2

Cases of Machinery Failure

Machine: 1549, "structure of any kind," from Middle French *machine* "device, contrivance," from Latin *machina* "machine, engine, fabric, frame, device, trick" (cf. Spanish, *maquina*, Italian, *macchina*), from Greek, *makhana*, Doric variant of *mekhane* "device, means," related to *mekhos* "means, expedient, contrivance," from Proto Indo European *maghana-* "that which enables," from base *magh-*" to be able," have power. Main modern sense of "device made of moving parts for applying mechanical power" (1673) probably grew out of 17th century senses of "apparatus, appliance" (1650) and "military siege-tower" (1656). *Machinery* (1687) was originally theatrical, "devices for creating stage effects"; meaning "machines collectively" is attested from 1731. The verb is from 1915. Machine for living (in) "house" translates Le Corbusier's *machine à habiter* (1923).

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2.1 The Cases

In this chapter case material is presented about disputes resulting from the failure or damage sustained by machinery of various types. In each case a background is sketched for the dispute to provide the reader with a *raison d'étre* for the origin of the dispute. There are eight cases presented in this category, each involving the failure of a key component of machinery. In all but one case the failure of these key machinery components was identified as the defining cause of the dispute. In one special case, that dealing with a paper coater damaged in transit, the defining event of the dispute was the manufacturer's claim that the cost of repairs to the damaged paper coater would be substantially greater than the purchase of the original machine.

These eight cases are, in order of presentation:

Heavy Theatre Lights are Dropped From a Great Height – in this sample case the failure of a key component of the winching system, used for raising and lowering theatre lights, caused an accident. The plaintiff (theatre administrators) alleged that winch failure was caused by the supply of faulty components by the suppliers of the winch motors and gearboxes. Investigation of the failure identified faulty winch drive specifications as the prime cause of the accident. The underlying cause was traced back to a cost-cutting decision by local government and poor advice from the original engineers responsible for the mechanical services in the theatre.

The Main Bearing Breaks on a Tunnelling Machine – In this case a complex piece of machinery, a tunnel boring machine (TBM) was purchased in a tendering process by a local government agency for an underground rail loop. The machine specified to work in hard rock tunnelling for approximately 3000 hours. After almost 900 hours of operation the main cutting head of this machine weighing about 30 ton broke off. The assigned underlying cause of this accident was a failed main bearing that supported the head of the machine. Investigation into the mechanics of the failure found that the real cause of the accident was a design weakness in the main bearing support system. Further investigation of the history of the purchase of the tunnelling machine showed that the government agency responsible for the purchase of the tunneller made a serious error of judgement in accepting the cheaper tender for the machine ahead of a tender from a slightly more expensive but substantially more experienced hard-rock tunnelling machine manufacturer.

Brinelling Induces Unacceptable Vibrations in a Very Large Bottle Filler – A beverage manufacturer discovers a faulty filling machine and seeks to investigate the possibility of similar failures in other similar machines in their several large plants. Initial evaluation assigns the cause of the failure to a faulty bearing. Deeper investigation identifies the defining failure as being due to an installation error.

A Milk Tanker Takes a Spill – As suggested by the title of this case, a milk tanker overturns and spills its load while negotiating a bend in a country road. Initial examination of the trailer hitch shows that the kingpin hinge in the trailer hitch is of a non-standard design. Loss assesors assign the cause of the accident to this non-standard kingpin. Subsequent investigation of the accident shows that the underlying cause was poor judgement by the driver when negotiating the turn. Moreover, the kingpin bolt's failure in these conditions saved the prime mover from sustaining substantial damage.

A Paper Coating Machine is Damaged in Transit – The manufacturer clams that repairs to the damaged machine must be performed in America. The ultimate costs claimed against transport insurance are substantially greater than the cost of a completely new machine. Insurance loss assessors seek to investigate alternative means of repair to the machine. Investigation shows that local repairs are appropriate.

A High-speed Compressor is Damaged by Faulty Bearing Replacement – This is a case of poor judgement exhibited by maintenance engineers in attempting to repair specialised bearings on a high speed air compressor. Investigation into this case suggests that although outsourcing maintenance may save on overheads for a manufacturer, the cost of litigation resulting from incompetent maintenance can exceed these savings.

Two Large Vehicles Roll Over – Both of these cases are the result of weaknesses in design considerations. In the first of the two a weakness in re-designed track rod clamp causes a large fertiliser spreader to roll over. In the second case a weakness in the design of a steering knuckle on a tipper truck is exacerbated by its use in a specially hazardous environment.

A Large Paper Machine Dryer is Damaged and Discarded Prematurely – Paper-making industries are very conservative in both maintaining machinery in top-notch conditions and in the operational safety of their machines. The defining event for this case was an unexpected accident in which a large drying drum was damaged. On advice from non-destructive testing experts and based on the conservative estimate of the damage the drying drum was replaced under a machinery insurance claim. Subsequent insurance assessors questioned the need for replacing the drying drum, when repair options may have been available. The ensuing investigation suggested that the paper machine operator used the defining event as a means of an opportune replacement of the dryer with one of significantly improved performance.

Much of the information for case material is drawn from the author's experience in litigation consulting as an expert witness. Cases are presented as fully as possible without disclosing the real names of participants. To avoid the painfully obvious ploy of using initials to replace proper names, in each case various groups of names have been invented, drawing on the world of music, sport, theatre and horticulture. No apologies are offered for this approach, since it makes the cases a little more readable.

2.2 Heavy Theatre Lights are Dropped From a Great Height

2.2.1 The Case Culture

Flying battens are substantial steel girders that hold a host of specialised stage equipment used in the sound and light management of theatre performances. The *Erewhon City Concert Hall* (*ECH*) had several such flying battens, weighing in excess of a 100 kg each, mounted high above its stage. These flying battens are raised or lowered during performances by means of a motorised cable and winch system operated from a central control panel. Figure 2.1 is a general view of the concert hall stage. In this image the flying battens are used to carry sound reflectors to improve the acoustics of the auditorium. Figure 2.2 is a simplified schematic partial view of the flying batten and winch system.

The *ECH* was an ambitious project by the Erewhon City governors, and the original specifications for the flying battens called for hydraulic winch motors. However, as often happens with major civic projects experiencing serious cost overruns in an election year, the hydraulic motors were replaced by cheaper geared motors. Although this may seem a trivial change in the cost of such a large project, there were in fact 90 such motors involved in the whole art centre project, of which the concert hall was only one element. In total the initial cost difference between hydraulic and geared motor winches was of the order of US$ 1 million.

