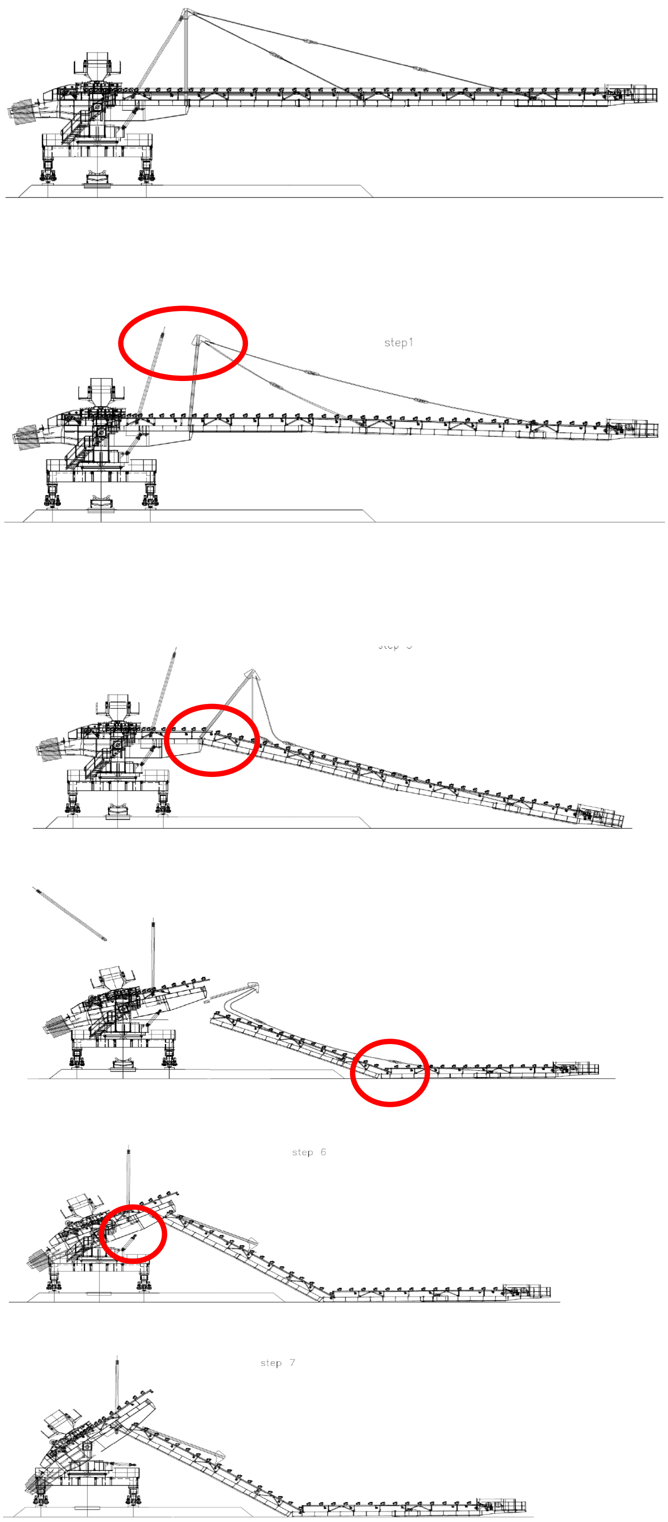
At commencement of the investigation, the OEM advised that the luffing cylinder should be under a compressive load under all normal operating conditions. The maximum operating static tensile load will occur with an empty boom belt which was the state of the system at the time of collapse. Considering a cylinder clevis failure, the following behaviour of the superstructure could be expected:

- The boom would rotate upwards due to the counterweight acting under gravity.
- The tie beams would crash into the superstructure of the tripper car.
- Note that the machine operator perceived an upward movement of the boom at the time of the collapse.
- Tie-beam connections are expected to fail in shear as a consequence of impact against the tripper car structure.
- The boom would drop down under gravity when the tie-beam connections are lost.
- The counterweight mass would drop down resulting in a boom failure at the main connection as shown in Figure 5.7 below.

Figure 5.7: Collapsed machine with luffing cylinder and main boom connection highlighted.

5.8.2 TIE-BEAM CONNECTION FAILURE

In this hypothesis, it is anticipated that a failure at either of the two tie-beam connections, would lead to an impact load on the adjacent tie-beam member and hence the collapse of the boom. A graphic illustration of the sequence of events is provided in Figure 5.8 below. Attention is drawn to the criticality of the tie-beam elements and associated connections.



Prior to failure

Tie-beam connection fails

Main boom moment connection fails

Main boom intermediate connection fails
Southern tie-beam dislocates

Cylinder clevis fails

Ballast drops under gravity and drags boom towards the carriage

Figure 5.8: Graphic illustration of the 2nd failure hypothesis (19).

The above representation was found to be consistent with site observations.

5.9 CONTRACTUAL DOCUMENTS

The author reviewed the original contract between the OEM and the employer for the design, manufacturing, construction, commissioning and training of operations personnel (11).
Key findings from the investigation follow:

- The contract required that the design of the equipment was to be conducted in accordance with ISO 5049-1 (1994).
- The employer specifically stipulated the requirement for double redundancy on stationary tie systems i.e. a single tie-beam, as already discussed, had to be capable of carrying the entire boom load. The contract made provision for a reduced safety factor in view of double redundancy.
- The employer specifically stated that the boom belt should be designed for flooded belt conditions.
- The stockyard equipment for both the raw plant feed and the beneficiated product was covered under the same contract. The respective bulk densities for the two materials handled at the separate stockyards are approximately 1100 and 850 kg/m^3 respectively. The stackers provided under the contract were identical and designed for *850 kg/m^3*, thus matching the beneficiated product bulk density. The failed stacker was situated at the raw plant feed stockyard and consistently handled *raw plant feed material of 1100 kg/m^3*.

5.10 ORIGINAL DESIGN AUDIT DOCUMENTATION

A third-party design audit was initiated by the implementation team of the original capital project for purposes of oversight and assurance. This investigation was completed approximately eight months prior to the collapse. The author established from the audit report that the original third-party auditor "found that the structural design was adequate in all respects" (21).

5.11 STRUCTURAL DESIGN REVIEW

The author, in conjunction with an appointed third-party consultant and OEM designers, conducted a comprehensive design review of the machine. The validation of applied loads, load factors, load combinations and boundary conditions as prescribed by the normative design standard, i.e. ISO 5049-1 (1994), was the key focus of the structural design review. The design loads from various load cases used in calculations reflected in this chapter were obtained from the OEM designer's analysis output model.

The OEM designed structural members by applying DIN18800-1 (1990) while the loading was obtained by the rules stipulated in ISO 5049-1 (1994).

Only calculations directly related to the findings of the investigation are reflected in this document. All calculations provided in Section 5.11 were carried out by the author. *Note that the author is not in favour of a design approach which combines an allowable stress standard i.e. ISO 5049-1 (1994) with a limit states design standard i.e. DIN 18800-1 (1990) since the basis of these standards is different.* This aspect is revisited in Chapter 6.

5.11.1 LOADING

Material loading on boom belt – Since the incorrect material bulk density was used in the original design analysis model, as explained in Section 5.9 above, the material and encrustation loads were underestimated by a factor of 1,3 as shown below in Equation (1):

$$\begin{aligned} \rho_{\text{ratio}} &= \frac{\rho_{\text{actual}}}{\rho_{\text{design}}} \\ &= \frac{1100\ \text{kg/m}^3}{850\ \text{kg/m}^3} \\ &= 1{,}3 \end{aligned} \qquad (1)$$

Where:

ρ_{actual} = realistic bulk density [kg/m^3]

ρ_{design} = bulk density used for design [kg/m^3]

Upon review of the loading parameters, it was found that the design was based on a boom conveyor loading of 2000 ton/hr. However, ISO 5049-1 (1994), clause 3.1.2.1.1 b, states that “Where there is no capacity limiter, the design capacity is that resulting from *the maximum cross-sectional area* of the conveyor multiplied by the conveying speed ...” Flooded belt conditions were therefore to be catered for in this particular instance, which implies an equivalent belt loading capacity of approximately 4000 ton/hr as calculated by the author (12). The applicable standard for these calculations is ISO 5048 (1989), Continuous mechanical handling equipment, belt conveyors with carrying idlers, calculation of operating power and tensile forces. The requirement to assess belt loading with flooded belt conditions was also specified in the contract documentation.

Considering Equation (1) above, the original design therefore underestimated the material loading on the belt by a factor of 2,6 as shown below in Equation (2):

$$\text{Material loading ratio} = \frac{\rho_{\text{ratio}} \text{ x flooded belt loading}}{\text{Peak design belt loading}} \qquad (2)$$

$$= \frac{1{,}3 \text{ x } 4\ 000 \frac{t}{hr}}{2\ 000 \frac{t}{hr}}$$

$$= 2{,}6$$

Although this is a significant error, the unit weight per metre of boom structure and mechanical equipment, such as the conveyor belt, idlers, pulleys and plate work, is much greater than the material load. The implications of this error are discussed in greater detail in Sections 5.13.9 and 5.14.

5.11.2 BOOM BELT LOADING, OTHER FACTORS

The boom belt speed was designed at 2,8 m/s while receiving material from a yard conveyor running at 3,8 m/s. It is common engineering practice to match the belt speeds of feed and receiving conveyors, especially where belt widths match. The mismatch of belt speeds would worsen the boom belt loading conditions, especially when boom belt slippage occurred. Also refer to Section 5.14.

The above discussion concludes the major findings pertaining to the review of the design loadings used for the stacker.

5.12 FAILURE HYPOTHESIS 1 – CYLINDER CLEVIS FAILURE

This section explores the details related to the first postulated failure mechanism i.e. the failure of the clevis of the hydraulic luffing cylinder. Also refer to Section 5.8.

If a clevis failure under overload conditions can be proven, it would be possible to eliminate the first failure postulate which is based on a defective or inadequate clevis capacity. Both failure postulates are nevertheless based on a cylinder clevis failure under a direct tensile load. In the second failure postulate, however, the cylinder clevis failure did not cause the collapse. Images obtained from the site do not provide any reason to suspect a compression failure. The fracture plane is shown in Figure 5.9 below.

Figure 5.9: Cylinder clevis fracture plane, lubrication channel noticeable.

5.12.1 METALLURGICAL INVESTIGATION – CYLINDER CLEVIS

The need for a metallurgical investigation of the luffing cylinder clevis was identified and initiated during initial group discussions. Detailed metallurgical examination by specialists, however, revealed no abnormalities in the clevis casting (15). The material conformed to the ASTM A536 60-45-10 which is a ductile cast iron. The relevant material properties are shown below in Table 5.1. *(Blue shaded table values denote figures which will be referred to later in the document.)*

Table 5.1: Material properties of clevis.

Property	
S_{ut}	487 MPa
σ_y	353 MPa
BHN	160
Where: $S_{ut} =$ Ultimate tensile strength (MPa) $\sigma_y =$ Yield strength (MPa) BHN = Brinell hardness number	

An extract from the key findings of the metallurgical report (15) is given below:

- "The microstructure of the clevis consisted predominantly of ferrite with nodular graphite and some pearlite."

- "The silicon content of the clevis of 2,97 % is relatively high which could decrease the materials impact toughness. The small amount of pearlite present within the microstructure could also decrease the impact toughness compared to a fully ferritic structure. The impact toughness of the clevis, although lower than that of fully ferritic ductile cast iron, is still relatively high, and therefore is probably not the root cause of failure, as indicated by the ductile fracture mode at the origin."
- "The hardness and tensile tests indicated that the clevis conforms to ASTM A536 60-45-10 specification for ductile cast iron."
- "Scanning electron microscope analysis showed that the clevis experienced a slower strain rate (slower than during tensile testing) during the crack initiation which accelerated as the crack progressed, as indicated by the transition from a ductile to brittle fracture mode."
- "No indications of a pre-existing defect or fatigue crack was found at the origin."
- "The clevis failed by overload. The clevis did not fail under impact conditions and the influence of both the silicon and pearlite are probably not significant."

5.12.2 CYLINDER CLEVIS LOAD CAPACITY

Considering that the material properties have been established, as discussed in Section 5.12.1 above, it is required to calculate the ultimate load-bearing capacity of the cylinder clevis to determine if the design was adequate. From OEM cylinder drawings, the effective area of the fracture plane as shown in Figure 5.9 can be approximated as shown below in Equation (3):

$$\begin{aligned} A_{ne} &= \text{Clevis width x sum of clevis side thicknesses} \\ &= 51 \text{ x } (20 + 26) \text{x } 10^{-6} \text{ m}^2 \\ &= 2{,}346 \text{ x } 10^{-3} \text{ m}^2 \end{aligned} \tag{3}$$

Where:

A_{ne} = Effective area of the fracture plane [m^2]

The failure load in tension can therefore be calculated as 1142,5 kN as shown below in Equation (4):

$$F_{failure} = S_{ut} \times A_{ne} \qquad (4)$$
$$= 487 \times 10^6 \text{ Pa} \times 2.34 \times 10^6 \text{ m}^2$$
$$= 1142{,}5 \text{ kN}$$

Where:

$F_{failure}$ = Failure load of the clevis under tension [N]

S_{ut} = Ultimate tensile strength [Pa]

A_{ne} = Effective area of the fracture plane [m^2]

The stacker boom is effectively a balanced beam. The luffing cylinder predominantly acts as a compression member. In conjunction with the OEM designers, it was established that under empty boom conditions, a cylinder tension force of 437 kN is required to maintain system equilibrium. This is a substantial force which highlights the sensitivity of the relative position and magnitude of the ballast mass. The reader's attention is drawn to the fact that the magnitude of the cylinder tension will decrease as the belt loading increases. The actual weight of the ballast mass was optimised during commissioning. Excessive ballast mass will result in extreme cylinder tension force. Although the OEM increased the ballast mass during commissioning, the overall system balance was deemed acceptable. The value of 437 kN is therefore considered the worst cylinder tension loading condition and was obtained from the OEM's adjusted analysis model.

The clevis failure load of 1142,5 kN, as calculated above in Equation (4), is significantly greater than the normal operational load. The factor of safety against failure of the clevis was calculated at 2,6 as shown below in Equation (5):

$$\text{Factor of safety} = \frac{F_{failure}}{F_{ultimate}} \qquad (5)$$
$$= \frac{1142.5 \text{ kN}}{437 \text{ kN}}$$
$$= 2{,}6$$

Where:

$F_{failure}$ = Failure load of the clevis under tension [N]

$F_{ultimate}$ = Ultimate cylinder tensile loading [N]

Note that the luffing cylinder and tie-beam forces are interdependent, but altogether different. These values are not to be confused.

Since calculations and metallurgical examination clearly indicate that a severe overload condition caused the failure of the cylinder clevis, the first failure hypothesis can be eliminated.

5.13 FAILURE HYPOTHESIS 2 – TIE-BEAM CONNECTION FAILURE

This section explores the details relating to the second postulated failure mechanism i.e. the failure of the tie-beam connection on either the eastern or western side of the boom.

Calculations provided in Section 5.13 were recalculated from the investigation report originally prepared by an appointed professional consultant (12) with oversight by the author.

The tie-beam elements are depicted in Figure 5.10 below while the connection detail is shown in Figure 5.11.

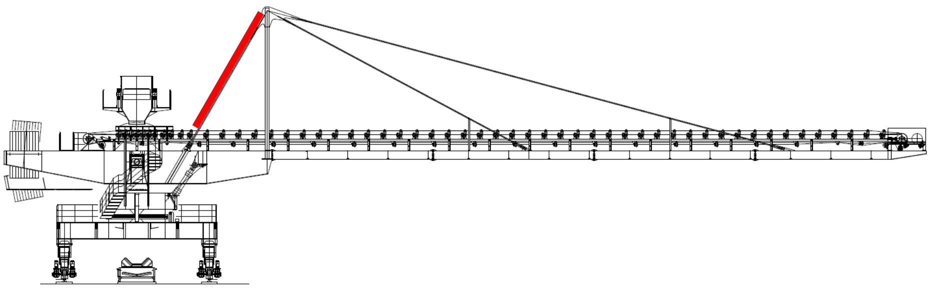


Figure 5.10: Tie-beam elements highlighted.

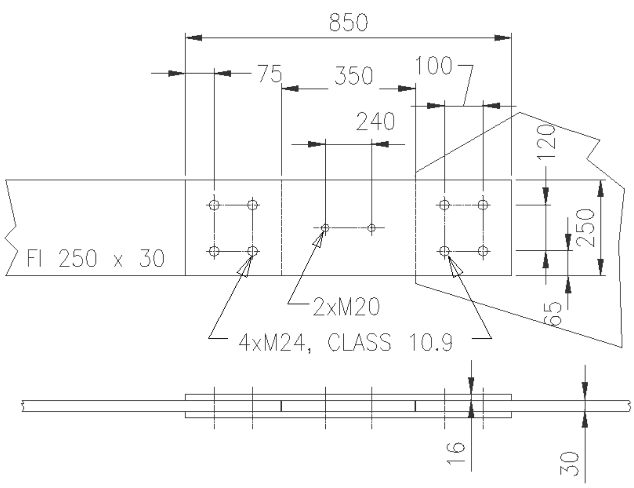


Figure 5.11: Tie-beam connection.

5.13.1 FASTENERS, TIE-BEAM CONNECTION

The tie-beams and associated connections are primary load bearing elements which are subjected to cyclic loading and moderate impact due to the variable service loading conditions and boom luffing actions.

ISO 5049-1 (1994) (38), clause 6.2.1 and 6.2.2 require that bolt shanks must bear against the full length of the hole and prohibit the use of non-fitted bolts on primary load bearing-members. DIN 18800-1 (1990) (27), clause 8.2.1.1 makes provision for designing a connection with a shear plane intercepting the threaded part of a bolt. Clause 8.2.1.5 of the same standard, however, prohibits such a design in fatigue shear loading applications. The use of this fastener configuration on critical connections is considered poor engineering practice since the fastener thread within the shear plane introduces a stress concentration on a critically loaded element. This topic is discussed further under Section 5.13.5 and 5.13.11.

High strength, Class 10.9, electroplated fasteners were utilised in the tie-beam connections.

5.13.2 DESIGN VALIDATION, TIE-BEAM CONNECTIONS

The validation of the connection design was done by the author in line with the procedures detailed in ISO 5049-1 (1994) and FEM SECTION II (1992). Unless otherwise noted, the factors of safety on the elastic limit of the material in question, as outlined in the above-mentioned standards, are as shown below in Table 5.2.

Table 5.2: ISO 5049-1, Factors of safety on elastic limit.

	Load Case I	Load Case II	Load Case III
σ_a	$\frac{R_{p0.2}}{1.5}$	$\frac{R_{p0.2}}{1.33}$	$\frac{R_{p0.2}}{1.2}$
	Where: σ_a = Allowable Stress (Pa) $R_{p0.2}$ = Yield point stress (Pa) Refer to ISO 5049-1 (1994) for load case definitions		

After revising the belt loading and density parameters in line with the findings already discussed above in Section 5.11, the most severe loads were obtained from the OEM's analysis as shown below in Table 5.3.

Table 5.3: Design loads from ISO 5049-1 (12).

Load case	LC I	LC II	LC III
Design load	1194 kN	1210 kN	1268 kN
	Where: LC = Load Case, obtained from ISO 5049-1 (1994), Section 4		

The relevant tension calculations for the tie-beam plate elements and connection gussets were considered, but are not reflected in this document since the shear loading of the bolts was found to govern the connection design. It must be noted that *the bolt bearing capacity of the tie and gusset members was only marginally higher than the shear capacity of the bolts and was also found to be overloaded.* Local yielding could be observed at the bearing surfaces of the plate members as expected for heavily loaded connections equipped with non-fitted bolts. The allowable stress for Class 10.9 bolts was determined from FEM SECTION II (1992) (31) as shown below in Table 5.4.

Table 5.4: Allowable stress for Class 10.9 fasteners, FEM SECTION II (1992) (31).

ISO bolt grade	load case	permissible tensile stress ($0.625\ \sigma_a$)	bolts in fitted holes				bolts in non-fitted holes	
			single shear		double shear			
			permissible shear stress ($0.6\ \sigma_a$)	permissible bearing stress ($1.3\ \sigma_a$)	permissible shear stress ($0.8\ \sigma_a$)	permissible bearing stress ($1.75\ \sigma_a$)	permissible shear stress ($0.5\ \sigma_a$)	permissible bearing stress ($1.0\ \sigma_a$)
		N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2
	I	375	360	780 x	480	1050 x	300	600 x
10.9	II	423	406	880 x	541	1184 x	338	677 x
	III	469	450	975 x	600	1313 x	375	750 x
		Where: x bearing strength exceeds elastic limit of plate material FE 510 i.e. 360 N/mm^2 σ_a = Allowable stress						

The shear capacity of the tie-beam connection can be calculated as shown in Equation (6) below.

$$V_r = \tau_a \, x \, \frac{\pi}{4} \, x \left(D_n{}^2 + D_m{}^2\right) x \, n \qquad (6)$$

Where:

V_r = The shear resistance of the bolt group based on allowable shear stress. [N]

τ_a = The allowable shear stress of the fastener. [Pa]

D_n = The nominal shank diameter of the bolt. [m]

D_m = The minor diameter of the bolt thread. [m] (44)

n = The number of bolts

From Equation (6), the shear resistance of the bolt connection was calculated as shown in Table 5.5 using the permissible stresses from Table 5.4 and design loads from Table 5.3 above.

Table 5.5: Tie-beam connection bolt overload factors.

Description	**Unit**	**LC I**	**LC II**	**LC III**
τ_a, Allowable shear stress (Table 5.4)	MPa	300	338	375
D_n, Bolt size	m	0,024	0,024	0,024
D_m, Bolt thread minor diameter,	m	0,0203	0,0203	0,0203
n, Number of bolts		4	4	4
V_r, Shear resistance to ISO 5049-1	kN	931	1049	1164
Design loads, ISO 5049-1 (Table 5.3)	kN	1194	1210	1268
Factor		1,28	1,15	1,09
Overload factor		**28%**	**15%**	**9%**

The bolts of the tie-beam were shown to be overloaded by a factor of approximately 28% under conditions defined under LC I. It can hence be concluded that the bolt capacity did not comply with the requirements as set out in ISO 5049-1 (1994). From the design audit, it was established that the key reason for this error was utilising incorrect material bulk density and belt loading conditions as discussed in Section 5.11 above.

The bolt capacity will be revisited under Section 5.15 Similar Equipment, which deals with the modified connection details.

5.13.3 FAILURE CAPACITY, TIE-BEAM CONNECTION

The calculations provided in Table 5.4 above evaluated the aspects regarding the *compliance* of the connection with ISO 5049-1 (1994). It is, however, required to determine the actual *failure* *load capacity* of the connection in order to understand the hypothetical belt or boom loading which could cause a collapse to be initiated.

The tie-beam connection, as provided in Figure 5.11 above, is a four-bolted connection, using M 24 Class 10.9 fasteners acting in double shear. Due to the shank length selected in the design, bolt threads are intercepted by a shear plane. The adjacent shear plane acts through the bolt shanks. From first principles, the failure load, acting in shear, of a single fastener can be calculated at 481 kN as shown in Equation (7) below (41):

$$\begin{aligned} F_{failure} &= 0{,}62 \text{ x } S_{ut} \text{ x } A_b \\ &= 0{,}62 \text{ x } 1000\text{x } 10^6 \text{ Pa x } \frac{\pi}{4} \text{ x } (0{,}0203^2 + 0{,}024^2) \text{ m}^2 \\ &= 481 \text{ kN per bolt} \end{aligned} \qquad (7)$$

Where:

$F_{failure}$ = Failure capacity of the bolt under double shear. [N]

S_{ut} = Minimum ultimate tensile strength for a Class 10.9 bolt is 1000 MPa.

A_b = Effective area of the combined shear planes per bolt. [m^2]

The root diameter for an M24 bolt is 20.3 mm (46).

Hence the failure shear capacity for the connection can be calculated as shown below in Equation (8):

$$\begin{aligned} F_{T\,failure} &= F_{failure} \text{ x } n \\ &= 481 \text{ kN x } 4 \\ &= 1925 \text{ kN} \end{aligned} \qquad (8)$$

Where:

$F_{T\,failure}$ = The failure capacity of the connection in double shear. [N]

n = The number of effective bolts per connection.

Considering the ultimate loads listed in Table 5.2, a factor of safety against failure of approximately 1,4 was provided as calculated below in Equation (9):

$$\text{Factor of safety} = \frac{F_{T\ failure}}{F_{design}} \qquad (9)$$

$$= \frac{1925\text{kN}}{1268\ \text{kN}}$$

$$= 1{,}5$$

Where:

$F_{T\ failure}$ = Failure capacity of the connection in double shear. [N]

F_{design} = Worst case design load as obtained from ISO 5049-1 load cases. [N]

The above calculations are therefore not conclusive in explaining why the tie-beam connection failed. Also refer to Section 5.13.10.

5.13.4 DESTRUCTIVE TESTING

Destructive tensile tests were conducted in a test laboratory to support the calculations done to determine the failure capacity of the tie-beam connections (19). Due to capacity constraints on the test machine, representative down-scaled connections were developed. The machine's tie-beam connections had 8 bolts in total per connection i.e. 4 bolts per side. These connections were replicated using 2 and 4 bolted configurations with identical fasteners, material properties and plate thicknesses as the original connection, as shown below in Figure 5.12.

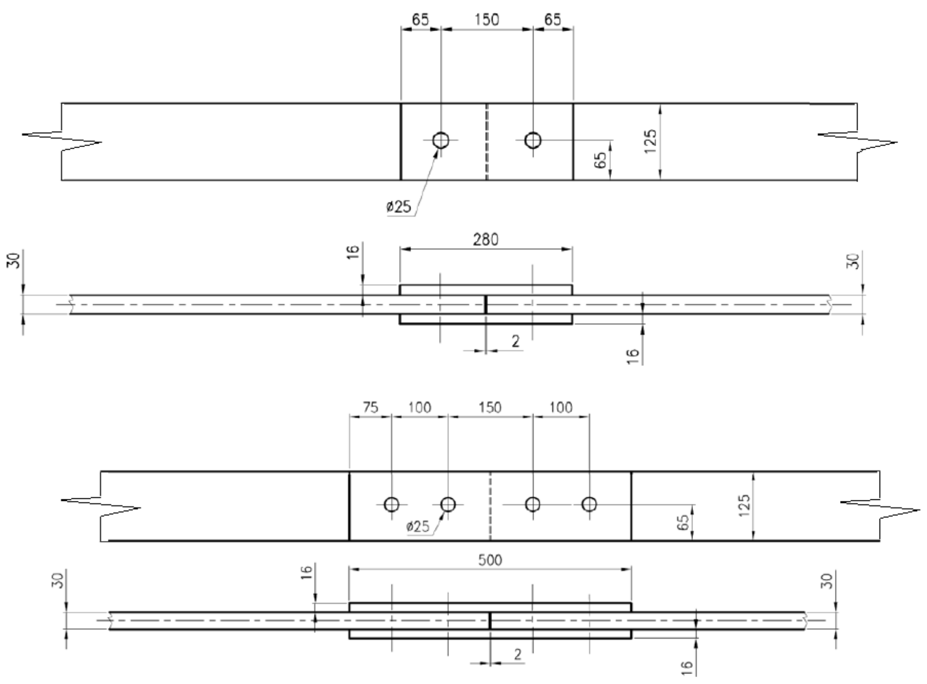


Figure 5.12: Down-scaled connections used for destructive testing.

A total of 24 destructive test samples were prepared using three types of bolt configurations as shown below in Figure 5.13:

Configuration 1 – bolts with shanks within both shear planes.
Configuration 2 – threaded bolts with threads intercepted by the shear planes on both sides.
Configuration 3 – bolts with shanks and threaded part intercepted within the respective shear planes.

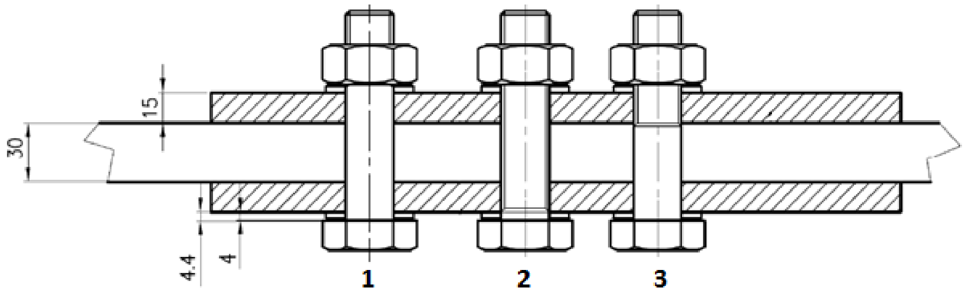


Figure 5.13: Bolt configurations for test specimens.

All surfaces were lubricated and pre-tension torque settings of 250 and 500 Nm were used on respective identical samples. Although the turn of the nut tightening method is recommended by SAISC, a pre-tension torque setting is considered acceptable since tests were carried out in controlled laboratory conditions. Two samples of each test configuration were tested. Table 5.6 below shows available results from the OEM (19).

Table 5.6: Available results for destructive tests.

Bolt configuration	Connection type	Pre-tension (Nm)	Representative Failure Load (kN)
1	2 bolt	250	-
1	2 bolt	500	-
1	4 bolt	250	-
1	4 bolt	500	2 000
2	2 bolt	250	1 660
2	2 bolt	500	-
2	4 bolt	250	1 500
2	4 bolt	500	1 600
3	2 bolt	250	-
3	2 bolt	500	1 840
3	4 bolt	250	1 900
3	4 bolt	500	-
Average failure load for configuration 3 (kN)			*1870*

Shown below in Figure 5.14 are images of 4 bolted connections (i.e. 2 bolts per side) after the laboratory tensile test.

Figure 5.14: Typical 4-bolted connections after the tensile test (19).

A typical force versus elongation graph recorded during the destructive test is shown below in Figure 5.15. The graph represents a full-scale connection failure capacity of approximately 2 x 950 kN = 1900 kN. The average failure load achieved for test configuration 3 is approximately 1870 kN which is consistent with the theoretical capacity of 1925 kN as already calculated in Section 5.13.3 above. The remaining test results which were made available corresponded well with theoretical failure loads calculated for configurations 1 and 2, although not shown in this document.

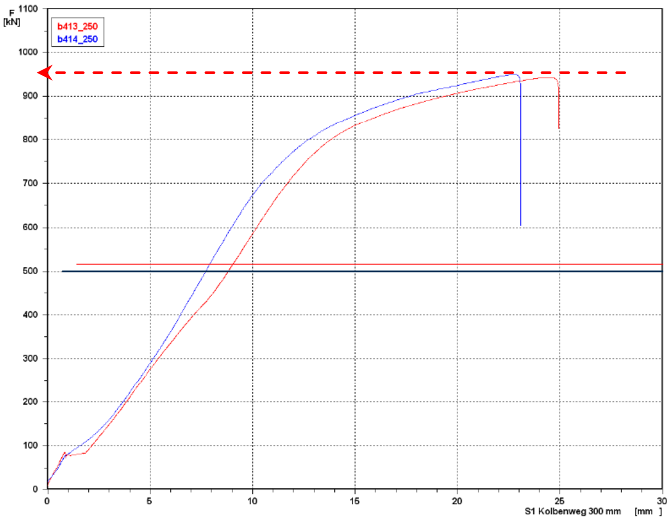


Figure 5.15: Force vs displacement graph for 4-bolt connection (19).

It can be concluded that the ultimate capacity of the tie-beam connection as calculated in Section 5.13.3 and simulated in controlled laboratory conditions was significantly greater (by a factor of approximately 1,5) than the worst-case design load combinations obtained from the OEM design office as shown in Table 5.3. This statement must, however, be considered in conjunction with Section 5.13.9.

It must also be kept in mind that the laboratory tests merely affirm the validity of the theoretical calculations done to evaluate the capacity of the tie-beam connection; these were based on the bolts meeting the appropriate specification for Class 10.9 bolts. Brittle fracture due to material defects was therefore not evaluated with laboratory testing. Brittle fracture will be discussed in the section below.

5.13.5 METALLURGICAL EXAMINATION

The metallurgical examination of tie-beam connection fasteners is discussed in this section. As already mentioned, high-strength Class 10.9 HSFG zinc electroplated fasteners were utilised in the tie-beam connections. The manufacturing process of these fasteners includes an acid pickling cleaning procedure prior to the electroplating process. It is known within the industry that these fasteners are susceptible to hydrogen embrittlement (HE) while being coated (29, 54). To eliminate the risk of hydrogen effects, a “heat soak” baking process is utilised prior to the electroplating process to drive off hydrogen ions from the material grain boundaries. Mechanical cleaning was recently proposed by SAISC because of quality issues regarding the above process (29).

Metallurgical examination of various bolts was undertaken by three independent experts in the industry (16, 17, 18). It must be noted that the identification of the origin of failed bolts is not always possible. Not only tie-beam connection fasteners were therefore tested. Reports from three independent laboratories nevertheless noted the high probability of hydrogen embrittlement based on factual evidence gathered during metallographic examination. Furthermore, in an additional test batch, the hardness of six out of seven specimens exceeded the requirements of ISO 898-1 (2009) for Class 10.9 fasteners. As discussed in Section 5.13.11, hardness levels above the upper threshold value increases the risk of brittle failure (41).

Cracks within the thread roots are shown in Figure 5.16. The metallurgical test report concluded that these cracks “were likely due to hydrogen embrittlement ...”