2.2.2 Defining Event

Sometime after the opening of the concert hall during a rehearsal one of the flying battens fell to the stage from a height of about 20 metres. Fortunately no one was injured, but an investigation of this accident was demanded by the insurers of the whole art centre complex, as well as occupational health and safety authorities. A brief technical evaluation of the failed winch system by an independent consultant, called here *Goalie Material Testing* (*GMT*), revealed the following:

1. The winch motors and gearing were specified by the technical staff of the art centre complex architects *Forward* Ltd.
2. All winching equipment was imported from an American supplier, let us call them the *Ruckman* Corporation.
3. Original specifications of the flying battens called for a maximum loading of 200 kg and the system has been tested satisfactorily under static loading to nearly 500 kg.
4. Hardness tests of the shafts of the gearing system showed both input and output shafts to be 30% below specified values in material tensile strength.

Figure 2.1 A general view of the concert hall stage

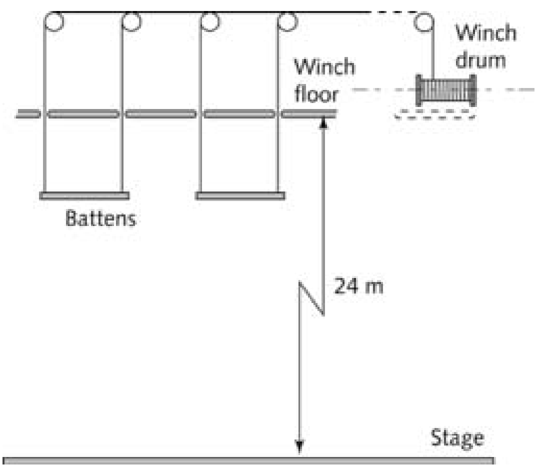


Figure 2.2 Schematic partial view of the flying batten and winch system

5. American ASME standards for shaft design. This evaluation showed the shafts to be undersized by approximately 8%.
6. The failed winch shaft was found to have failed in fatigue and machining roughness on this shaft was identified as the originating cause of the eventual failure.

Based on the above findings writs were issued by the art centre insurers against thirteen defendants, including the original architects of the centre, the technical staff and specifiers and suppliers of the winch system, *Forward* and *Ruckman*.

2.2.3 Parties to the Dispute

- *ECH insurers* – the plaintiff in this case took on the role of injured party on behalf of ECH administrators.

- *Forward Ltd.*, the original architects of the concert hall – the major defendant in the case had the responsibility for the design and specification of the failed winching system.

- *Ruckman Ltd.*, subsidiary defendant, supplier of the failed winch equipment – they were enjoined in the case through the main defendant *Forward*.

2.2.4 The Client

- *Centre and Pocket Ltd.*, Lawyers acting for ECH on the advice of GMT – they briefed me on the background to this case and requested expert engineering opinion about the identified faults in the winch drive system.

2.2.5 The Expert's Role and Associated Investigation

In general, the expert is required to respond to specific questions relating to the technical substance of a dispute. In this case the technical substance was evident, the winch had failed to perform as expected and there were continuing safety issues with all the winches – although no one had been injured in the first accident, there was very high risk of injury should other flying battens be dropped unexpectedly.

My personal involvement in this case commenced after the GMT evaluation as well as after the issue of the various writs against the thirteen defendants. Figure 2.3 shows a winch motor attached to a worm gearbox, from which the winch drums are driven. The smaller diameter component attached to the motor is a separate armature brake. This device is intended to stop the motor from over-revving or freewheeling in case of a power failure. Figure 2.4 is a view of the winch floor and Figure 2.5 is a typical winch motor shaft with the worm screw attached to it. At the time of my involvement in this matter, virtually no expense had been spared to investigate the mechanical issues identified by Goalie Ltd. My brief was to evaluate the whole system of winches and winching systems in addition to that found by Goalie Ltd. I suspected that my involvement was supposed to heavily reinforce the Art Centre insurer's case against the thirteen defendants and specifically against Ruckman, the American suppliers of the winching motors and gearboxes.

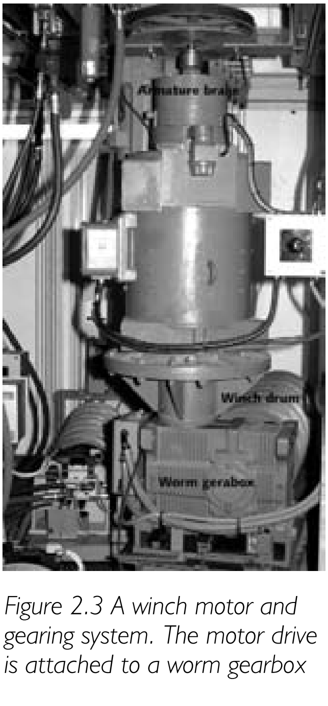


Figure 2.3 A winch motor and gearing system. The motor drive is attached to a worm gearbox

Figure 2.4 Winch floor showing the main winch drums and part of the pulley system

Figure 2.5 Typical motor shaft and worm screw

As requested by my brief, I inspected the winch systems, the failed components and the various technical reports by GMT (there were several) as well as the writ issued against the thirteen defendants. In general, where human injury or life is placed at risk, design specifications should call for a fail-safe system rather than a safe-life system. The fail-safe design, in the case of the flying batten winches, would require the system to be safe even when some critical part of it, such as a motor shaft, should fail. With a worm drive at the output end of the winch motor the designers may have assumed that the worm gearbox was not capable of being "back driven". In other words, should the motor shaft fail, the torque on the winch drums would not be sufficient to drive the worm gear/worm screw combination. The evidence of the fallen flying batten showed this assumption to be false. In any case there was no evidence that any tests or calculations were conducted to support the above assumption.

In elevator design the cabin of the elevator is brought to a halt should the cable system fail. This is a fail-safe design where elaborate braking systems are included in the design of the elevator cabin guidance system. In the art centre winching system the design did call for a disc brake system to be installed on the motor armature. The armature brake supplied by Ruckman was, in fact, a drum brake (see Figure 2.3). However, none of the design specifications or the eventual supply and documentation of acceptance of the winch system had any evidence of fail-safe design considerations.

2.2.6 Lessons Learnt

Original design specifications for the winch system at ECH called for hydraulic motors. Perhaps intuitively, or from the wisdom of experience, the architect's engineering staff considered the fail-safe behaviour required of the winch system. Hydraulic motor drives would have provided that feature of the design. Unfortunately this key issue was not explicit in the documentation. As a consequence, there was no clearly identifiable reason why a cost-cutting review of the specifications should not change from an expensive hydraulic drive system to a cheaper geared electric motor drive. That too would have sufficed had the specification stipulated that braking should be installed on the winch side of the drive.