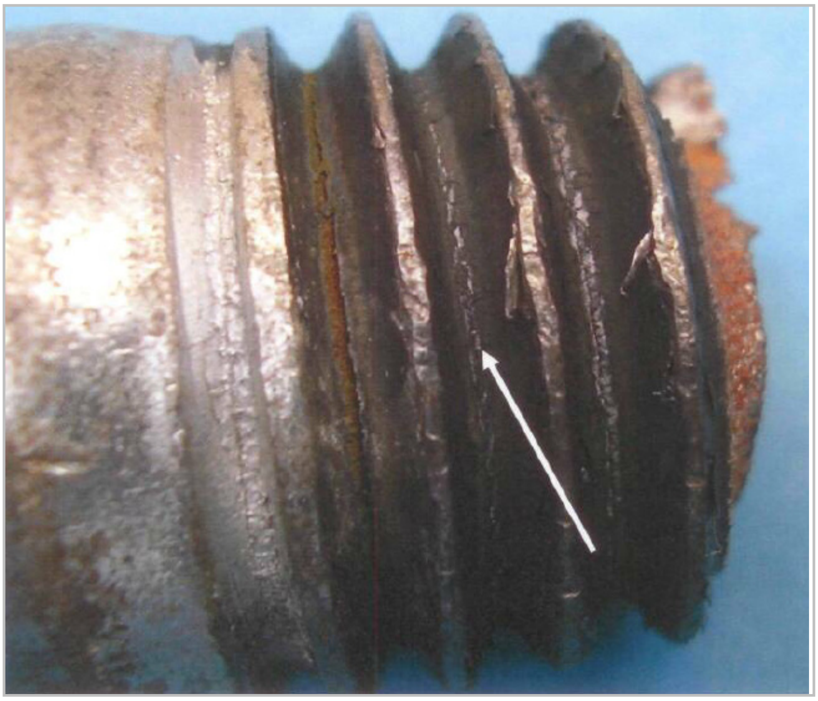


Figure 5.16: – Bolt specimen, cracks shown in thread roots (17).

Additional metallurgical tests were carried out on bolts removed from the tie-beam connection that did not fail to support arguments for hydrogen embrittlement. Residual bulk hydrogen contents of 7,9 ppm and 8,2 ppm were found in two of the samples. Because hydrogen ions tend to migrate towards areas where the stress concentration is most severe, according to expert opinion (55), hydrogen levels could be as much as 10 times higher at the thread roots. The metallurgical report (18) advised that these hydrogen levels will generate a residual internal pressure of 95 MPa and 98 MPa respectively, which will be supplementary to the stress developed during tightening. This data was utilised to calculate the reduced capacity of the tie-beam connection as shown in Section 5.13.10 and Section 5.13.11.

Note that in Ham *et al.*'s publication entitled "Evaluation method of sensitivity of hydrogen embrittlement for high strength bolts" (32), a pre-charge hydrogen content level of between 4 and 7 ppm was utilised to evaluate HE in Class 10.9 fasteners.

A summary of the relevant metallurgical test results which are directly related to HE is shown below in Table 5.7.

Table 5.7: Summary of metallurgical test results related to HE.

Test report title	Fastener details	Findings extract
Analysis of failed M24 and M20 bolts (16)	M20 – worked loose M24 – fractured (From *reclaimer*, supplied under the same contract as the failed stacker)	Brittle fracture "typical for hydrogen embrittlement" "indication that cracks were initiated by HE ... propagated by fatigue"
Investigation into aspects of the failure of two bolts (12)	2 off, Class 10.9 (from failed stacker)	"pre-existing root cracks 0,7 mm deep" "evidence of hydrogen cracking"
M24 bolt failure (17)	4 off, Class 10.9 Only two tested (from failed stacker)	"possibility of HE" "root cause not HE, cracks contributed or hastened failure" "cracks likely due to HE"
Evaluation of bolts (18)	5 off, Class 10.9 Only 2 tested (from failed stacker)	8 ppm bulk hydrogen content

5.13.6 ADDITIONAL INFORMATION

Although hydrogen embrittlement on high-strength fasteners is not unknown in the industry, the topic remains controversial. Evidence revealed by metallurgical testing, as discussed above, can nevertheless be supported by photographic evidence taken a few months prior to the collapse of the stacker (14). Figure 5.17 below shows two views of a failed bolt located on the eastern tie-beam connection (at normal boom stacking position) from both sides.

Figure 5.17: Fractured bolt at tie-beam connection months prior to the collapse (14).

A separate investigation was conducted to establish if the fractured bolt was replaced prior to the collapse (22). Bearing marks within the corresponding bolt holes, as shown in Figure 5.18, suggest that a bolt was present at the time of collapse. Note that the bolt did not fail in shear. The pre-tension of the fasteners is discussed under Section 5.13.10 and Section 5.13.11, Hydrogen effects and preload.

5.13.7 SPECIALIST CONSULTATION

During a consultation discussion held with Dr G Krige and Dr J Wannenburg, it became evident that findings from the investigation, at the time of discussion, were not entirely conclusive. The preceding investigative work had focused predominantly on structural design matters, while protection systems were not investigated in sufficient detail. More detailed audits of the protection systems were subsequently initiated. The effect of a sideward collision of the boom into the material pile and the consequence of boom luffing with the presence of raw material at the boom tip had to be investigated in addition to the work already conducted.

5.13.8 OPERATOR ABUSE

The default machine operation mode is the automatic mode in which the stacker's long travel and stack-out operations are controlled by the selected PLC program. It is nevertheless possible to select a manual operational mode. The possibility of operator abuse was investigated since photographic evidence shows interaction between the stockpile and the stacker boom as shown in Figure 5.18 below.

Figure 5.18: Scrape marks at the bottom of a stacker boom.

The effect of the machine’s collision with a stockpile was simulated with the cooperation of the OEM’s design engineer. The collision force was found to be insufficient to cause the structural collapse of tie-beam connections, although it would induce significant loads and subsequent damage to machine slewing components (12). No signs of significant impact could be identified on similar machines used at the same operation. The instrumentation protection systems were, however, found to be insufficient (13). Refer to Section 5.14.1.

5.13.9 LUFFING OF A BURIED BOOM

The purpose of this section is to quantify whether the tie-beam connections would have failed when the stacker boom is luffed against the full hydraulic trip value of the luffing cylinder. This section must be read in conjunction with Section 5.13.1, Loading conditions and machine protection systems.

It was identified from the instrumentation system audit (13) that the machine did not trip out at the designer’s prescribed cylinder force value of –871 kN (C). The boom conveyor was modelled empty and fully loaded. The load at the boom tip was determined to ensure that the hydraulic trip value would be achieved for each load case (12). The results from the OEM’s analysis are graphically presented in Figure 5.20 below and must be read in conjunction with Table 5.8. (Values shown are not entirely consistent with design loads provided in Table 5.3, but were nevertheless obtained from the OEM.)

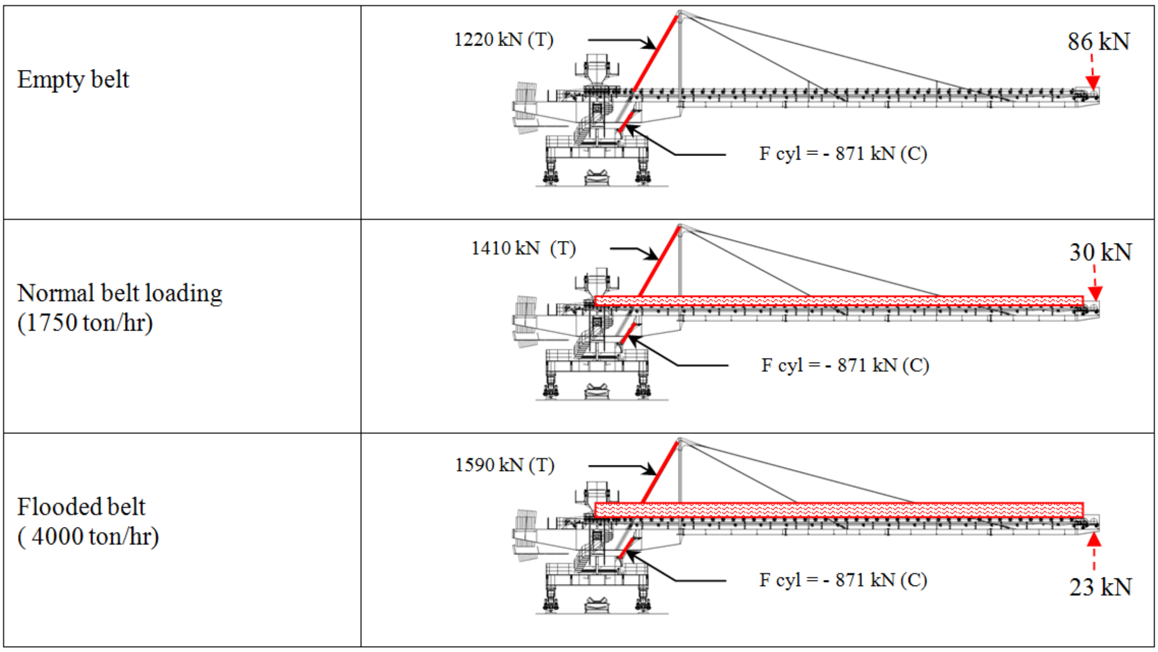


Figure 5.19: Graphic summary of analysis results.

As shown in Table 5.8 below, for a flooded conveyor, the luffing cylinder should have tripped the system out. To obtain a force equal to the luffing cylinder trip force value of –871 kN (C), requires a negative force of 23 kN (i.e. boom needs to be supported so as not to exceed the trip value) at the boom tip which implies that no additional load as a consequence of blocked chute conditions could be accommodated. The corresponding tie-beam connection force for unfactored flooded belt conditions was calculated as 1590 kN.

It is therefore obvious that the boom luffing cylinder should not have been luffed in situations where the boom tip was buried in the stock pile. It is pertinent that the machine's protection systems should not have permitted luffing action under overload conditions.

Table 5.8: Unfactored tie-beam capacity with point load at boom tip.

Description	Empty (kN) LCI	Normal (kN) LC III	Flooded (kN) LC III
Downward force (12)	86	30	-23
Tie-beam design load (12)	1220	1410	1590
Vr, Shear resistance (Table 5.5)	931	1164	1164
Overload	289	246	426
% Overload	**31%**	**21%**	**37%**

5.13.10 REVISED FAILURE CAPACITY, TIE-BEAM CONNECTION

The ultimate shear capacity (the failure load) of a single Class 10.9 M 24 bolt, acting in double shear, with one shear plane intercepting the thread, was previously calculated as 481 kN in Section 5.13.3. If the residual stress due to hydrogen effects is taken into account, the shear capacity can be recalculated by considering the first principal stress equal to the ultimate tensile strength of the bolt using Equation (10) below (34).

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2} + \text{Hydrogen induced residual stress} \qquad (10)$$

Where:

σ_1 = First principal stress. [Pa]

σ_x = Stress component in x-direction. [Pa]

σ_y = Stress component in y-direction. [Pa] = 0

τ_{xy} = Shear stress component in xy-direction. [Pa]

Calculations in Table 5.9 and 5.10 below show that the revised shear capacity per bolt can be subsequently reduced to: 186 kN + 75 kN = 261 kN

Table 5.9: Failure shear force through bolt shank considering hydrogen effects.

Description	Value	Unit	Remark
D_n, Nominal bolt shank diameter	0,024	m	Shank diameter
σ_1, Bolt ultimate tensile strength	1000	MPa	Class 10.9
Bolt pre-load tension force	257	kN	SASCH p 6.16 (52)
σ_x, Normal tension stress	568	MPa	$\sigma_x = \frac{bolt\ preload}{Nominal\ shank\ area}$
Residual stress	100	MPa	Hydrogen effects (18)
τ_{xy} , Allowable shear stress	547	MPa	From Equation (10)
Maximum shear stress factor	1,33		4/3 for round section (34)
τ, Average shear stress at failure	411	MPa	$\tau = \frac{\tau_{xy}}{1{,}33}$
V, Failure shear force	186	kN	V = τ x nominal shank area

Table 5.10: Failure shear force through the thread considering hydrogen effects.

Description	Value	Unit	Remark
D_n, Bolt minor diameter	0,0203	m	Thread root diameter
σ_1, bolt ultimate tensile strength	1000	MPa	Class 10.9
Bolt tension force	257	kN	SASCH p 6.16 (52)
σ_x, Normal tension stress	794	MPa	$\sigma_x = \frac{bolt\ preload}{area\ based\ on\ D_n}$
Residual stress	100	MPa	Hydrogen effects (18)
τ_{xy} , Allowable shear stress	309	MPa	From Equation (10)
Maximum shear stress factor	1,33		4/3 for round section (34)
τ, Average shear stress at failure	232	MPa	$\tau = \frac{\tau_{xy}}{1{,}33}$
V, Failure shear force	75	kN	v = τ x nominal shank area

It is not known how many tie-beam connection bolts were affected by hydrogen embrittlement (HE). The hypothetical reduced tie-beam resistance can nevertheless be demonstrated as shown in Table 5.11 below:

Table 5.11: Hypothetical tie-beam capacity considering hydrogen effects.

Condition	Tie-beam connection shear capacity (kN)
All bolts to specification	1792
1 bolt effected by HE	1705
2 bolts effected by HE	1485
3 bolts effected by HE	1265
4 bolts effected by HE	1044

Considering the specified system trip value for an –871 kN (C) luffing cylinder force, from Figure 5.20, the tie-beam force required to sustain a flooded boom belt condition was found to be 1590 kN. When compared with the reduced tie-beam connection values as shown in Table 5.11 above, the possibility of a failure becomes plausible. SCADA recordings as discussed under Section 5.14, show that excessive cylinder force values were captured just prior to the collapse. The peak loading magnitude could not be captured on the recording system; –921kN (C) was nevertheless recorded. Refer to Figure 5.24.

5.13.11 HYDROGEN EFFECTS AND PRELOAD

This section explores the influence of HE on a high strength fastener, when subjected to preload. The suitability of the designer's selection of high-strength electro-galvanised fasteners is also discussed.

From a tie-beam connection fastener examined, it was found that the hydrogen induced area extended approximately 1 mm from the thread to the core (17). At fracture, the stress intensity of the material is equal to the plane strain fracture toughness for the metal as shown in Equation (11) (10).

$$K_{IC} = \sigma\beta\sqrt{\pi a} \qquad (11)$$

Where:

K_{IC} = Plane strain fracture toughness. [$Pa.m^{0.5}$]
a = Crack depth. [m]
β = Stress intensity factor.
σ = Fracture stress. [Pa]

The geometry for an external crack in the circumference of a rod in tension is shown below in Figure 5.20.

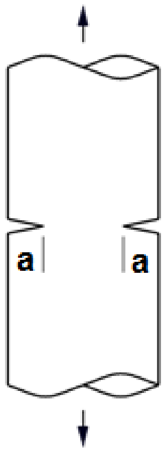


Figure 5.20: Geometry for circumferentially cracked rod in tension (8).

For a preloaded fastener, Brennan & Dove (8) suggest a stress intensity factor of 1,6 as shown in Figure 5.21 below.

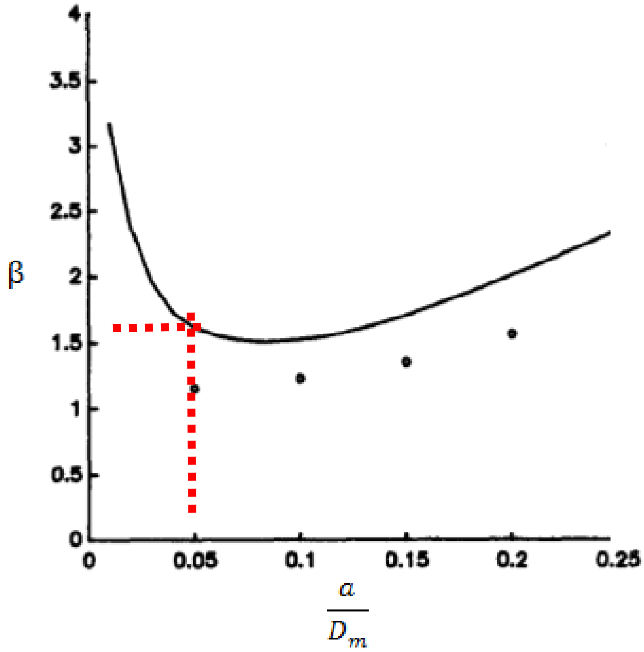


Figure 5.21: Stress intensity factor, circumferential crack in a round bar under tension (8).

Literature suggests that the shear capacity of high-strength fasteners is not reduced by preload under normal conditions (49, 57). When embrittlement effects are considered, however, it can be shown in tabulated calculations from Table 5.12 below that the bolt pre-tension was sufficient to result in the bolt failure. This may explain why the fractured bolt was noted months prior to the collapse, as previously illustrated in Figure 5.17.

Table 5.12: Fracture stress of a Class 10.9 bolt, 1 mm crack depth.

Description	Value	Unit	Remark
K_{IC} , Plane strain fracture toughness	50	$MPa.m^{0.5}$	(10)
a, Crack depth	0,001	m	(17)
D_m, Bolt minor diameter	0,0203	m	
$\frac{a}{D_m}$	0,05		$\frac{0,01}{0,0203}$
β, Stress intensity factor	1,60		(8)
σ, Fracture stress	558	MPa	
Residual stress due to hydrogen	100	MPa	(18)
σ , Fracture stress considering HE	458	MPa	

The nominal tensile stress that will cause fracture in the cracked bolt was calculated at 458 MPa in Table 5.12 above, which is far less than the 794 MPa which would be induced under normal preload conditions as calculated in Table 5.10.

It can therefore be concluded that the bolts used in the tie-beam connections could have failed under preload only.

The topic of corrosion and embrittlement is discussed at length in the American Institute of Steel Construction's (AISC) Guide to design criteria for bolted and riveted joints (41). From laboratory tests referenced Kulak *et al.* note "… it became apparent that the higher the strength of the steel, the more sensitive the material becomes to both stress corrosion and hydrogen stress cracking. The study indicated a high susceptibility of galvanized A490 bolts to hydrogen stress cracking." It is ultimately concluded that "*galvanised A490 bolts should not be used in structures. The tests did indicate that black A490 bolts can be used without problems from brittle failures in most environments.*" (A490 bolts are the direct equivalent of the Class 10.9 used in South Africa).
Considering the layout-configuration of the tie-beam connection as shown in Figure 5.11, it can be argued that there was no apparent reason why high strength fasteners were required. The crux of the matter, however, lies in the poor selection choice of electro-galvanised high-strength fasteners which are susceptible to HE.

5.14 FORENSIC INVESTIGATION

In spite of various attempts, neither the owner nor the insurer could source the services of a suitable professional forensic investigator. After concluding the structural design audit and structural failure analysis in conjunction with the appointed professional consultancy, the author carried out the forensic investigation by interpreting the outcome of additional investigations carried out by various task teams across engineering disciplines, as detailed below.

5.14.1 LOADING CONDITIONS AND MACHINE PROTECTION SYSTEMS

Independent systems audits were conducted by subject experts on the hydraulic and instrumentation systems associated with the stacker. Findings from these specialist investigations were used to evaluate whether the machine was adequately protected against abnormal operating conditions.

5.14.2 SIDEWARD COLLISION OF THE BOOM

The probability of a sideward collision of the stacker boom with the material pile, prior to the collapse, could not be ruled out and required detailed investigation. Calculations by the OEM designer indicated that the sideward collision force as a consequence of the boom and pile interaction would be insufficient to cause the tie-beam connection to fail.

The instrumentation audit findings found that the instrumentation systems installed to avoid interaction between the coal pile and stacker boom were insufficient (13). The instrumentation auditor recommended that additional material sensors be fitted to the stacker boom. Scraping marks below the stacker booms confirmed the occurrence of this interaction. However, no visible damage to walkways and structural members was found. The investigation team concluded that no major collision between the stacker boom and material pile had occurred prior to the collapse.

5.14.3 MATERIAL LOADING AND BELT SLIPPAGE

The slipping of the boom belt will cause an increased material load per running length of the belt, to the extent that flooded belt conditions can occur. The contract specified that flooded belt conditions had to be catered for.

The instrumentation audit revealed that the speed switch setting of the stackers was incorrectly set at 20 % of belt speed instead of 80 %, which is the norm for fixed-speed conveyors as dictated in the Anglo American Corporate Specification AA 673018 (2009) (4).

Boom belt slippage was a common occurrence at the operation prior to the collapse. Remedial measures taken to eliminate belt slippage included an increased conveyor take-up tension setting and the ceramic lagging of the boom conveyor's drive pulley.

5.14.4 MATERIAL LOADING – OTHER FACTORS

Matters relating to the boom belt loading are discussed above. There are nevertheless additional factors that need to be considered.

The boom belt speed was designed at 2,8 m/s. It received material from a yard belt running at 3,8 m/s. It is common engineering practice to match the belt speeds of a feed and receiving conveyor, especially where belt widths match. The mismatching of belt speeds causes increased boom belt loading conditions, especially when boom belt slippage occurs.

Raw material from other mining sources was dumped at the raw material yard and fed onto the yard belt with a feeder arrangement at an average capacity of 600 ton/hr and peak capacity of 900 ton/hr. Due to the absence of control interlocks at the time of the collapse, this additional material feed possibly contributed to the overload. Nevertheless, a slipping boom belt, considering the incorrect speed switch setting, could also have resulted in abnormal loading i.e. flooded belt conditions.

5.14.5 BOOM OVERLOADING

This section explores aspects relating to the overloading of the stacker boom and conditions under which luffing operations could be carried out. During a chute blockage, material may be deposited at the boom tip. Calculations by the OEM designer indicated that the luffing operation, with the presence of a small volume of material at the boom tip, would exert significant forces onto the tie system of the stacker. Refer to Section 5.13.9.

Figure 5.23 below shows a snapshot of the luffing cylinder pressures and corresponding cylinder force exerted over the time just prior to the collapse. Refer to the red scale for cylinder forces with equivalent belt loading values marked for 2000 ton/hr, 2700 ton/hr and 3300 ton/hr, respectively. The PLC relies on an algorithm to calculate the system cylinder force based on the cylinder pressures recorded.

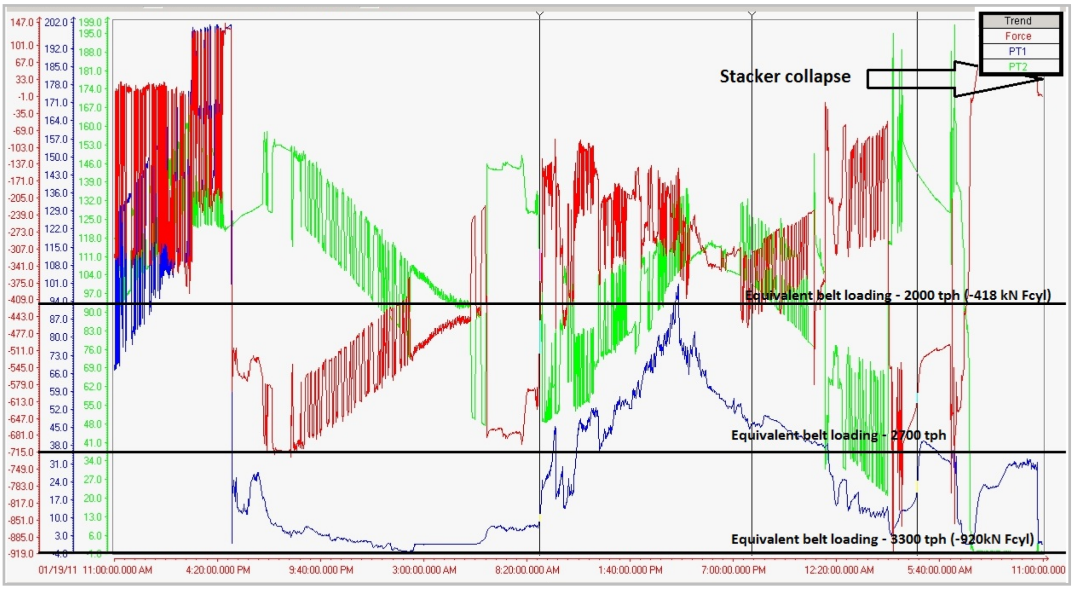


Figure 5.23: SCADA recordings prior to collapse.

The luffing cylinder force associated with peak design conditions of 2000 ton/hr is approximately –420 kN (C). The OEM designer specified a hydraulic trip value at a corresponding cylinder force of –871 kN (C). Flooded belt loading conditions will induce a cylinder force of approximately –1214 kN (C). Note that luffing cylinder force values should not be confused with tie-beam forces.

From the SCADA recordings prior to the collapse, it is clear that normal boom loading was significantly exceeded and attempts to luff the stacker boom were recorded. The cut-off value on the recordings was limited at –920 kN (C) as shown in Figure 5.24 below.

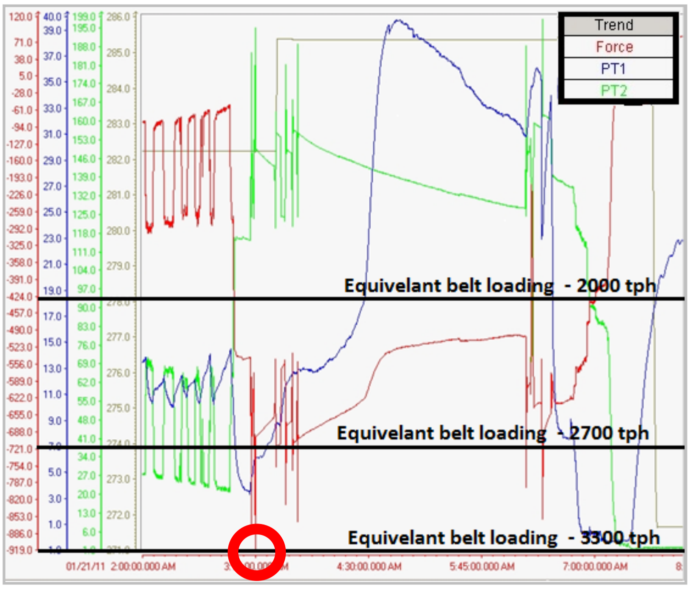


Figure 5.24: SCADA recordings showing cylinder force value in excess of –920 kN (C).

With specific reference to the instances where the cut-off value of –920 kN (C) was recorded just prior to the collapse, there are at least two possibilities for achieving the SCADA signature as recorded. (Refer to SCADA recordings from approximately 3:00 am and 6:30 am):

1. An attempt to luff a boom with the tip buried within raw material (regardless of the material load on the boom belt).
2. An attempt to luff an overloaded boom partially resting on the raw material pile.

Considering the above, the cylinder loading recorded on the SCADA prior to the collapse is not surprising.

Regardless of the exact reason for reaching the luffing cylinder force values in excess of –920 kN (C), the SCADA log revealed no indication that the hydraulic trip value, as specified by the designer, was ever triggered. The value of –871 kN (C) is nevertheless considered to be too high. A reduced value would allow more response time to initiate corrective action before structural overload would occur. The machine tripped out on under-speed of the boom conveyor (13).

Although abuse could not be ruled out, it must be noted that, with the absence of the hydraulic protection system, the normal operation of the machine could have led to loading conditions beyond the safe working range intended by the designer.

According to the operations personnel, the software program activated at the time of the collapse had the hydraulic trip settings incorporated as specified by the OEM. However, the event log program was not comprehensive. Subsequently hydraulic trips may have been triggered without being recorded. The stacker was nevertheless tripped out on under-speed of the boom belt.

5.14.6 BOOM OCCILATIONS

The effect of the vertical boom oscillations as observed and recorded on the SCADA logs several hours prior to the collapse are discussed in this section.

The malfunctioning of the hydraulic system which manifested as a vibration on the luffing cylinder and subsequently a vertical oscillation motion of the boom structure, could introduce abnormal loading conditions of unknown magnitude. The effect of a suddenly applied load versus a static or gradually applied load is widely published in text books. It can therefore be concluded that under boom overload conditions the oscillation motion of the boom structure could introduce abnormal loading conditions. The SCADA log shows that oscillation occurred during the time that the machine was on stop because of the overloading of the machine. No abnormal luffing cylinder forces were captured. It must, however, be noted that the SCADA system is configured in such a way that data capturing is not continuous, but set at a predetermined sampling rate. It is therefore possible that peak loads may not be recorded.

5.15 SUMMARY OF KEY FINDINGS

- Initial investigations highlighted the possibility of two failure modes i.e. the luffing cylinder clevis failure and the tie-beam connection failure.
- Metallurgical testing and analytical work made it possible to eliminate the possibility of the clevis failure postulation as the primary cause of the collapse.
- The failure mechanism associated with the tie-beam connection failure correlates perfectly with site observations of the collapsed machine.
- It was concluded that the perceived upward movement of the boom as captured in the operator reports, was as a consequence of the malfunctioning of the luffing hydraulic system as noted in the SCADA recordings.
- It was established that the shear capacity of the tie-beam connection was not compliant with the requirements of ISO 5049-1 (1994). Loading conditions were underestimated because of the use of an incorrect material density and belt fill factor.
- The ultimate capacity of the tie-beam connection was calculated from first principles and confirmed by the results obtained from destructive tests carried out in a test laboratory, disregarding the effects of HE.
- Although not compliant with the requirements of ISO 5049-1 (1994), the ultimate capacity of the tie-beam nevertheless exceeded the design loads as determined from the most severe load case combination as derived from ISO 5049-1 (1994).
- Considering the two aspects directly above, the tie-beam connection was a marginal design, which was overloaded, based on the requirements of ISO 5049-1 (1994). Fastener threads intercepted a shear plane.
- The high probability of the presence of hydrogen embrittlement in the high-strength electro galvanised fasteners used in the tie-beam connection was discussed.
- Fracture mechanics were applied to demonstrate that the tie-beam connection fasteners could have failed under preload only, when hydrogen embrittlement was considered.
- The load-carrying capacity of the tie-beam connection fasteners could have been reduced by hydrogen effects to a value far below what would be required to sustain a boom load associated with the luffing operation of an overloaded stacker boom.
- Photographs taken of a fractured tie-beam connection bolt months prior to the collapse provide a basis for a plausible hypothesis that a tie-beam connection bolt could have failed prior to the overloading event and subsequent collapse.
- SCADA recordings show that the boom loading significantly exceeded the intended design parameters.
- It was proven, from the interpretation of SCADA recordings and the hydraulic and instrumentation systems audit findings, that the protection systems on the machine were

inadequate to ensure that structural loading remained within the intended design parameters.

- It remains a mystery why the boom eventually collapsed after the material load was completely removed and not during the overloading event which preceded the failure.

The various factors which is deemed to have contributed to the failure is summarised below in Table 5.13.

Table 5.13: Summary of factors which contributed to the failure.

			Related to		
Catergory		**Item**	**Contribution to root cause of failiure**	**Non compliance with Supply Contract**	**General engineering practice**
Design loading	1	Mis-matching belt speed (2.8 vs 3.8 m/s)	x		x
	2	Incorrect contractual bulk density. (850 kg/m3 vs 1100 kg/m3)	x	x	
	3	Boom belt not designed for flooded belt conditions. (2000 tph vs 4000 tph)	x	x	
	4	Designers in Europe never had sight of Supply Contract. Bulk density and belt loading items. (Lack of communication)	x		x
	5	Additional coal feed on yard belt from open cast. (Insignificant - Contract specified flooded conditions)			
Structural design	6	Tie-beam connection bolts under-designed and not compliant to ISO 5049-1	x	x	
	7	Poor connection concept for critical element. (pin in double shear is common detail)	x	x	x
	8	Poor fastener selection. (High Class electro galvanised bolt susceptible to hydrogen embrittlement)	x		x
	9	Poor engineering practice. (Thread in shear plane, critical connection, dynamics)	x	x	x
	10	Double redundancy of tie systems implied in Supply Contract but not incorporated in the design.		x	
	11	Failure to provide safeguard against unforeseen failure or loss of overall structural integrity specified in Supply Contract.		x	
	12	ISO 5049-1's requirement for fitted bolts disregarded on a dynamic machine application.		x	x
Materials	13	Metallurgical testing confirms a high probabilty of hydrogen embrittlement. (bolts from failed tie connection can't be traced)	x		
Protection system	14	Alarm level specified was too high to avoid overload. (2000 tph design vs 3000 tph@ - 871 kN)	x		x
	15	Stock yard system was suppossed to be fully automatic? Obligation to provide sufficient overload protection.		x	
	16	Boom instrumentation not sufficient to avoid collision.			x
	17	Protection insufficient to protect machine. Luffing of overloaded boom must be impossible.	x	x	
Commissioning	18	Alarm level of - 871 kN exceeded but machine didn't trip out.	x		
	19	Speed switches incorrectly commissioned. (Trip out commissioned at 20 % of belt speed vs requirement of 80 %)	x	x	x
	20	Machines used without proper commissioning.	x	x	x
Abuse	21	Although abuse could not be proven it can not be ruled out altogether	?		

The root cause of the stacker collapse can therefore be summarised as follows:

Design deficiencies contributed to a marginal design of critical connections of which the carrying capacity was further exacerbated by defective bolts. The absence, malfunctioning and incorrect commissioning of machine protection systems did not prevent an overload condition which led to the catastrophic collapse of the stacker.