2.2.7 Outcome

Unfortunately for the art centre insurers, my report on this matter turned out to be unacceptably damaging to their case. The mitigating element of the case was that the art centre authority could not fully deny responsibility in accepting the design specifications when it clearly neglected the issue of fail-safe provisions. The matter was eventually settled out of court.

2.3 The Main Bearing Breaks on a Tunnelling Machine

2.3.1 The Case Culture

"The modern era of machine tunnelling was born in the early 1950s. The designers of machines and the contractors who used them thought they were developing completely new techniques and new equipment. However, a view of machine tunnelling history from a clearer perspective shows that the tunnellers of the 1950s were redeveloping methods that had their origins in methods that existed more than 100 years earlier.

The publication of the scholarly and well researched Handbook of Mining and Tunnelling Machinery,[2.1] *brought the background of mechanized tunnelling into focus for the first time. Had the designers of 35 years ago been able to read the history of developments that had preceded them, their designs undoubtedly would have been affected materially."*[2.2]

In modern urban environments, tunnelling is probably the most effective way of constructing underground passage-ways for sewers, underground rail services or for cable ways. The scale of underground works has been increasing, ever since tunnelling machines, colloquially called *moles*, have been built. In Chicago the Tunnel and Reservoir Project (TARP) consists of a series of tunnels and reservoirs, some dug as deep as 360 feet, constructed parallel to Chicago's river systems. The system, when complete will extend more than 130 miles. TARP is being constructed using a tunnel boring machine that cuts a hole 33.5 feet in diameter through bedrock deep beneath Chicago's surface.

In Norway's Lillehammer a giant underground dome was built for the olympic hall of the XIIth Winter Games in 1994 using tunnelling machinery. This is the largest ever underground cavern with a net area of over 10,000 m^2 capable of seating 5100. Figure 2.6 indicates the scale of tunnelling machinery used in major earth works.

2.3.2 The Defining Event

Some years ago the city of Mytown's rail transport authority (*MRTA*) decided to extend the suburban rail system in their city to include a loop around its busy central business district. Due to the existing urban development, an underground loop was the only feasible solution. After a tendering process *Caster-Pollux Pty. Ltd.* (*CP*) won the contract for the construction of four single-track tunnels, on two levels, feeding into the city's other surface suburban train lines. The tunnelling plan called for 10 km of tunnels and the mechanical removal of 900,000 m^3 of rock and earth. *CLF* was chosen as the contractor largely because they had an established record of experience with such major earth works. Unfortunately, CP was also an

2.1 Stack (1982)

2.2 Robbins (1987)

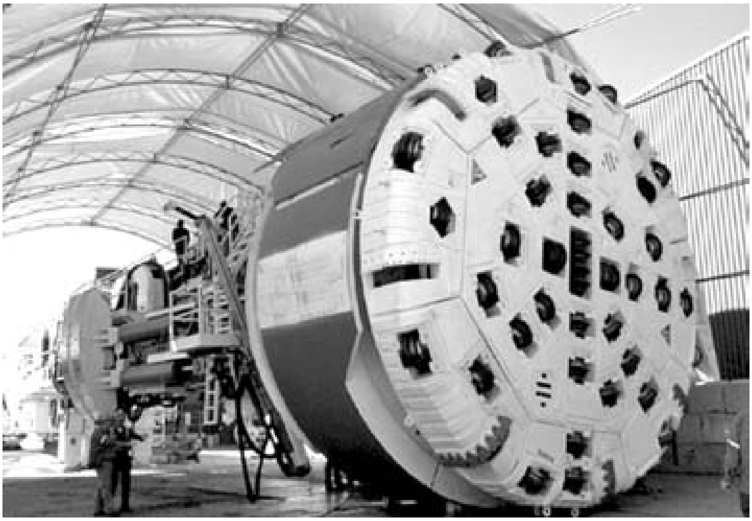


Figure 2.6 Full-face tunnelling machine just before entering tunnel portal

independent entity formed from a "quango",[2.3] somewhat constrained in their contracting by long-standing, conservative, government-established procedures. Subsequent events suggested that they chose the tunnelling machine based on economic considerations, rather than internationally proven reliability.

Based on underground surveys, it was estimated that the tunnel would take approximately 3000 hours to dig and having specified the rock characteristics (based on the said surveys) tenders for a tunnelling machine were called. The favoured tenderer was *Rawaj Pty. Ltd.* providing a tunnel boring machine (TBM) for approximately AU$ 2 million, or about 10% of the estimated total contract cost. Several tenderers, including Rawaj, had international reputation and expertise in tunnelling. However, there is always considerable uncertainty about the nature of the rock composition through which tunnels are dug. TBMs are usually purpose-built for a specific contract. Once the contract is concluded the machine is retired or rebuilt for a further application. In general, the salvage value of a TBM is insignificant in terms total contract cost of digging and constructing a tunnel. Some TBM manufacturers design machinery to meet the specified rock characteristics, while others choose to design machines that are so robust that they can withstand considerable variability in rock strength and distribution. The former approach yields lighter and less costly TBMs, while the latter approach results in a more expensive but commensurately more robust design.

After about 900 hours of operation the main bearing of the machine broke and the whole cutting head of the machine, weighing about 30 ton,

2.3 An organization or agency that is financed by a government but that acts independently of it.

fell off.[2.4] Fortunately for all concerned, the accident event took place near one of the main ventilator shafts and the machine head was recovered with minimal need for reversion to blasting and old-fashioned mining procedures. The cost of repairs was estimated at approximately AU$ 500,000. As well, there were substantial delays in construction and correspondingly substantial costs incurred in *liquidated damages.*[2.5] As a quango, CP would need to rely on government financing to bail them out, should the insurance loss be influenced by issues arising from defective contract planning or project management. It would have been inconceivable that CP would actually pay any liquidated damages to the government (CP was being financed by the government) and it was the travelling public that would have had to bear the discomfort of the resulting transport delays. Hence, facing an election year, the government of the time expressed the need for urgency in sorting out the root causes of the disaster, and getting the contract back on the rails.

2.3.3 Parties to the Dispute

- *CP* and their insurers – they saw themselves as plaintiffs in this case against Rawaj, the TBM supplier.
- *Rawaj* and their insurers, the defendants.
- The city of *Mytown* and specifically the transport authority *MRTA*, to whom CP was the main contractor for the rail loop project.

2.3.4 The Client

The government asked for an expert evaluation of the causes of this disaster. Due to the specialised nature of tunnelling there are very few experts one is able to call on with confidence. In fact, there are very few experts in the large-scale tunnelling business who are able to offer advice that is seen by a court to be free of conflict of interest. This is due to the fact that most of the available expert specialists work for tunnelling machinery or contracting companies. As noted earlier, experts are in the "credibility business" from a legal point of view. Conflicts of interest are easily discovered and brought to the notice of the judiciary, should the dispute proceed to litigation. In this case a highly experienced local expert was available, who could be relied upon to give unbiassed advice on the event. Unfortunately this expert, *David*, had worked consistently as advisor to both *CP* and for a large and internationally reputable tunnelling machinery supplier *Swallows Pty. Ltd.*, who just happened to be the "losing bidder" for the supply of a tunnelling machine for this contract.

Initial assessment of the causes of the damage to the tunnelling machine was carried out in house by *CP*, advised by their consultant *David*, who realised that there was a possible conflict of interest and asked for an inde-

2.4 Ton is a US unit of mass = 907.2 kg, or just slightly less than the SI unit of tonne.

2.5 "Liquidated damages" provides for the payment of a certain fixed amount in the event of a breach of a contract, including time delays.

pendent consultant to be appointed. As a design specialist I was appointed by CP to review the substance of the case, with the understanding that David would provide some "coaching" in the specialities relating to tunnelling machine design.

2.3.5 The Expert's Role and the Investigation

At first sight the damage could be attributed to one or more of several causes, acting singly or in combination, namely:

- Unexpected variations in the rock through which the tunnel was being dug.
- Faulty bearing material.
- Faulty adjustment or installation of the bearing.
- Inappropriate or faulty operational procedures.

  There were elements of the operation that could be called into question also. The cutting blades on the face of the machine required regular inspection and maintenance. These large rolling cutters had their own set of bearings to permit rotation as the cutting face of the machine was rotated. Should any of the blades be stopped for some reason (such as their own bearing failure) they would be soon worn down and the result would be a highly uneven load distribution on the cutting head of the machine. This uneven load would then transmit to the main bearing itself. So it would seem that there were issues of maintenance and inspection to contend with. These issues included some element of cost saving on the part of the contractor, since the cutters and their associated bearings were a very costly "consumable" item in the contract. In a tunnelling context TBMs are occasionally referred to as the "Box Brownie" part of the contract, making reference to the Kodak approach of selling the camera cheaply and recouping costs by the profits made on the sale of consumables – the film and developing.
- A design fault in the TBM.
- Some totally unexpected, unprepared-for cause (usually referred to in insurance terms as an "act of God").

The last two possible causes were originally seen as unlikely to be helpful to the dispute since design faults are very difficult to establish and there were no clearly identifiable indicators of an unexpected act of God. As a part of these early deliberations, the bearing manufacturer was called upon to answer for issues relating to the material and the mounting instructions for the bearing. It is useful to consider the possible failure scenarios due to the several causes listed above.

- *Unexpected variations in the rock through which the tunnel was being dug* – This could result in incorrect bearing specification, as the estimated loads on the bearing may have been less than conservative for this highly probabilistic load application. There were rock mechanics studies of the site prior to calling for tenders and the contractors were fully acquainted with the range of likely rock properties in the proposed tunnel. In spite of this, tunnelling is almost always fraught with considerable uncertainties in the distribution of rock properties. Moreover, the design of the rail loop tunnel called for steering by varying the thrust across the face of the cutting head. Quite apart from minor directional modifications there were four 90° "corners" to be negotiated by the machine as it completed the loop. Investigation of this aspect of the operation required the estimation of loads on the bearing and a calculation of its L_{10} life.[2.6]

• *Faulty bearing material* – bearings used on tunnelling machines are specially constructed bearings that require special attention to fitting, pre-loading, sealing and lubrication. The vibrating loads due to the rock cutting process introduces some special design requirements to cope with fatigue. Unusually harsh operating conditions in the tunnel can impose special material, sealing and lubricating requirements on the design. A maintenance report of the bearing appearance at failure was available for inspection. In addition the metallurgy of the bearing had to be checked against the manufacturer's specification.

• *Faulty adjustment or installation of the bearing* – recommendations for installation of these special bearings are normally provided by the manufacturer. Often there are sample mounting configurations provided from existing machinery. Designers may vary from these recommendations, but in general the fit (tolerances on the mounting inner and outer diameters) and the maximum misalignment (angular deviation from the vertical plane) need to be addressed in the design. Variations from these design requirements can result in early failure of the bearing. Investigation of this aspect of the accident required design calculations on the mounting arrangement of the bearing.

• *Inappropriate or faulty operational procedure* – an operator's log was available for each shift (eight hours) and these needed careful examination to determine if anything in the operation could have predicted the early failure of the bearing.

Figure 2.7 is a schematic view of a tunnelling machine operating in a tunnel. Figure 2.8 shows the operating stroke of the TBM schematically. The machine head is placed against the rock face by "walking" the machine for-

2.6 L_{10} life is a probabilistic measure of bearing life. It is the time after which (all other things being equal) 90% of these bearings are still operating successfully (are alive). For smaller bearings the L_{10} life is found by direct laboratory testing. However, for the size of bearings used in tunnellling machines, it would be inconceivable to test sufficient numbers to get a direct measure of the L_{10} life. Consequently this value must be estimated from load figures and formulae provided by the manufacturer.