5.16 SIMILAR DESIGNS

Subsequent to the investigation findings, remedial work had to be done on the remaining three stackers used at the operation. A retrofit design solution was developed with oversight from the author to equip these machines with a 5-bolt connection configuration by utilising M27 fitted bolts, which were custom made. Although not an ideal solution, the modification nevertheless provided an increased margin of safety and satisfied the requirements as specified in ISO 5049-1 (1994). The modified tie-beam connection is shown below in Figure 5.25.

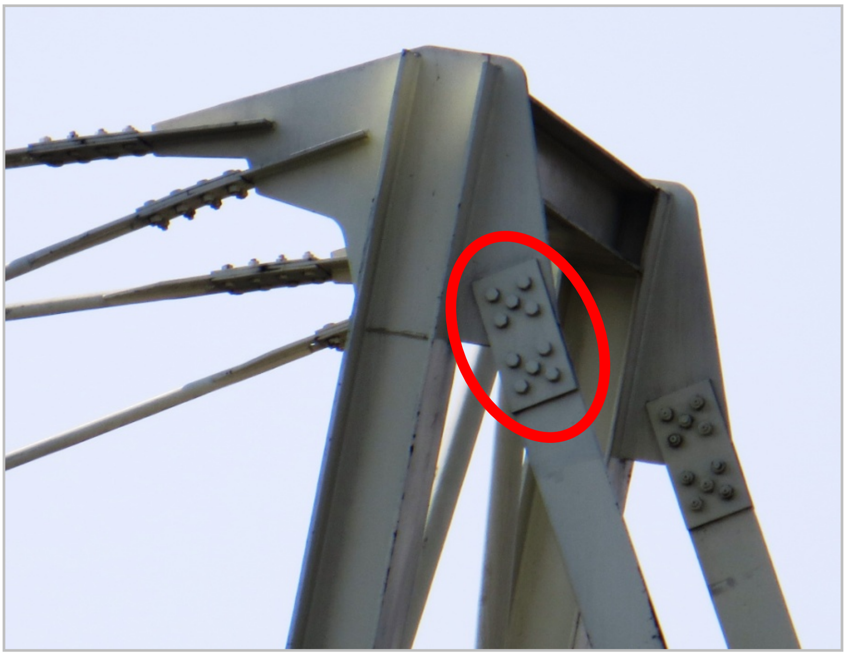


Figure 5.25: Modified stacker tie-beam connection with fitted bolts.

Shown below in Figure 5.26 is an example of a revised tie-beam connection design concept that was developed for another operation where the lessons from the failure, as discussed in Chapter 5, were incorporated in the new stacker design. The use of single shear pins on tie systems associated with mobile BMH equipment is, however, not uncommon.

Figure 5.26: Revised tie-beam connection design, pins in double shear.

5.17 DISCUSSION

In order to align the detailed failure investigation with the outlined study objectives and the theme developed in preceding chapters, it is required to take a retrospective view of the factors that contributed to the collapse of the stacker.

The malfunctioning and incorrect commissioning of machine protection systems resulted in accidental loads, which were much higher than originally anticipated in the structural design. It must be understood that, at the time of the collapse, the machine had not been finally handed over to production personnel. Production pressures required that machines had to be operated before the final handover was possible. To have a machine in production for a year before final commissioning introduces a range of risks that are typically not recognised in design, but it should be remembered that neither protection systems nor calibration of instrumentation devices are complete. Although an attempt was made to protect the machine, the trip-out setting values specified by the designer could not prevent system overload. The need for an early warning system could have possibly been identified during a structured design risk assessment. This said system must ideally trip out the yard conveyor while increasing the boom belt speed in order to reduce the loading on the stacker boom and subsequently on the tie-beam connection.

Although the structural design did not comply with the rules stipulated in ISO 5049-1 (1994), the standard requires no mandatory machine protection systems. The operations personnel did not understand the sensitivity of the structural design. Protection systems should nevertheless not permit actions such as luffing operations if these actions impose loads outside the design parameters.

The Anglo American Corporate Specification AA 114/1 (2007) (1) requires that structures are inspected after construction by the design engineer as a final check point. It is the author's view that this practice should be incorporated in the final acceptance of mobile BMH equipment and must include acceptance of protection settings on all relevant systems across disciplines. This matter is discussed further in Chapter 6.

Although the respective bulk densities were specified in the contract together with specific requirements from the owner, it was found that the OEM designer, based in Europe, never saw the original contract and based the design on the design scope received from the South African branch of the company. This incident highlights the fact that either the final design parameters were never reviewed by the owner's project implementation team, or that the necessary experience or competence was not available to do so. The review of the final design parameters by representation from the owner is not the norm since smaller companies simply cannot afford to employ such expertise.

The requirement for double redundancy on critical tie systems was highlighted by the incident and was in fact specified by the owner. Notwithstanding that the manufacturing process for electro-galvanising of high strength fasteners has been revised to reduce the risk of hydrogen embrittlement (29), the use of these fasteners should be avoided where possible.

5.18 CONCLUSION

The complexity of collapses associated with mobile BMH equipment was explored in this case study. It is anticipated that the reader will appreciate the need for a systematic and structured investigation approach in order to establish the root cause of any failure. The contribution of various interdependent factors to the catastrophic collapse of the machine was presented. The underlying study theme pertaining to mobile BMH machine protection systems, which are currently not addressed in ISO 5049-1 (1994), was once again highlighted. The requirement for the structural design engineer to play a more prominent role in the final acceptance of mobile BMH equipment was also pointed out.

6 PROPOSED DESIGN APPROACH

In this chapter, a retrospective view is taken of the case studies already discussed in the preceding chapters in order to identify shortcomings and omissions from the current design practices followed by leading mobile BMH equipment OEMs. The outcome of the conducted survey, as briefly discussed in Chapter 2, is discussed. The revision of the ISO 5049-1 (1994) Standard will be proposed with specific focus areas. A risk-based design approach, which is integrated across engineering disciplines, is proposed.

6.1 CURRENT DESIGN PRACTICE

This section must be read in conjunction with Section 2.8 OEM Survey, and the detailed survey responses as provided in Appendix C. The key outcomes from the survey are discussed below.

Design approach – Most mobile BMH machines are internationally designed in accordance with the rules and requirements specified in ISO 5049-1 (1994) except for in Australia where AS 4324-1 (1995) usually applies.

Although the majority of OEMs follow an allowable stress design approach, some follow a limit state design approach whereby ISO 5049-1 (1994) is merely utilised as a loading code. The design of structures and parts is then carried out by applying a limit state based design code such as DIN 18800-1 (1990). Limit state load factors are therefore applied during analysis, in conjunction with loading obtained from an allowable stress based standard.

Sufficiency of ISO 5049-1 (1994) – All OEMs who participated in the survey considered ISO 5049-1 (1994) to be adequate to facilitate the design of safe mobile BMH equipment although personal correspondence acknowledged shortcomings (30, 56).

Conservatism of designs – OEMs seem to hold very different opinions on this matter. Although some OEMs are of the opinion that heavier, more conservative designs would be beneficial when taking a long-term view, others suggest that the rules as dictated by ISO 5049-1 (1994) are, in some cases, over conservative.

Machine protection systems – The general consensus indicates that OEMs are of the view that mobile BMH machines are adequately equipped with protection systems, although operator abuse is of some concern.

Risk assessments – Although design risk assessments are incorporated by some OEMs, the extent to which these are carried out varies widely. Responses indicate that risk assessments are often done in isolation by the designer, without input from the end-user, while other OEMs will conduct a risk assessment only if required by the client.

Integration of machine protection systems and structural design – The responses obtained from the survey were not conclusive on this topic. It is nevertheless implied that the integration between machine protection systems and structural design is deemed adequate.

Commissioning involvement of the structural design engineer – Feedback response suggests that there is normally very limited involvement of the structural design engineer during the commissioning and final handover of mobile BMH equipment to the owner.

6.2 KEY OBSERVATIONS FROM CASE STUDIES

The author's observations regarding the design approach followed by OEMs, as noted in preceding case studies, are discussed below. These observations are evaluated in the light of the survey responses already discussed in Section 6.1, Current design practice, above. This section must therefore be read in conjunction with Chapters 3 and 5.

6.2.1 CASE STUDY 1 – WASTE SPREADER COLLAPSE

Although the encrustation loading allowance was supposedly done in compliance with ISO 5049-1 (1994), in fact only the minimum prescribed loading was adopted without considering a realistic design load.

The wind loading was initially correctly assessed, but the designer obtained a concession to relax the design wind intensity resulting in an underestimation of the wind loading by a factor of 2,5. The overall structural stiffness of the spreader was not acceptable. ISO 5049-1 (1994) does not provide rules or guidelines for serviceability criteria of mobile BMH equipment.

It could be argued that a risk-based design approach, with input from an experienced owner's team, may have identified the above issues as shortcomings, which would then have ensured greater redundancy in the design. The additional capital cost of catering for the correct wind loading and encrustation loading would be insignificant compared to the overall equipment cost. OEMs, however, want to design to tried and tested parameters which they understand, and which can confidently be met within the quoted price. Input from owners' teams is not welcomed when it calls for additional design requirements. A high premium is placed on the machine price if OEMs do agree to additional design requirements, which may mean that the project manager pressurises the owner's team to waive the additional requirements. If designs are properly executed, the cost premium may, in many cases, be reasonably low (44).

6.2.2 CASE STUDY 2 – PORTAL RECLAIMER COLLAPSE

The author's conclusion, derived from analysing documentation relevant to this case study, is that the malfunctioning (material proximity probes) and incorrect commissioning of machine protection systems (electronic shear key) and reclaimer drive equipment (fluid coupling fill level) resulted in accidental loads which were much higher than was originally anticipated in the structural design. Insufficient change control resulted that oversized electric motors were procured without consultation with the design engineers which resulted in a higher system torque.
The design rules provided in AS 4342-1 (1995), facilitate a safer overall machine design than ISO 5049-1 (1994) does.

It must be noted that it was proposed by the client's technical support team who assisted with the re-instatement of the machine that structures be strengthened to cater for collision events. The additional cost for such remedial work *in situ* will always exceed by far the cost of providing a more robust design in the beginning. The quality of site work will seldom, if ever, match that which can be achieved in the fabrication workshop.

Although OEMs seem to be satisfied that interdisciplinary design integration is adequately addressed within their organisations, this case study suggests otherwise. The extent to which engineering disciplines were compartmentalised during the detail design phase of the machine became apparent during the failure investigation. The author is of the opinion that the interdependence of various engineering disciplines and the effect of equipment selections (e.g. reserve margins of installed power above absorbed power, motor start-up characteristics, fluid coupling, electronic soft starter, variable speed drive, etc.) are not sufficiently emphasised. This calls for an understanding which goes beyond the limitations of the design engineer's specific field of expertise.

The survey indicated that limited input from the structural design engineer is generally required during commissioning and final handover to the owner. This case study demonstrates that the final acceptance and approval of the machine by the structural design engineer is crucial, but would require a thorough understanding of the key elements from other engineering disciplines as this has a direct impact on the structural loading and protection. It could therefore, be argued that with proper commissioning oversight from an experienced structural engineer, who understood the implication of loads generated within the system when load limiting devices are not correctly set up, the collapse could have been prevented.

6.2.3 CASE STUDY 3 – PARTIAL COLLAPSE OF A SHIP LOADER

This case study portrayed an incident where the encrustation loading allowance, as dictated by ISO 5045-1 (1994) was complied with. Nevertheless, after an extended period of time in service, the structure collapsed subsequent to the decommissioning of a cleaning system. The sensitivity of the original design was lost to operations personnel in service at the time of the collapse.

When evaluating the spillage loading at the time of the collapse, it was evident that the structure had more than sufficient redundancy to withstand normal (and perhaps abnormal) spillage conditions and exceeded the design parameters. The author is of the opinion that the original design was sound, and that the root cause of the failure should instead be classified under change control. Had the risk been properly assessed in consultation with a structural engineer prior to the decommissioning of the vacuuming system, the cleaning system would most likely have been reinstated or substituted with manual labour. Individual responsibility and vigilance remains a key aspect of safety.

The application of ISO 5049-1 (1994) must be done with caution when designing for special cases for which the mentioned standard was not intended. The requirement for a risk-based design approach, in line with the guidelines provided in the AA 248/2 (2010) (3) standard, is therefore emphasised again with the need for a basic understanding of the background which initiated the development of ISO 5049-1 (1994).

6.2.4 CASE STUDY 4 – DRUM RECLAIMER DAMAGE

This case study demonstrated that insufficient design integration existed between the structural, mechanical, electrical, and control and instrumentation engineering disciplines during the detail design phase of the original project. The control systems associated with the long travel of the machine were not fail-safe. Abnormal loads, not anticipated in the original structural design, were subsequently exerted on machine members. The equipment was nevertheless operated successfully for many years prior to the skewing incident. The damage could have been avoided by the incorporation of additional protection instrumentation for negligible additional capital cost. Machine protection systems are not addressed in ISO 5049-1 (1994).

The author is of the opinion that an integrated risk-based design approach, with appropriate representation from the relevant engineering disciplines of the OEM and client, may have identified this latent defect early on in the design. From the survey outcome, it is clear that design risk assessments are not necessarily carried out on new designs.

6.2.5 CASE STUDY 5 – CATASTROPHIC COLLAPSE OF A STACKER

The main case study is considered representative of a typical complex failure event. A number of common themes already discussed in the short case studies above are repeated.

Critical tie-beam connections were designed marginally and were in fact overloaded based on the design requirements of ISO 5049-1 (1994), although the ultimate capacity exceeded the most severe design load combination. These connections were configured in such a way that bolt threads were intercepted by shear planes. The additional cost required to achieve a more conservative design in a critical machine element would have been negligible if considered during the early stages of the detail design.

Design integration between the structural design and machine protection systems was lacking since normal operational conditions (chute blockage and boom belt slip) could introduce excessive loading conditions on the boom structure which were not envisaged nor catered for by the machine protection systems.

Furthermore, the incorrect commissioning of protection systems (boom belt under speed and hydraulic cylinder over pressure set point) and the inadequacy of protection systems (stockpile detection sensors) jeopardised the safety of the equipment.

Under boom overload conditions the malfunctioning of the hydraulic system, which manifested as a vibration on the luffing cylinder and subsequently an oscillation motion of the boom structure, could introduce abnormal loading conditions.

It could be argued that the supervision of the machine protection systems' commissioning by an experienced structural engineer from the OEM, with a thorough understanding of critical machine protection systems may have prevented the collapse, although there remain uncertainties regarding the integrity of the critical tie-beam connection as discussed in detail in Chapter 5.

Emergency loading conditions could have been identified during a design risk assessment early on in the design during which the structural designer could have validated design parameters such as material density, which would have resulted in a more robust tie-beam connection design.

6.3 ADDRESSING CURRENT DESIGN PRACTICE SHORTCOMINGS

The intent of this study is ultimately to improve structural safety on mobile BMH equipment without tarnishing the reputation of established OEMs serving the industry. Although the survey indicated that OEMs have consensus that ISO 5049-1 (1994) is sufficient to facilitate the design of safe mobile BMH equipment, literature seems to suggest otherwise (23, 42, 48).

The author acknowledges the specialised expertise of reputable OEMs which has been developed and refined over many years with the successful delivery of a significant number of mobile BMH machines to the mining and minerals industry. It is further acknowledged that some OEMs will circumvent shortcomings in design standards better than others. Although readers, especially those representing the opposition OEM’s not involved in any of the failures covered in this document, may wish to distance themselves from the errors or shortcomings which led to the failures as discussed in the preceding case studies, it is nevertheless the author’s observation that no supplier can safeguard itself against the possibility of equipment failure or design error. The nature of the mobile BMH equipment environment is complex and, in spite of established practice and design rules, the possibility of human error cannot be ruled out. Although reputable and well-established OEMs providing mobile BMH equipment follow different design approaches within the constraints of an international standard i.e. ISO 5049-1 (1994), the author, representing the viewpoint of an end-user, is of the opinion that shortcomings are prevalent within the current design practice generally followed. The outcome of the survey results referred to above, and the case studies already discussed, provide the basis for the author’s recommendations for future designs of mobile BMH equipment as discussed in this section.

6.3.1 RISK-BASED DESIGN APPROACH

Although experienced designers may argue that machines provided within the range of a specific brand are designed on the same basis, specific site conditions may vary to such an extent that loading conditions may occur of which the designer is not aware. In addition to design rules provided in ISO 5049 (1994), it is therefore recommended that a risk assessment be conducted once the supply contract has been awarded. The AA 248/2 (2010) (3) standard provides guidelines pertaining to design risk assessment.

Representation by experienced operational personnel is essential so that operational practice can be aligned with the basis of the design. Of particular importance are matters relating to accidental loading, e.g. erection loading conditions, the collapse of the stockpile onto a bucket wheel reclaimer during reclamation, the bucket wheel working loose resulting in machine imbalance, and aspects regarding the collision of machines into other stockyard equipment or the stockpile. The bulk density of raw material may also vary quite significantly, depending on the mining conditions e.g. where previously undermined areas are open-cast mined, instantaneous slugs of rock may be handled, the bulk density of which is far greater than that of the average raw material handled in the system.