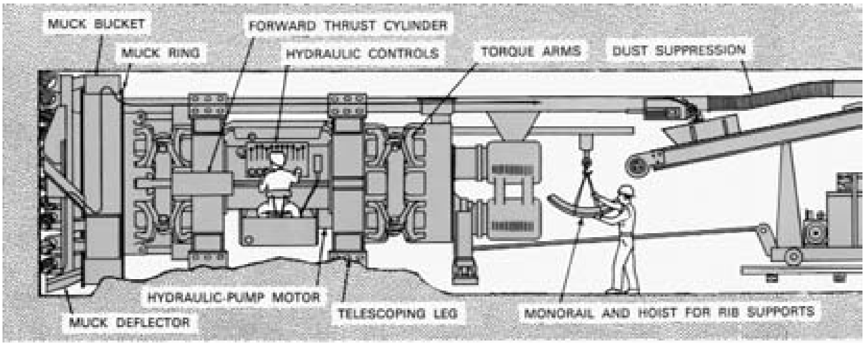


Figure 2.7 Schematic view of a TBM operating in a tunnel

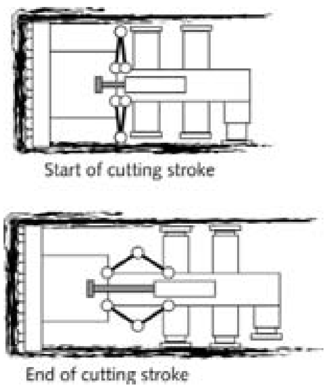


Figure 2.8 Schematic view of a TBM operating stroke

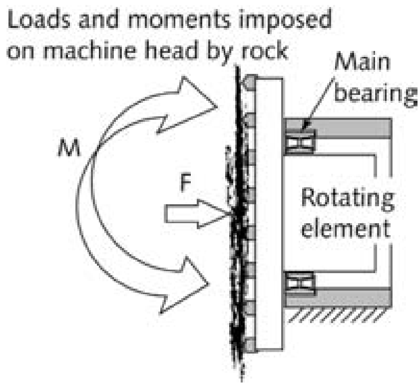


Figure 2.9 Schematic view of the main bearing indicating terminology

Figure 2.10 Broken main bearing of the MRTA tunnelling machine

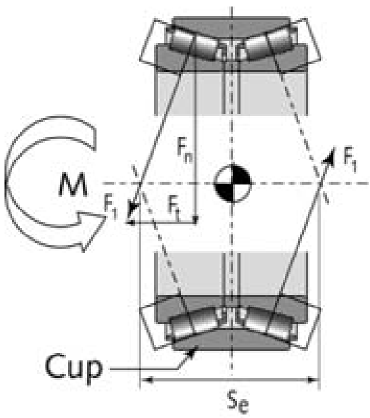


Figure 2.11 Schematic view of the main bearing indicating terminology

ward on its hydraulic supports. In this condition the main hydraulic rams of the machine are fully contracted. The hydraulic supports of the machine are then extended to grip the walls of the tunnel. Following this step, the cutter head is rotated at approximately 5 r.p.m. while the main hydraulic rams push the head forward into the rock face. Figure 2.9 is schematic view of the head of the tunnelling machine, indicating the nature of loads imposed on the head by the rock face. These loads are a combination of a thrust force, F, and a moment, M, the former varying in magnitude and the latter varying in both magnitude and direction as a result of variations of rock strength in the tunnel. Figure 2.10 is a photograph of the failed bearing with a 1.7 m tall person to provide an idea of the scale of the failed component.

Figure 2.11 is a schematic view of the TBM main bearing, a double row taper roller bearing manufactured by the Torrington Corporation, estimated to cost $20,000 in 1975, at the time of the accident.

In single row taper roller bearing terminology the outer race of the bearing is known as the *cup* and the inner race including the roller assembly as the *cone*. These terms are assigned to the bearing due to their overall physical shapes. In the *MRTA* failure it was the front section of the outer race (part of the cup) of the main bearing that broke away to permit the cutting head to fall out of its support. Figure 2.11 also indicates the types of loads acting on the bearing. Quite apart from the direct thrust load imposed on the bearing, resulting in the direct forces F_1 being carried on the outer race (cup), the moment M on the machine head induces extra varying loads on these bearing cups. The weight of the machine head, outside the central plane of the bearing results in an added moment being carried by forces on the bearing cup.

Figure 2.12 Bearing roller and part of the failed bearing cup. The notch was used for establishing material properties

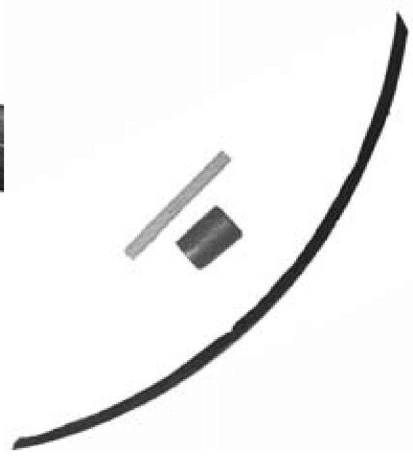


Figure 2.13 Part of the bearing broken away during the accident, with a roller and a 300 mm ruler to indicate scale

A small sample of bearing material was taken from the broken section of the bearing and examined by a metallurgy laboratory. It was found to match the maker's specification for this bearing. Having reviewed the operator's log and the bearing material it was now necessary to estimate bearing life and operating loads.

The tunnelling machine had the following specifications:

- Total mass = 250 ton
- Cutting diameter = 23 ft 2 inch (7.08 m)
- Cutter head mass = 30 ton
- Bearing housing = 15 ton
- Ring erector = 6 ton
- Main bearing diameter = 85 inch (3.35 m)
- Normal bearing thrust[2.7] = 350 ton (7.6×10^5 lb force)
- Bearing type Torrington S-34887-C double row tapered roller bearing.

Figure 2.12 is a close-up of a bearing roller, together with part of the bearing cup that broke away during the failure of the bearing. Figure 2.13 shows the complete broken part of the bearing cup representing approximately 120° of arc or about the arc length corresponding to 1/3rd (16) of the total number of bearing rollers (this bearing had 48 rollers in each row).

Of course, the housing itself is part of the design of the machine into which this type of bearing is fitted. In general, these types of bearings are used in applications requiring robust and reliable behaviour with substantial bearing stiffness, which is associated with the capacity of the bearing to withstand overturning moments. In Figure 2.11 the following nomenclature has been used:

- S_e is the "effective spread" provided by the bearing geometry, or the distance at the centre line of the machine, which is the moment arm for resisting overturning moments. For the application under consideration the value of S_e was 48.4 inch (1229 mm);
- F_1 is the reaction load at the bearing due to the overturning moment M acting on the bearing;
- F_n and F_t respectively are the normal and axial components of F_1. The axial components cancel out and the resisting moment of the bearing becomes $F_n \times S_e = M$.

The relevant overturning moments resulting from the statistical estimates of the operating conditions for the machine were taken as:

5% of the time $M = 7 \times 10^7$ inch-lbs (8×10^6 Nm)

30% of the time $M = 4 \times 10^7$ inch-lbs (4.5×10^6 Nm)

2.7 This is a nominal value only. The actual thrust for calculating bearing loads is based on a statistical assessment of the median (most commonly occurring) thrust at various distances from the machine centre, expressed as a moment acting on the machine cutting head.