Special conditions, which the equipment designer may not be aware of, may be present at certain sites. E.g. Spontaneous combustion is often encountered where previously undermined coal reserves are accessed with open cast mining methods. During windy conditions, excessive raw coal temperatures may be periodically experienced at raw coal stockyards.

6.3.2 REDUNDANCY

The outcome of the survey clearly shows that OEMs have consensus that designs carried out in accordance with the AS 4324-1 (1995) will result in heavier and more expensive mobile BMH equipment because more stringent design rules are dictated than in ISO 5049-1 (1994). AS 4324-1 (1995) was, indeed, developed in response to a number of machine failures which occurred in Australia (48). Discussions with OEM specialists (30, 56) indicated that certain clauses within AS 4324-1 (1995) are deemed to be too onerous. As mentioned in Chapter 2, the latter Standard is currently under review and will be amended in the foreseeable future.

The author acknowledges that capital constraints on projects and subsequent pressure on OEMs to provide competitive solutions are realities within the mining and minerals industry. From an end-user perspective, it is nevertheless considered that more conservative designs for long-term operations are a worthwhile investment, and also provide room for possible future capacity upgrades. The cost and production interruption associated with remedial work to strengthen structures once the equipment has been erected and commissioned is significant. Refer to Chapter 3, Case Study 4. The author does not advocate an uneconomical design approach, but rather that designers make allowances for greater margins of safety where the risk assessment highlights the probability of loading conditions not specifically addressed by the applied design standard.

The author is furthermore of the opinion that safety critical elements, such as tie systems and luffing mechanisms, should be designed with double redundancy, but at a reduced safety margin. Once again, these considerations should be informed by the outcome of the risk-based design approach as discussed above. The additional costs associated with double redundancy on safety critical items, are considered a worthwhile investment for a long-term project. If planned carefully, the additional cost may be insignificant for double redundancy on tie systems. Luffing mechanisms are, however, more expensive and will attract additional capital expenditure.

6.3.3 MACHINE PROTECTION

As discussed in Chapter 2, both ISO 5049-1 (1994), FEM SECTION II (1992) and AS4324-1 (1995) focus on the design of the steel structures and some mechanical aspects associated with mobile BMH equipment. Although additional parts that would address mechanical, electrical and other aspects were initially planned for all of these standards, they were never published. The said standards available to the mobile BMH equipment industry are therefore silent on rules and requirements for machine protection systems. By implication, it is therefore left to the OEM to provide protection systems which are deemed adequate to ensure safe equipment.

The author is aware of mobile BMH equipment contract agreements which were structured in such a way that the structural design and instrumentation design packages were done by different parties. In this instance, the instrumentation design was done only after the structures were fabricated and fully erected (35). A single point of overall responsibility is highly desirable to minimise risk associated with design integration and potential misunderstanding between designers. Where it is desired to have certain portions of the overall design or construction contracted to third parties, the author strongly recommends that the OEM is paid a management fee to retain overall responsibility for the entire machine(s).

The design of machine protection systems calls for an integrated design approach which is discussed below.

6.3.4 INTEGRATED DESIGN APPROACH

The lack of interdisciplinary design integration, as discussed in case studies 2, 4 and 5, is of concern. This is probably a highly controversial topic which design engineers would generally not want to embrace. Of course some BMH equipment OEMs will address this engineering challenge better than others. Unfortunately the facts presented in the above case studies demonstrate that design engineers often design with an engineering discipline-specific approach, without the required understanding of design details from counterparts representing the other engineering disciplines. This may have a direct influence on the overall performance of the equipment. The author acknowledges that discipline-specific specialists are nevertheless required for the successful design of mobile BMH equipment, the appeal is merely for better design integration, which is not based on perception, but rather on a thorough understanding of interdependence between engineering disciplines. Although the competitive nature of the mobile BMH industry generally leads to a tendency amongst OEM's not to openly share design content with the respective client representatives, it would be advantageous to both parties, especially where the client appoints a third-party design auditor. While it is more common for larger corporate clients to have skilled engineering staff assigned to capital projects for the purposes of engineering oversight, smaller enterprises generally rely entirely on the OEMs for the successful delivery of functional mobile BMH equipment as specified in the supply contract. Liaison between the OEM's design engineers and the respective client's engineering discipline leads is invaluable for ensuring successful project delivery. Furthermore, larger corporate clients often have a number of operations where the same or similar mobile BMH equipment may be utilised in ways other than what was envisaged under the supply contract. The input from operational personnel who are responsible for the daily operation and general maintenance of existing equipment, must not be underestimated, but the ability of such individuals to influence new designs remains largely dependent on their skill and experience.

Figure 6.1 below illustrates a non-ideal, but nevertheless common design team organisation structure.

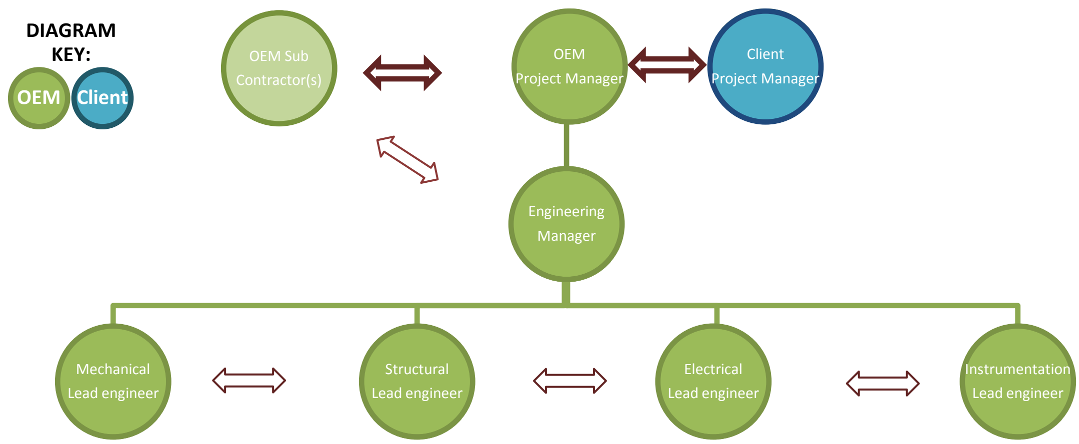


Figure 6.1: Non-ideal design team structure resulting in a poor level of design integration.

The following aspects characterise a non-ideal team structure:

- Engineering disciplines are largely compartmentalised without a broader understanding of the overall mobile BMH equipment design. A multi-disciplinary interface approach is lacking or inadequate.
- No engineering input, oversight, liaison or review is obtained from the client.
- Specific design requirements are agreed between project managers.

An integrated design team organisation structure which is conducive to better design integration with a systems design approach is depicted below in Figure 6.2.

Figure 6.2: Ideal design team organisational structure facilitating design integration.

The following aspects characterise an ideal team structure:

- Within the OEM's design team organisational structure, there is a free flow of information *directly related to design interfaces* between engineering disciplines *without interference with discipline-specific matters*.
- Design interfaces are approached as *an integrated system* with input from relevant role players across engineering disciplines as a team effort.
- The respective engineering disciplines have a sound understanding of how equipment selections and systems dictated by engineering counterparts influence their individual designs.

- The client owner's team participates in the design scope definition and design risk assessment. Engineering input, oversight, liaison and progressive review is provided by the relevant representation from the client.
- Specific design requirements are agreed between the OEM and client owners team within the agreed contractual arrangement.
- There is a free flow of information between the discipline engineers from the owner's team and their OEM counterparts responsible for the design, without compromising the latter party's intellectual property rights.

Although it is expected that most OEMs will embrace and advocate the latter integrated model, case studies unfortunately suggest that the non-ideal model shown in Figure 6.1 is commonly encountered within the industry. It must be noted that the integrated model would require a high level of trust between the OEM and the owner. In addition to conventional contract agreements, ground rules would have to be established for legal and commercial complexities which may arise as a consequence of the integrated model, e.g. the owner would have to carry the costs where the need for design enhancements or additional protection are identified during the detail design if the specific features were not covered within the scope of the tender. The author is aware of a recent project where the integrated model was successfully used for a significant contract value which included the supply of four large BMH machines.

Figure 6.3 below summarises how inadequate design integration across engineering disciplines contributed to the machine failure discussed in Case Study 2.

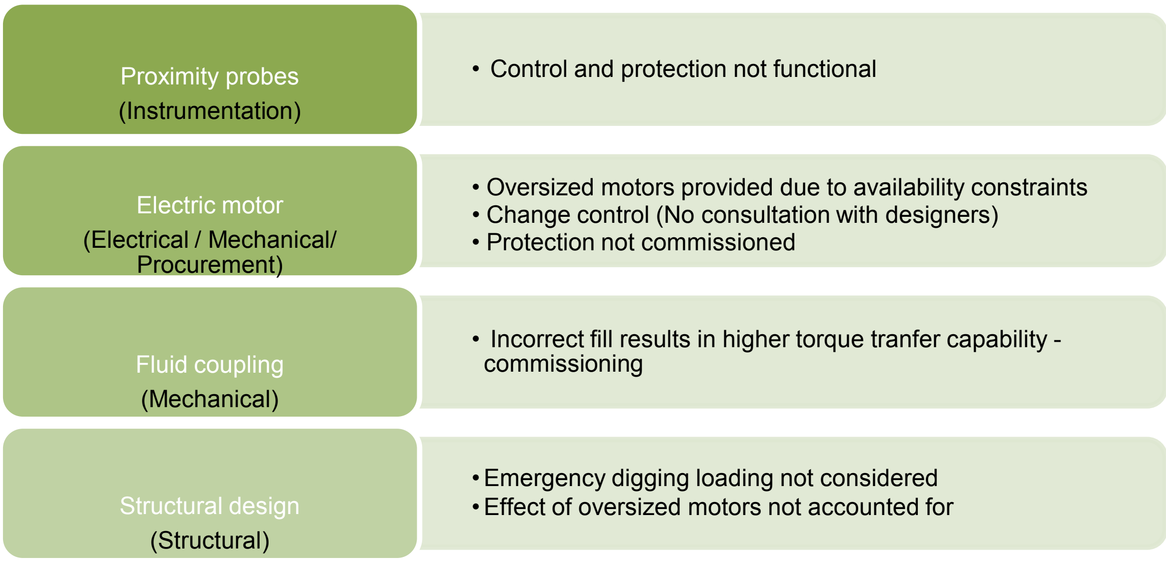


Figure 6.3: Inadequate design integration as observed in Case Study 2.

The management of design interfaces between the various engineering disciplines becomes even more challenging when subcontractors are utilised. Sub-contractors should always be employed under the main contractor so that the overall responsibility for project delivery remains with a single party i.e. the OEM.

6.3.5 BASIS FOR STRUCTURAL DESIGN

ISO 5049-1 (1994) is based on an allowable stress design approach. As discussed in Section 6.1 above, reputable OEMs utilise this standard as a loading code (only) while applying limit states design standards such as DIN 18800-1 (1990) for the design of structural elements. FEM SECTION II (1992) (31) is then still used as a basis for the design of mechanisms and fatigue life. Sparse guidance on the design of structural elements is provided in ISO 5049-1 (1994), and OEMs invariably reference a number of additional standards and guidelines such as FEM SECTION II (1992) and DIN 22261 (2006) (28) and IIW's *Recommendations for fatigue design of welded joints and components* (39), which covers allowable stress design principles in much greater detail.

The application of allowable stress standards in conjunction with limit state design standards is nevertheless to be done with caution. From the survey conducted, it can be noted that some OEMs are strongly opposed to the above practice and subsequently follow a strict allowable stress design approach. From an end-user perspective, the author finds it unacceptable that well-established, reputable OEMs, could have such different views on this fundamental matter.

6.3.6 FINAL HANDOVER AND ACCEPTANCE

The Anglo American Specification AA 114/1 (2007) (1), applicable to permanent structures, requires that the structural design engineer, where applicable, inspect the constructed works for conformity with the design. The role that the structural design engineer fulfilled in the commissioning and final acceptance of mobile BMH equipment discussed in case studies 2, 4 and 5 above is deemed to be insufficient. The author acknowledges that engineering disciplines are interdependent and structural design engineers cannot be expected to have the same proficiency in mechanical, electrical, and control and instrumentation engineering disciplines in addition to their structural engineering competence. It is nevertheless proposed that the structural design engineer play a more prominent role in the final handover of mobile BMH equipment because of his/her in-depth understanding of the sensitivity of the structural design and loading that can be accommodated by the structure. For larger machines, the optimisation of the ballast mast is usually part of the commissioning procedure which is often supervised by a structural engineer anyway. It is therefore proposed that the structural engineer also be closely involved with the verification of alarms and set points associated with machine protection systems, in conjunction with the specialists responsible for the design and commissioning of these systems, before final handover of mobile BMH equipment. This necessitates that the knowledge of the structural engineer responsible for final acceptance of mobile BMH equipment prior to final handover to the owner, must extend beyond his/her discipline-specific engineering expertise.

6.4 REVISION OF ISO 5049-1 (1994)

The revision of ISO 5049-1 (1994) is, in itself, deemed to be somewhat of a contentious issue. It can be argued that mobile BMH equipment OEMs have a vested interest *in not having an all-inclusive updated design standard* since the lack thereof provides a loophole when things do go wrong. When a collapse occurs and the OEM can prove design compliance with the current international standard, ISO 5049-1 (1994), the owner has difficulty proving that the OEM did not fulfil the anticipated design obligation, unless specific additional requirements were documented in the contract agreement.

The need for a review of ISO 5049-1 (1994) has been raised in the past at a number of events (44) including the International Conference on Structures for Mining and Related Materials Handling held in South Africa in 2012 (43). In view of the case studies discussed in preceding chapters and the issues highlighted above, the author supports such a review and wishes to encourage stakeholders to drive this matter at the appropriate forums in order to obtain safer mobile BMH equipment designs in future. To strengthen this appeal, it must be considered that the technical committee responsible for the development of AS 4324.1 (1995) considered ISO 5049-1 (1994) inadequate for providing safe mobile BMH equipment to the Australian commodity based economy, and subsequently developed the aforementioned standard. It is not implied that AS4324.1 (1995) should become the norm for future designs, but rather that ISO 5049-1 (1994) be revised.

The author therefore proposes that a technical committee be tasked with the revision of ISO 5049-1 (1994) with specific focus on the following matters:

1) An integrated systems design approach where specific rules and guidelines are provided regarding machine protection systems. These design rules should be based on a risk assessment and the efficiency of proposed mechanical and electronic controls.
2) The compilation of design rules pertaining to loading control on reclaimers, taking into consideration the requirements already detailed in AS 4342.1 (1995).

3) Guidelines for the interpretation of the analysis results obtained through the application of finite element methods in conjunction with the standard (30), in order to mitigate disputes on highly stressed regions in the structure.
4) Guidelines for designers who wish to follow a limit state design approach.
5) Consideration should be given to the revisions envisaged to the AS 4324-1 (1995) standard facilitated by the Australian Standards Committee ME43.

Additional aspects for consideration in the standard were highlighted in personal correspondence and discussions with OEM specialists (30) as listed below:

1) The use of alternative light-weight and compound construction materials.
2) The use of new rope technology for tie systems.
3) New calculation principles, which were introduced after the initial publication of ISO 5049-1 (1994).

6.5 SUMMARY

In this chapter, the current design practice followed by reputable OEMs within the mobile BMH equipment industry was discussed. The shortcomings in the current design approach, as perceived from an end-user's perspective, were explored, while recommendations and proposed solutions were provided. An integrated systems design approach, based on a comprehensive risk assessment was proposed in the absence of an all-inclusive international design standard for mobile BMH equipment. An appeal was made to the stakeholders to work together in bringing about a revised version of ISO 5049-1 (1994) Standard while specific focus areas were highlighted.

A summary of the case studies highlighting the main factors which contributed to the respective failures, and preventative measures to avoid repeat incidents in future, is provided in Table 6.1 below for quick reference.

Table 6.1: Summary of case studies with preventative measures to avoid repeat incidents.