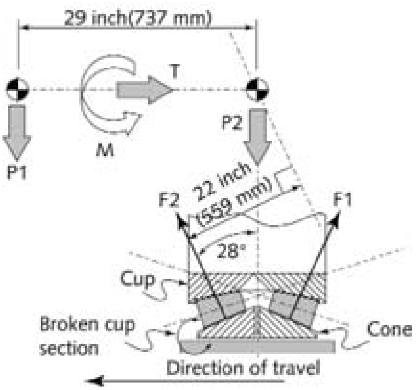


Figure 2.14 Detailed load model of TBM bearing used in analysis of retaining ring deflection

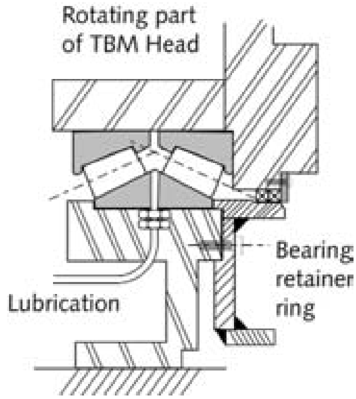


Figure 2.15 Original design of bearing mounting. This partial view shows the relatively flexible retainer ring

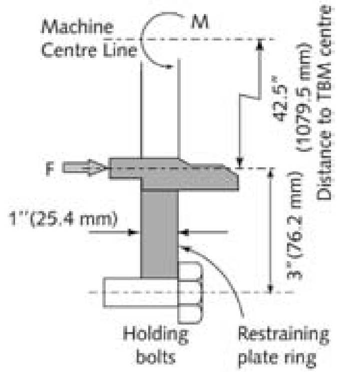


Figure 2.16 Bearing retainer ring dimensions used in analysis of deflection

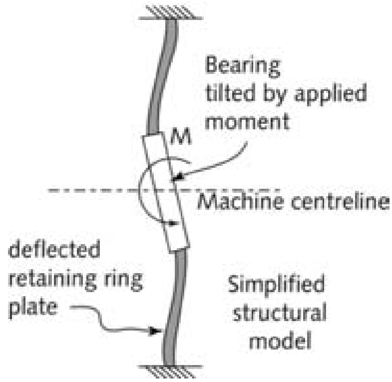


Figure 2.17 Simplified model of the retainer ring deflection

65% of the time $M = 2.7 \times 10^7$ inch-lbs (3×10^6 Nm)

Henry Timken patented the first taper roller bearing in 1898, because he recognised the need for a rolling element bearing capable of carrying a combination of thrust and radial loads. The Timken Company was responsible for developing bearing life formulae based on an observed level of damage to the bearing surface.[2.8] When the Torrington Company (now part of the Timken organisation) supplied the S-3487-C bearing to the tunnelling machine manufacturer, they carried out a life calculation using the above load conditions. After confirming their estimates of bearing loads it

2.8 Interested readers can find detailed information about bearing life calculation at www.timken.com/products/bearings/fundamen/calculate.asp; www.bearings.machinedesign.com/guiEdits/Content/BDE_6_1/bdemech6_60-1.aspx; www.epi-eng.com/BAS-BearingLife.htm.

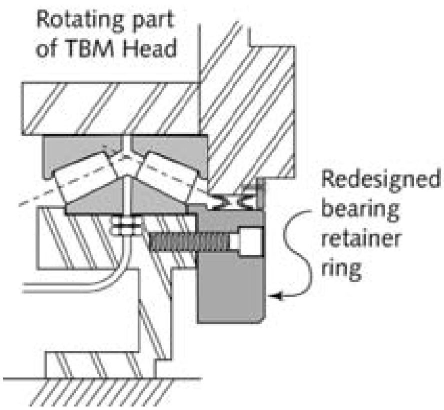


Figure 2.18 Bearing mounting with redesigned retainer ring

was found that their conservatively calculated life of 21,000 hours far exceeded the operating requirements of the contract in dispute.

Figure 2.14 is a schematic representation of the various loads acting on the bearing and in turn on the retainer ring designed to maintain the bearing in its mounting within the deflection limits specified by the manufacturers. Torrington's specification for this bearing was a maximum tilt of the central plane of the bearing of 0.5 minutes of arc. Consequently it was necessary to investigate the mounting geometry and its flexibility.

Figure 2.15 shows part of the sectional drawing of the TBM with its original bearing mounting configuration.

Figure 2.16 shows a partially dimensioned section through the retainer ring and Figure 2.17 is a simplified structural model adopted for the evaluation of flexure of the retainer ring. It was modelled as a simple constant-thickness annular circular plate with a built-in outer edge (the holding bolt pitch circle), with a moment applied at the annulus.[2.9]

Using even the smallest estimated moment (3×10^6 Nm) acting on the bearing, the flexure of the retainer ring permitted the bearing to deflect by approximately 2 minutes of arc. This value exceeded the manufacturer's safe recommendation of 0.5 minutes of arc by a factor of four for about 65% of the operating time. The rest of the time the flexure was greater than this conservative value. It was estimated that the outside edges of rollers in the front row of the bearing "rode" on the front lower part of the cup that eventually broke away, permitting the head to fall off.

2.3.6 Lessons Learnt from This Case[2.10]

Figure 2.18 shows (approximately to the same scale as the original retainer shown in Figure 2.15) the sectional drawing with the modified bearing retainer. Clearly, the new retainer ring has been substantially increased in thickness, as a result of this investigation. Fortunately, there was sufficient space available in the original design to permit the replacement of the original retainer with this substantially thicker retainer ring.

2.9 Readers interested in the detailed calculation of flexure should refer to Young, (1989) p. 435, Article 10.2.

2.10 For his wisdom and guidance in the tunnelling project I am indebted to David Sugden.

When faced with the complex and interwoven threads of information initially presented to me I found it hard to imagine that the accident might have been caused by a design error. After all, I was investigating an accident involving the machine of a highly reputable manufacturer with respected international expertise in tunnelling. Only after assembling and reviewing all the various ways in which the accident might have been initiated did it become necessary to examine the detailed design of the machine. In the words of Sir Arthur Conan Doyle, as spoken by Sherlock Holmes *"... when you have eliminated the impossible, whatever remains, however improbable, must be the truth."*

2.3.7 Outcomes

A relatively short-term outcome was that the machine was capable of repair and return to operation within a few months. The supplier of the TBM was successfully sued for the costs involved in repairing the TBM. Unfortunately for CP's insurers, there was sufficient uncertainty about the actual operating conditions of the machine and this brought into some doubt the range of forces specified in the CP tender for the TBM supplier. As a consequence the liquidated damages component of the dispute was settled out of court.

A longer-term outcome of this case was that Swallows initiated a substantial research programme on the direct measurement of forces experienced by tunnelling machines while excavating hard rock.[2.11] When I first became involved with this case I was offered the opportunity to be "brought up to speed" about the design and manufacture of tunnelling machinery. Part of this process involved a visit to the Swallows factory in Seattle California. There I met Dick Swallows, the chief designer and CEO for the company. At that time Swallows had considerable numbers of tunnelling contracts in progress and they were very interested in collecting hard rock data from wherever tunneling was going on. During this brief interview I was asked to initiate a programme of instrumentation on the repaired Rawaj machine. Dick was interested in rock mechanics and my interest was the design of machinery. I remarked that *"we will also measure the forces on the cutters during this process".* Dick said, as he calmly wrote out a cheque for US$50,000 (the first installment of our research programme) *"why bother?; we already make these maches as strong as we can".*

2.3.8 Technical Analysis

A key aspect of the technically evaluating the deflection of the bearing retaining ring was the estimation of the statistical distribution of loads acting on the machine head during the excavation of the tunnel. This estimate was based on rock mechanics data available from the drilling surveys taken by CP when initially quoting for the tunnelling contract.

2.11 See Samuel and Seow (1984).

2.4 Brinelling Induces Unacceptable Vibrations in a Very Large Bottle Filler

"Filling and capping are tasks of central importance in beverage and food production for only once the containers have been filled and capped using a method which is suitable for the product and at the highest technological level can the best product be manufactured for the consumer. We regard it as our duty to create the conditions for the right filling technology.

With a programme of rotary fillers which is rich in variety, Krones is offering the correct solution for a broad product spectrum. Especially designed to suit product demands, the individual equipment components guarantee an optimum output and the best product treatment.

Mechanical, electronic, volumetric, gravity or vacuum filling systems and a multitude of system variants provide the correct solution for each individual application. The product summary shown here provides you with the entire filler series at a glance. It goes without saying that the different variants have been adapted to suit the different container types such as glass bottles, PET bottles or cans.

After filling comes capping – and Krones can supply the correct capping technology for your container, meaning that the filling and capping process can be carried out perfectly using a continuous system, using intelligent solutions and the most modern technology." Krones AG[2.12]

2.4.1 The Case Culture

Although there are many manufacturers providing automatic packaging machinery for a variety of beverages, few are able to provide machinery for the high-volume packaging of beer. The company involved in this particular case, let me call it The *Bittersweet Lager Company* (*BLC*), produces approximately 1.4 billion *stubbies* annually at its plant located at *Tattaly*, a small country town in Northern Australia. A stubby is a 350 ml bottle designed for efficient packaging. There are few packaging machinery suppliers capable of delivering this rate of throughput reliably even with several machines operating simultaneously. At the Tattaly plant there are three production lines, each with its own washer and filler machine supplied by

Figure 2.19 A 350 ml Stubby

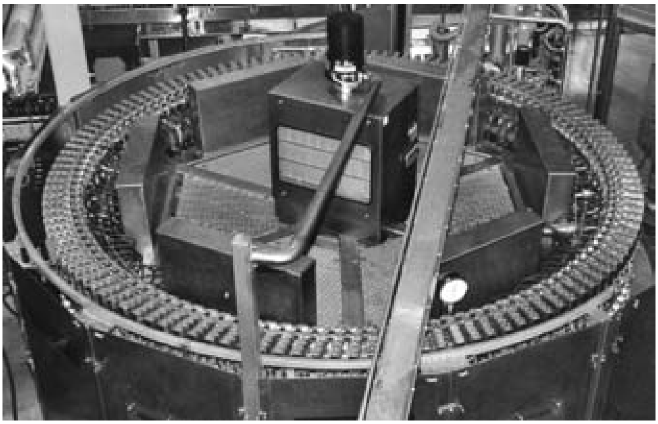


Figure 2.20 A general view of the bottle filler operated by BLC at their Tattaly plant.

2.12 http://www.krones.de/krones/en/104_110_ENG_krones_group.htm.

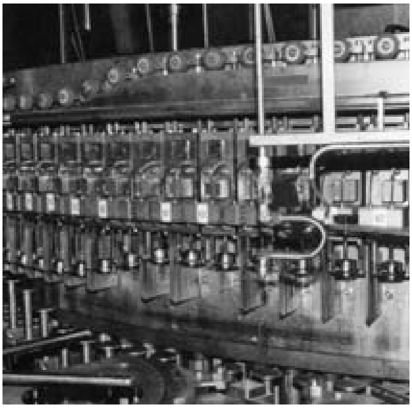


Figure 2.21 Close up view of filling stations

Figure 2.22 Bearing damage. The upper photo shows the damage to the inner (fixed) race. The lower photo is that of the damage to the moving (outer) race

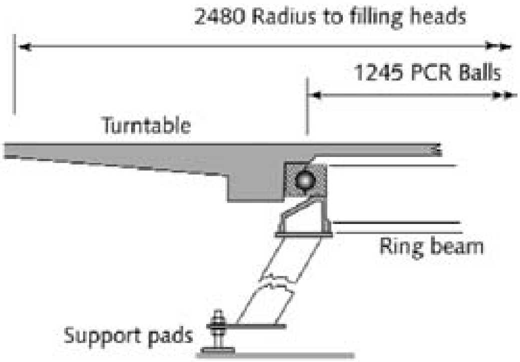


Figure 2.23 Schematic section of filler showing bearing mounting arrangement (not to scale)

Figure 2.24 Photograph of a typical leg support pad

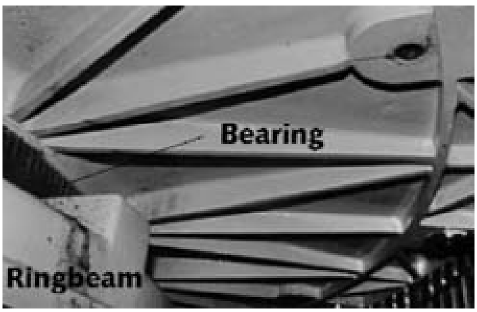


Figure 2.25 Photograph under turntable

Figure 2.26 Ball and spacer arrangement

GBF corporation. Figure 2.19 shows a typical stubby bottle, Figure 2.20 is a general view of the bottle filler at Tattaly, Figure 2.21 is a close-up view of filling stations and Figure 2.22 is a sample photo of the typical damage observed on the bearing races. Figure 2.23 is a schematic section through the filling machine showing the location of the failed bearing, nominally a KD600 *slewing -ring* bearing, with full details of its life and loading curves available from the *Rothe Erde Large-Diameter Antifriction Bearing Catalogue*. Figure 2.24 is a close-up photo of one of the legs and levelling pads on which the filling machine is supported. Figure 2.25 is a photo showing the underneath of the machine turntable and Figure 2.26 shows a photo of the balls and spacers of the bearing. These balls are made of hardened steel and in their travel inside the bearing they exert substantial local loading on the bearing surface. If the bearing race has any surface imperfections or some foreign particle imposed on it from lubricating grease contamination, the passage of these hardened balls over these imperfections or foreign matter will generate an exaggerated jarring loading on the bearing surface. On first inspection the surface damage to the bearing shown in Figure 2.22 was suggestive of this type of initiating process.