Case study		Contributing factor	Category	Contribution to failure	Preventative measure
Waste spreader	1	Wind overload - Wind loading. Concession granted.	D	significant	Loading to be assessed in accordance with design standard understanding of local conditions. Design risk assessment.
	2	Overload - Encrustation allowance taken as minimum requirement from ISO 5049-1.	D, M	moderate	Loading to be assessed in accordance with design standard understanding of local conditions. Design risk assessment. Improved awareness of maintenance personnel.
	3	Insufficient stiffness of structure	D	moderate	Assess serviceability requirements in accordance with design standards.
Portal reclaimer	1	Overload - malfunctioning material proximity probes. (Protection systems)	C, M	significant	Machines must not be used unless protection systems are fully functional.
	2	Overload, drive protection systems. (Electronic shear key not set. Fluid coupling overfilled.)	DI, C	significant	A risk based, systems design approach, which is integrated across engineering disciplines is required.
	3	Procurement of oversized motors.	CC	significant	Change control process must ensure that design engineers are consulted when equipment specification sheet is not complied with.
	4	Structural design insufficient.	DI	moderate	Emergency loading conditions can be identified with a risk based, integrated systems design approach.
Ship loader	1	Change control during when decommissioning a cleaning system.	M	significant	A risk assessment forming part of the operations' change management procedure should have identified the potential threat.
	2	Individual responsibility and vigilance.	M	significant	Continued awareness of a safety culture at operations.
Drum reclaimer	1	Long travel control systems not fail safe.	DI	significant	Emergency loading conditions can be identified with a risk based, integrated systems design approach.
Slewing stacker	1	Design loading underestimated	D	significant	Proper communication. Compliance with design standards.
	2	Improper design of critical structural connection.	D	significant	Compliance with design standards. Design competence.
	3	Bolts, susceptible to hydrogen embrittlement, used on critical connection.	MS	significant	Design competence.
	4	Protection systems insufficient to protect machine overload.	D, DI	significant	Emergency loading conditions can be identified with a risk based, integrated systems design approach.
	5	Incorrect commissioning of protection systems.	C	significant	Oversight of commissioning of protection systems and final acceptance of the machine by competent person.
	6	Operator abuse	A	unknown	-

Legend:
A - Abuse
C - Commissioning
CC- Change Control
D - Design
DI - Design integration across engineering disciplines
M - Maintenance management
MS - Material selection

7 CONCLUSION

7.1 LITERATURE REVIEW

The relevant standards used by reputable international OEMs specialising in the design, supply and commissioning of mobile BMH machines were explored in Chapter 2. It was pointed out that the recognised standards i.e. FEM SECTION II (1992), ISO 5094-1 (1994) and AS 4324-1 (1995), used for the design of mobile BMH equipment, focus largely on structural design and only some mechanical aspects. Additional chapters or parts, which would address mechanical, electrical and other aspects of design, were originally planned for the above-mentioned standards, but never published. Only DIN 22261-2 (2006) and AS 4324-1 (1995) have some reference to the design of machine protection systems. The latter standard is currently under revision. Survey results, as provided in Appendix C, indicate that the design approaches followed by various leading OEMs are not consistent in the application of design standards. A summary of the review of the relevant literature is provided in Section 2.10.

7.2 ROOT CAUSES OF FAILURES

A number of case studies were discussed in order to fulfil the first study objective, which was to provide an insight into the identification of typical root causes of failures, and the complex interaction of such causes in most catastrophic events associated with mobile BMH equipment:

The failure of a waste spreader was discussed in Case Study 1, where it was concluded that the root cause of failure could be attributed to overloading conditions exerted on a machine structure of insufficient stiffness due to a combination of design errors and the underestimation of loading conditions in the design. It was furthermore highlighted that ISO 5049-1 (1994) provides no guidelines regarding serviceability criteria for mobile BMH equipment. Refer to Section 3.1.3.

The failure of a portal reclaimer was covered in Case Study 2, where the malfunctioning and incorrect commissioning of machine protection systems resulted in accidental loads which were much higher than were originally anticipated in the structural design. A number of compounding factors directly related to protection systems were highlighted. The lack of design integration between engineering disciplines was emphasised. Refer to Section 3.2.3.

The collapse of a ship loader was discussed in Case Study 3 where the importance of change management in the operational processes, pertaining to structural integrity, was highlighted. The failure was caused by overloading of the structure which was intended to operate under spillage-free conditions. Refer to Section 3.3.4.

7.3 COMPLEXITY OF FAILURES

The second objective of the study was to demonstrate the complexity of typical collapses and the systematic investigation approach which is required to establish the root cause of failures. In order to achieve this, a stacker failure investigation was used as the basis for the main case study and was covered in Chapter 5. The complexity of typical collapses and the systematic investigation approach which is required to establish the root cause of failures was discussed. Although some uncertainty regarding the failure will probably never be resolved, it was nevertheless shown how numerous factors contributed to the disaster. It was ultimately concluded that design deficiencies contributed to a marginal design of critical connections of which the carrying capacity was further exacerbated by defective bolts. The absence, malfunctioning and incorrect commissioning of machine protection systems did not prevent an overload condition which led to the catastrophic collapse of the stacker. The approach followed during the investigation of this collapse was discussed in Chapter 4.

7.4 LIMITATIONS OF ISO 5049-1

The third study objective was to demonstrate some limitations of the internationally recognised design standard, ISO 5049-1 (1994), which is most commonly used in the consulting industry worldwide.

These limitations were discussed in various case studies and include:

- Serviceability criteria for mobile BMH structures are not addressed. Refer to Section 3.1.3.
- The standard focuses predominantly on matters pertaining to structural design and provides no rules or guidance to facilitate an integrated design approach across engineering disciplines. Refer to Section 3.2.3 and Section 3.4.3.
- No rules or guidance are provided for machine protection systems. Refer to Section 3.2.3 Section 3.4.3 and Section 5.15.

The latter item is considered very significant. It was demonstrated in Case Study 2 (which covered the failure of a portal reclaimer), Case Study 4 (which covered the structural damage done to a drum reclaimer) and the main case study discussed in Chapter 5, that insufficient or incorrectly commissioned machine protection systems were the direct cause or contributed substantially to the failure of the machines in question.

Full compliance with ISO 5049-1 (1994) does not necessarily guarantee safe mobile BMH equipment which can withstand the conditions to which it will be exposed during its operational life. Latent defects may only manifest in a catastrophic collapse after years of successful operation. A functional machine is therefore not necessarily a safe machine. Although ISO 5049-1 (1994) is internationally utilised as a basis for the design of mobile BMH equipment, it has limitations and shortcomings which must be recognised. While a design standard can never be a substitute for a pragmatic design approach, the need for a revised all-inclusive international standard for the design of mobile BMH equipment was nevertheless highlighted.

Shortcomings, as perceived from an end-user's perspective, regarding current design practices followed by reputable OEMs in the supply of mobile BMH equipment, were discussed. Considering the limitations of ISO 5049-1 (1994), an integrated risk-based design approach is recommended to facilitate the design of safer mobile BMH equipment.

The author proposed that a technical committee be tasked with the revision of ISO 5049-1 (1994) with specific focus on the items as detailed in Chapter 6 to facilitate the design of safer mobile BMH equipment in the future and to mitigate potential disputes between owners and OEMs.

7.5 INVESTIGATION METHODOLOGY

The proposal of a methodology for failure investigation of mobile BMH equipment and recommendations to facilitate the identification of the root cause of failure were stated as being the fourth objective of this study. Such a methodology for the successful investigation of catastrophic failures associated with mobile BMH equipment was proposed in Chapter 4, based on a retrospective view of the processes followed during the investigation of the main case study discussed in Chapter 5. An outline model defining a typical investigation team was provided and key functions were defined for the major role players.

The relevance of the proposed methodology was demonstrated by evaluating probable investigation activities associated with the case studies covered in Chapter 3.

7.6 PROPOSED DESIGN APPROACH

Although most readers would have expected a high level of interdisciplinary design integration associated with the design of engineering interfaces when it comes to mobile BMH machines, the lack and absolute need therefore was reiterated in Case Study 2 (which covered the failure of a portal reclaimer), Case Study 4 (which covered the structural damage done to a drum reclaimer) and the main case study discussed in Chapter 5.

A systems design approach, which is properly integrated between the respective engineering disciplines and is based on a comprehensive risk assessment, was recommended in Chapter 6 where observations from the case studies in this regard were discussed in Section 6.2.

Addressing the shortcomings in current design practices, as observed from the case studies, was discussed under Section 6.3 and concluded in Section 6.6. Measures to avoid similar future incidents were provided in table format for quick reference.

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9 APPENDIX A

Terminology and abbreviations referred to in the study are provided in this section.

9.1 TERMINOLOGY

Brazier buckling:	When an initially straight tube is bent uniformly, the longitudinal tension and compression which resist the applied bending moment also tend to flatten or ovalise the cross-section. This in turn reduces the flexural stiffness EI of the member as the curvature increases (9).
Bogie:	A chassis or framework carrying wheels, thus serving as a modular subassembly of wheels and axles.
Belt extension:	The conveyor is extended by the insertion of additional belt and conveyor structure. The tail may be moved backwards or the head may be advanced.
Belt fill factor:	The ratio between the actual belt loading and the full belt capacity expressed as a percentage. Also refer to flooded belt below.
Bulk density:	The mass of a material (including solid particles, moisture and voids) per unit volume.
Chute:	The plate-work construction required to facilitate the transfer of material from a feed conveyor to a receiving conveyor or storage facility.
Conveyor:	A belt conveyor.
Conveyor take-up:	A device required to accommodate belt stretch and to maintain conveyor belt tension so that power can be transferred from the drive pulley onto the belt.
Conveyor gantry:	A latticed frame structure utilised to house an elevated conveyor.
Conveyor walkway:	An access walkway at the side of an elevated conveyor.
Cylinder clevis:	The end mounting on a typical hydraulic cylinder which facilitates load transfer.
Dragline:	A mobile machine used for over burden removal, the bucket is attached by cables and operates by being drawn towards the machine.
Encrustation:	Accumulation of spillage or material sticking to the machine.

End carriage:	A bogie.
Fitted bolts:	Special bolts of which the shanks are machined and installed in reamed holes to a specified fitting tolerance.
Flooded belt:	A conveyor belt loaded such that spillage occurs, (approximate fill factor of 140 %).
Long travel:	The longitudinal movement of a stockyard machine along a stockpile.
Luffing:	The up or down movement of a stacker boom.
Reclaimer:	A machine for the recovery of material from a stockpile onto a conveyor belt.
SCADA:	(Supervisory control and data acquisition) is a type of industrial control system which monitors and controls industrial processes.
Ship loader:	A machine utilised to load bulk materials into a cargo ship by means of conveyor systems.
Stacker:	A machine for delivering material onto a stockpile.
Spreader:	A stacker (see stacker).
Stockpile:	A quantity of material in storage.
Stockpile bed:	The surface onto which material is stacked and reclaimed.
Stockpile, longitudinal:	A stockpile associated with a stacker travelling along rails in a stockyard.
Stockyard:	An area which contains any number of stockpile beds.
Stockyard equipment:	Also referred to as stacking and reclaiming equipment. A collective term referring to materials handling machines utilised in a stockyard.
Slewing:	The angular rotation of a stacker or reclaimer boom.
Tripper, moving:	Also referred to as a tripper car. A structure mounted on bogie wheels which are coupled to a stacker, so that the material from the yard conveyor belt is elevated for delivery into the receiving chute of a stacker at any position along the longitudinal stockpile.

Turn of the nut method:	A pre-tensioning method whereby extension of the bolt is induced by a prescribed rotation after a "snug tight" condition was obtained on the joint.
Yard conveyor:	A belt conveyor utilised within the stockyard area.

9.2 ABBREVIATIONS

AA	Anglo American PLC, A global mining company
BMH	Bulk materials handling
CEMA	Conveyor Equipment Manufacturers Association
FEA	Finite element analysis
HE	Hydrogen embrittlement
BHN	Brinell hardness number
HSFG	High strength friction grip
OEM	Original equipment manufacturer
PLC	Programmable logic controller
SAISC	South African Institute of Steel Construction

10 APPENDIX B

The wind loading calculations applicable to the collapse of the waste spreader discussed in Chapter 3.1.2 are provided below:

The resultant wind force on latticed structures is expressed as:

$$F = C_f q_z A_e \quad (1)$$

Where:
C_f = the force coefficient.
A_e = the projected area normal to the wind speed [m^2].
q_z = free stream velocity pressure of wind at height z [N/m^2].

The free stream velocity is given by Equation (2).

$$q_z = k_p V_z^2 \quad (2)$$

Where:
k_p = a constant dependant on site altitude
V_z = characteristic wind speed at height z [m/s].

- From SANS 10160-1 (1989), clause 5.5.3.1, by interpolation k_p = 0,52 at a site altitude of approximately 1200 m above mean sea level.
- If the terrain category is conservatively taken as Category 2, even though site photographs suggest Category 1, SANS 10160-1 (1989), clause 5.5.3.1, the wind speed multiplier k_z = 1,164 for Class B structures.
- For a local wind speed of 40 m/s, the adjusted wind speed becomes 40 x 1,164 = 47 m/s

Equation (3) below shows the free stream velocities for a characteristic wind speed of 47 m/s at 60 m height.

$$q_z = k_p V_z^2 = (0{,}52)(47^2) = 1149 N/m^2 \quad (3)$$

The consultant's modified design criteria catered for a wind speed of 30 m/s:

$$q_z = k_p V_z^2 = (0{,}52)(30^2) = 468 N/m^2 \quad (4)$$

From Equations (3) and (4), the resultant wind force F, was therefore underestimated by a factor of: 1149/468 = 2,5

11 APPENDIX C

Survey responses from participating OEMs are contained in Appendix C.

It must be noted that:

- The company details were omitted on the last response on request of the participating OEM.
- Where permission could not be obtained, the responses were not published.
- Where required, minor editing was done without changing the original intent of the OEM responses obtained. (Some participants are not fully conversant in English)

11 - 2

Company:	**Huadian Heavy Industries, (HHI)**
Information provided by:	Huang Zhenguo
Designation:	Chief designer
Design experience (years):	9
Number of bulk handling machines desiged:	5

The following questions are specifically related to the design of mobile equipment for continious handling of bulk materials

	Question:	**Response:**
1	To what extent are the following standards and specifications used if no requirement from the client is provided:	
	a) ISO 5049	It is required by domestic projects to do the structure design.
	b) FEM 2.131 / 2.132	It is required by domestic projects to do the structure
	c) AS 4324.1	It is only required by Australian projects to do the structure
2	What other standards and specifications are utilised?	All the related Australian standards and client's
3	Is the structural design based on allowable stress or a limit state approach?	It is based on an allowable stress approach.
4	Do you consider it general practice to apply a limit state approach based on loading obtained from ISO 5049-1?	No, all current designs are using an allowable stress approach,which has been adopted in Australian projects.
5	Do you consider the requirements from ISO 5049-1 to be adequate to deliver safe structural designs?	Yes, for structures it is enough. Australian standard is considered include too many extreme cases thus making equipment heavier with more redundancy.
6	How do you go about adressing items where ISO 5049-1 is perhaps not suitable or comprehensive enough?	The client, auditor and designer can sit together to have disscussion to meet a extra requirements as agreed on the concerned items.
7	Should the client requires compliance with AS4324.1, what percentage increase do you expect on the structural steel mass and overall cost.	Compared with ISO standards, our current experience is that the structure mass may increase about 20 to 30 %, overall cost may increase about 20 to 25 %.
8	To what extent are basic hand calculations done?	Mechanical power selections, mechanical components selections, fatigue calculations, shafts calculations, etc. are done based on basic hand calculations.
9	How heavily do you rely on finite element analysis? Please comment on the use of bar elements vs shell and brick elements.	Structures must be certified by FEA.Shell elements are used.
10	Would you prefer to design machines more conservative if economical constraints were less severe or do you feel that machines are designed robustly enough to withstand accidental loads encountered during its life cycle?	It is preferred that machines are designed robust enough to withstand possible accidental loads encountered during its life cycle.
11	To what extent are previous designs modified for new machine designs?	Due to different requirements on the foundation arrangement, only the structure type can be applied while details must be revised accordingly.
12	How concerned are you about operator abuse and, are you satisfied that your equipment is suitably equipped with protection systems?	Operators must comply with the operation manual and site regulations. Yes, good protecting system will result in less accidents.
13	Do you ever do a design risk assesment with the client before commencing with designs?	Yes, before contract award, the client will invite us to see what design risks exist. Usually a mitigation team will be despatched to assist with the design.
14	How do you integrate the design of the control and machine protection systems with the structural design?	Electrical design limitation will be considered in structural design load calculation. The calculated loads will be used as input into the FEA model.
15	To what extent is the structural design engineer involved with the commissioning and final release of the machine?	Slewable machines usually require that the superstructure balance is checked. The structrural engineer will pay attention to site commissioning data to make adjustments. Machines are required be be weighed, the results are to be checked to meet AS4324.1 5 % deviation is permitted - if within the 5 % tolerance, final release of the machine can be done.

Company:	**SANDVIK Mining and Construction, Materials Handling GmbH & Co KG**
Information provided by:	Frank Feger (Leoben, Austria)
Designation:	R&D/Engineering Manager Surface Mining (IPCC) & Materials Handling
Design experience (years):	>10 years
Number of bulk handling machines designed:	I have in different roles been responsible for more than 75 mobile machines for bulk materials handling (stockyard & port equipment) as well as mining equipment (soft rock & hard rock)

The following questions are specifically related to the design of mobile equipment for continious handling of bulk materials

	Question:	**Response:**
1	To what extent are the following standards and specifications used if no requirement from the client is provided:	In case there is nothing specifically required we would choose ISO & FEM. In Australia the AS4324 code will always apply.
	a) ISO 5049	
	b) FEM 2.131 / 2.132	
	c) AS 4324.1	
2	What other standards and specifications are utilised?	In addition to the three codes mentioned, the DIN 22261 is commonly followed for calculation for mining equipment.
3	Is the structural design based on allowable stress or a limit state approach?	Our design calculations are issued based on allowable stress.
4	Do you consider it general practice to apply a limit state approach based on loading obtained from ISO 5049-1?	No, we do not follow this approach.
5	Do you consider the requirements from ISO 5049-1 to be adequate to deliver safe structural designs?	Yes, we consider ISO 5049 adequate for the calculation of safe designs for bulk solids handling equipment.
6	How do you go about adressing items where ISO 5049-1 is perhaps not suitable or comprehensive enough?	We consult other relevant codes such as e. g. FEM, DIN, AS, etc.
7	Should the client requires compliance with AS4324.1, what percentage increase do you expect on the structural steel mass and overall cost.	Depending on the machine type, we would assume a 10 - 15 % increase in the mass of steel structures due to several load scenarios and safety factors specifically considered for AS-Code and also specific requirements for secondary structures. Increase in mass of the various assembly groups as well as the specific load scenarios relate to higher power requirements creating bigger and thus more expensive drive units. Commercial impact of the latter can only be determined for the particular projects.
8	To what extent are basic hand calculations done?	Calculation results are derived from calculation programs and also include results from pre-configured excel calculation tables. We spot check by hand calculation. All results obtained from FEA, spread sheets or hand calculation are reviewed by structural engineers, experienced in the design of mobile bulk materials handling equipment.
9	How heavily do you rely on finite element analysis? Please comment on the use of bar elements vs shell and brick elements.	FEA is mostly used in support of design calculations and in areas where high stresses are expected from complex loading scenarios.
10	Would you prefer to design machines more conservative if economical constraints were less severe or do you feel that machines are designed robustly enough to withstand accidental loads encountered during the life cycle?	In our opinion the machines are sufficiently designed and as per some codes even over-designed. There is no need to change to more conservative approach with or without considering commercial implications. The load assumptions together with the combinations have to be correct, other than that the actual as well as more conservative assumptions may lead to false designs.

Continued on next page

… SANDVIK response continued:

11	To what extent are previous designs modified / adopted for new machine designs?	The machines are designed according to the customer's requirements and the codes we are contractually bound to. In majority of the cases that means that the equipment is to very large extent subject to new design.
12	How concerned are you about operator abuse and, are you satisfied that your equipment is suitably equipped with protection systems?	We have sufficient protection in place, but the protection (even in some instances from a safety point of view) are balanced by economical and practical contraints. No system can fully safeguard against operator willful or negligent damage. Even systems would require redundancy and redundancy on top of the redundancy to ensure a higher likelihood of "no damage".
13	Do you ever do a design risk assesment with the client before commencing with designs?	A HIRA and HAZOP is done and it is anchored in our design process to complete a risk assessment.
14	How do you integrate the design of the control and machine protection systems with the structural design?	Structural engineering defines the limit settings as per the relevant codes. Handover sheets with those limits are supplied to the group implementing the control system and this is contained in the FOD, which is the basis for the operating of the machine. The set values are tested in FAT test and during the commissioning of every machine. After handover the maintenance and care of these systems and hardware transfer to the operator/owner.
15	To what extent is the structural design engineer involved with the commissioning and final release of the machine?	The structural engineer will calculate and get feedback on the cylinder loads as per the as-built machines. This gives an accurate indication of the actual achieved COG of the superstructure and with that the stability of the machine. This information is fed back to the structural calcualtion and the machine is rebalanced to the received values to maintain satisfactory cylinder pressures on an iterative basis. Designs are audited. Furthermore for critical functions we empirically determine the final weights as well as COG's by jacking up (cylinder forces, edge weights, etc) and redo calculations if necessary based on the results received.

Company:	**Tenova Takraf**
Information provided by:	Tobias Schiller
Designation:	Engineer
Design experience (years):	(Undisclosed)
Number of bulk handling machines desiged:	(Undisclosed)

The following questions are specifically related to the design of mobile equipment for continious handling of bulk materials

	Question:	**Response:**
1	To what extent are the following standards and specifications used if no requirement from the client is provided:	In principle we prefer to use FEM or ISO for the design of stockyard machines.
	a) ISO 5049	Yes, will be used.
	b) FEM 2.131 / 2.132	Yes, will be used.
	c) AS 4324.1	Yes, will be used.
2	What other standards and specifications are utilised?	We use among others the DIN 22261, DIN ISO,and EN. In addition to these we conform to all international standards.
3	Is the structural design based on allowable stress or a limit state approach?	We mainly use allowable stress design. In exceptional cases or according to customer requirements we use also limit stress design.
4	Do you consider it general practice to apply a limit state approach based on loading obtained from ISO 5049-1?	In terms of ISO 5049 we use the allowable stress design.
5	Do you consider the requirements from ISO 5049-1 to be adequate to deliver safe structural designs?	Yes.
6	How do you go about adressing items where ISO 5049-1 is perhaps not suitable or comprehensive enough?	ISO 5049 is suitable for designing continuous conveyors, however not for steel buildings, e.g. transfer towers.
7	Should the client requires compliance with AS4324.1, by what percentage increase do you expect the structural steel mass and overall cost.	Typically we expect an increase in the range of 10 to 20 %.
8	To what extent are basic hand calculations done?	Local analyses and calculations are performed in Excel or by hand.
9	How heavily do you rely on finite element analysis? Please comment on the use of bar elements vs shell and brick elements.	For bar (beam) structural analysis we use Krasta and R-Stab (RISA, STAAD) For plated structures we use Ansys with shell and brick elements.
10	Would you prefer to design machines more conservative if economical constraints were less severe or do you feel that machines are designed robustly enough to withstand accidental loads encountered during it's life cycle?	Extraordinary loads are calculated in a way ensuring 20 % safety relative to the yield limit of the material. This avoids permanent deformation. Our approach is "elastic" without "plastic" reserve load capacity.
11	To what extent are previous designs modified for new machine designs?	Practice-approved design solutions are re-used as needed.
12	How concerned are you about operator abuse and, are you satisfied that your equipment is suitably equipped with protection systems?	Machine operators are given special training. Operator manuals are prepared for the machines. The machines are provided with various safety and monitoring devices (var. limit circuits, overload protection system, touch guards). However, willful abuse cannot be ultimately prevented.
13	Do you ever do a design risk assesment with the client before commencing with designs?	According to the contract conditions, assumed loads and design criteria are to be submitted to the customer for approval.
14	How do you integrate the design of the control and machine protection systems with the structural design?	Control and protection devices are included in the structural design. Specific load cases are only taken into account if so required by the applicable standard code, i.e. limit switches (travel) have to work to prevent contact at end stops.
15	To what extent is the structural design engineer involved with the commissioning and final release of the machine?	The machine will be commissioned by electrical engineers, mechanical engineers and the supervisor. A structural engineer will be involved in exceptional cases only.

Company:	**ThyssenKrupp Materials Handling**
Information provided by:	Carel van der Merwe and Michael Herzog
Designation:	Engineering Manager and Senior Engineer
Design experience (years):	11 and 12
Number of bulk handling machines desiged:	Since 2001: 14 different types, 25 machines

The following questions are specifically related to the design of mobile equipment for continious handling of bulk materials

	Question:	**Response:**
1	To what extent are the following standards and specifications used if no requirement from the client is provided:	
	a) ISO 5049	The ISO 5049-1 standard forms the basis of design of mobile equipment. This standard is closely related to FEM Section II which is thus suitable to be used in parallel.
	b) FEM 2.131 / 2.132	FEM Section II is used in parallel with ISO 5049, in particular to supplement it where certain aspects are not covered by ISO 5049.
	c) AS 4324.1	Only used when specifically required by the client, because this standard generally results in significantly more expensive machines.
2	What other standards and specifications are utilised?	SANS 10162 is often used for the design of secondary structures on mobile equipment like walkways and access. Numerous other standards are used to check the validity of the structural design or to address more specific detailed problems. For example, AWS standards may be used to address welding issues, IIW for the fatigue assessment of non-standard geometry welded structures, various DIN specs for specific mechanical components.
3	Is the structural design based on allowable stress or a limit state approach?	Generally, for mobile equipment, allowable stress design is used according to the ISO/FEM standards.
4	Do you consider it general practice to apply a limit state approach based on loading obtained from ISO 5049-1?	No. Limit states design is a very integrated statistical approach with regards to factored loads, their combinations and member resistances. It is not good practice to mix fundamentally different design concepts if this can be avoided, and hence we prefer to apply allowable stress design when loading is defined according to ISO 5049. Furthermore the load combinations from ISO can be used directly when checking safety against overturning etc., while using factored loads from a limit states structural analysis would not be sensible.
5	Do you consider the requirements from ISO 5049-1 to be adequate to deliver safe structural designs?	Yes, when supplemented for specific detail issues by other standards. This standard has been used successfully. Failures can normally be traced back to factors not directly related to the standard, such as machine abuse, faulty design assumptions or application beyond design basis.
6	How do you go about adressing items where ISO 5049-1 is perhaps not suitable or comprehensive enough?	We consult other standards, for example FEM Section II for checking the local stability of plates, and other standards which may provide a more comprehensive solution to a particular design question.
7	Should the client requires compliance with AS4324.1, what percentage increase do you expect on the structural steel mass and overall cost.	20 to 30% in both cases, as increased structural mass also impacts of the mechanical equipment, drives and electrical equipment. The added complication is that most of the current proven designs were done to ISO5049 and FEM II and hence could no longer be used as the basis of offers to clients, resulting in increased lead times for the supply of machines to AS4324.1

Continued on next page

…ThyssenKrupp response continued:

7	Should the client requires compliance with AS4324.1, what percentage increase do you expect on the structural steel mass and overall cost.	20 to 30% in both cases, as increased structural mass also impacts of the mechanical equipment, drives and electrical equipment. The added complication is that most of the current proven designs were done to ISO5049 and FEM II and hence could no longer be used as the basis of offers to clients, resulting in increased lead times for the supply of machines to AS4324.1
8	To what extent are basic hand calculations done?	Standard structural design programs like Prokon and Masterseries cannot be applied to the usually unique geometry of machines, resulting standard FEA combined with hand calculations (either manually or using calculation programs like Mathcad) being the most appropriate approach. Hand calculations therefore form an important element in the design of machines. Bolted connections, for example, can often be most effectively verified in this way.
9	How heavily do you rely on finite element analysis? Please comment on the use of bar elements vs shell and brick elements.	FEA is an important design tool for mobile equipment, and is used extensively, but always in combination with hand calculations according to codes/standards. It is particularly useful to verify and optimize the structural design of plate structures of non-standard geometry, short structural elements etc., as well as the more detailed assessment of buckling and fatigue on non-standard geometries. The compactness and shape of the structure, as well the purpose of the analysis determines which type of element is most suitable for the problem. A typical machine is modeled using a combination of beam and shell elements, with solid (brick) elements only being used occasionally in particular areas.
10	Would you prefer to design machines more conservative if economical constraints were less severe or do you feel that machines are designed robustly enough to withstand accidental loads encountered during it's life cycle?	Yes. A more conservative design also reduces the risk for the machine supplier, is easier to design, and potentially allows room for future upgrades in capacity etc. and from this perspective would be preferable. From an engineering perspective it is desirable to offer the client the best possible long term solution with maximum reserve capacity, functionality and maintainability. Commercial pressure however makes this impossible, the most cost effective solution for the stated requirements are necessary to win business. This puts suppliers that propose more robust and higher quality equipment, or with design features that make them easier to maintain at a disadvantage. This often forces a supplier to design the cheapest machine that meets the stated requirements of the client at the time of the enquiry, which is not necessarily the best long term solution. Machines that are properly designed to ISO 5049, and maintained and operated correctly are robust enough for most accidental loads. Bucket wheel reclaimers are however a higher risk (structure and overturning) than most other types of machines, and it is appropriate to specify additional safety measures for their design. When very dense materials are handled, it would be appropriate to increase the encrustation design loads as the encrustation on a structure is related more to the volume of spillage that can accumulate on a structure and that material's density, than to the material load on the conveyor. On such machines regular cleaning of spilled material is important.

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…ThyssenKrupp response continued:

11	To what extent are previous designs modified for new machine designs?	A family of machines will tend to share a broadly similar structural concept, but changes of a significant nature make full validation of structural integrity necessary. If the client has particular geometric requirements or equipment preference, this normally results in a greater deviation from previously built equipment and a more unique design. Loads and site conditions also tend to vary and a validation of an existing design may become necessary even when the basic design is the same.
12	How concerned are you about operator abuse and, are you satisfied that your equipment is suitably equipped with protection systems?	This is a real concern, and more so with bucket wheel reclaimers. The equipment is normally equipped with sufficient protection systems, but these systems must be maintained in full working order to be effective. Operator abuse, neglect of maintenance, structural modifications and the intentional disabling of safety devices are all a widespread problem in industry. It is impossible to cater for the whole spectrum of extraordinary loads which may result from irresponsible and sometimes reckless practices machines can be exposed to.
13	Do you ever do a design risk assesment with the client before commencing with designs?	Risk assessments of this nature have been done in the past. The effectiveness of such an exercise is typically compromised because the contractual programme generally does not allow for the start of the detailed design to wait for this time consuming procedure to be completed. The risks involved in the operation of materials handling equipment are not unknown and machines are equipped to minimize these risks. Further risk reduction by enforcing suitable procedures on site is the responsibility of the client, particularly to operate the equipment within its specifications and keep the equipment in good condition. An important benefit from a risk assessment is that it makes the client more aware of particular risks and his part in mitigating this.
14	How do you integrate the design of the control and machine protection systems with the structural design?	At conceptual design stage the load magnitudes and combinations are normally captured in a basis of design document, which is aligned with the functional description of the control system. This is done as the safety systems that are provided impact on the loads to be considered. As both the structural and EC&I system design is done in-house this is readily achieved. Furthermore our EC&I team is specialized in materials handling equipment and therefore familiar with the typical design concept of control and protection systems for a particular type of equipment. Inspection, maintenance and testing requirements for safety systems are covered in the machine specific operating and maintenance manuals that are delivered with the machine.
15	To what extent is the structural design engineer involved with the commissioning and final release of the machine?	The structural design engineer carries out periodic inspections at fabrication workshops and during the erection of the machine. He also inspects the machine on completion of construction to certify that it is structurally safe to commission, and carries out an inspection during hot commissioning.

Company: **Undisclosed**
Designation: Undisclosed
Design experience (years): 18
Number of bulk handling machines desiged: 60

The following questions are specifically related to the design of mobile equipment for continious handling of bulk materials

	Question:	**Response:**
1	To what extent are the following standards and specifications used if no requirement from the client is provided:	
	a) ISO 5049	
	b) FEM 2.131 / 2.132	FEM 2.131 / 2.132
	c) AS 4324.1	
2	What other standards and specifications are utilised?	DIN 18800 (edition 11.1990)
3	Is the structural design based on allowable stress or a limit state approach?	Based on limit states.
4	Do you consider it general practice to apply a limit state approach based on loading obtained from ISO 5049-1?	Yes
5	Do you consider the requirements from ISO 5049-1 to be adequate to deliver safe structural designs?	Yes
6	How do you go about adressing items where ISO 5049-1 is perhaps not suitable or comprehensive enough?	From experience it is comprehensive enough.
7	Should the client requires compliance with AS4324.1, by what percentage increase do you expect the structural steel mass and overall cost.	Possibly an increase of 10 % on the structural steel mass.
8	To what extent are basic hand calculations done?	Not much - less than 10 % of the total design.
9	How heavily do you rely on finite element analysis? Please comment on the use of bar elements vs shell and brick elements.	Results of FEM analysis have to be checked by simple hand calculations. FEM analysis should only used for complicated detail design with fatigue considerations.
10	Would you prefer to design machines more conservative if economical constraints were less severe or do you feel that machines are designed robustly enough to withstand accidental loads encountered during it's life cycle?	Machine design is robust enough, from experience problems with machines result from incorrect operation, this should be clarified by a design risk assessment.
11	To what extent are previous designs modified for new machine designs?	New machines are redesigned completely to reduce possible errors.
12	How concerned are you about operator abuse and, are you satisfied that your equipment is suitably equipped with protection systems?	Yes
13	Do you ever do a design risk assesment with the client before commencing with designs?	Yes, if the client requires it. However often the client wants a low-priced machine.
14	How do you integrate the design of the control and machine protection systems with the structural design?	There are adequate load cases and loadings defined.
15	To what extent is the structural design engineer involved with the commissioning and final release of the machine?	Not in an executive position, deviations to the planned and completed design at site are communicated and the effects therof will be considered.