The problem with vibration in a large high speed filling machine is the resulting bottle misalignment during filling and the consequent underfill as well as the failure to achieve proper closure on the crown seals of bottles. Operators at Tattaly were alerted to the problem when these faults began to occur regularly.

Before dealing with the technical issues involved in this bearing failure, it is essential to provide a clear picture of some company strategies that would influence and possibly constrain the outcome of this case and the way in which the investigation might be reported by the expert. As one of the largest beverage companies in the world, BLC has many plants with similar filling lines, all of them supplied by the same GBF. The occasional failure of one packaging machine would be relatively easy to absorb into general maintenance by BLC. In addition, it is most likely that GBF would compensate BLC for the cost of repairs. This was certainly the case with the first failure of the bearing. With their multitude of large continuous-flow production lines, it was not so much the one failure at Tattaly that concerned BLC. It was the odds that this was not simply an isolated case but perhaps a symptom of some more substantial underlying problem with all their bottle fillers. BLC had invested heavily in GBF machines in its several plants. Moreover, they were reliant on GBF for the supply and service of these unique large-volume bottle filling machines. To some extent BLC and *GBF* were *in bed together* as far as the continued success of developing fast and reliable bottle fill processing plants were concerned.

2.4.2 The Defining Event

Table 2.1 shows a gross event chronology for the vibration problems experienced at Tattaly. A failure, such as the one described in Table 2.1, can send

Table 2.1 Event Chronology

Date	Event
Late 1993	During a substantial upgrade of the Tattaly plant of BLC, a new high-speed GBF filling machine was installed and commissioned.
Early 1994	A plastic leg support on the new filling machine broke and as a result the machine received substantial vertical jarring. GBF replaced the broken support and the machine appeared to be operating successfully.
Late 1996	Serious problems were experienced with misalignment between bottles and filling stations on the machine. This resulted in several recurring maintenance outages, which were eventually traced to the damaged main bearing on the machine. The bearing was replaced in January 1997 and the machine had been operating successfully since then.
1997	Continuous monitoring of lubricating grease in all BLC plants was initiated in 1997. This monitoring revealed unacceptable levels of metal particles in the Tattaly machine lubricant a few months after the bearing had been replaced. This investigtion was commisioned by the technical staff at BLC to establish the real cause of failure of the main bearing on the GBF machine and to estimate the likelihood of similar failures occurring in other similar machines at other plants operated by BLC.

shivers up the spine of any high volume filling machine operator. This would be especially so if the operator had invested considerable capital in several packaging lines all closely dependent on similar machines. Every minute that the filling machine sits idle during unscheduled maintenance, production is shorted by 4250 stubbies. As a consequence, this investigation was intended to offer clues to the failure scenario with the original bearing. In addition there was substantial concern about the likelihood of an incipient second failure with the Tattaly machine and similar failures in other BLC operations.

Figure 2.26 is a photograph of the 35 mm diameter balls and the plastic spacers used in the machine bearing. The bearing outer ring has a slot that permits the entry of balls into the bearing during assembly. Lubricant grease is pumped into the bearing at regular time intervals and there is a location on the bearing where the "new" lubricant extrudes "older" used lubricant from the bearing. Samples are taken from the extruded lubricant daily and analysed for metal content. When the content of metal shards in the grease exceeds some level specified by the manufacturer, the bearing is deemed to be near incipient failure. A few months after the failed original bearing was replaced by GBF engineers, the lubricating grease monitoring process showed up with unacceptably high metal particle content. It was this event that initiated the investigation described here.

2.4.3 The Client and Possible Parties to the Dispute

My client was *George Melissande*, the technical manager of BLC operations, who asked for an independent investigation into the causes of failure in the

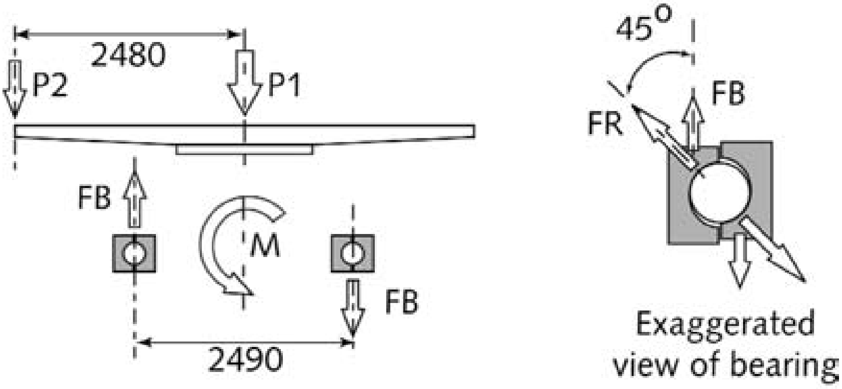


Figure 2.27 Loading on bearing

Rothe Erde slewing-ring bearing as well as an assessment of the overall reliability of the bearing. In essence this case was in the early stages of disputation planning. Had my investigation uncovered some form of design or material fault, then BLC may have initiated a dispute with GBF concerning the whole batch of bottle fillers and washers operating in all BLC plants. Had my investigations found a fault with the bearing material or its specification on the machine at Tattaly, the resulting disputation may have involved the bearing manufacturer.

2.4.4 The Expert's Role and the Investigation

The expert was asked to investigate the failure of the bearing and offer opinion about probable failure scenarios and if at all possible identify underlying causes for the failure and to evaluate the reliability of the bearing. The investigation focused on the following issues:

(a) *Was the bearing overloaded?* – The GBF company had a strong reputation for supplying beverage filling machines to the packaging industry. There were several such machines in operations around the world and also within BLC's many plants. The other machines were still operating successfully without the type of problem experienced with the machine at the Tattaly plant.

Although the bearing loads were not expected to be incorrectly assigned by the manufacturer, it was necessary to check both the load and the deflection on the companion structure to eliminate this mode of failure. As well as the static load on the bearing there was a small but significant out-of-balance load on the bearing due to the effect of the filling station loads over approximately 100° of arc on the machine circumference. This arc corresponded to 50 filling stations out of a total of 168, where the stubby bottles were pulled down from the filling heads by pull-down cams. Figure 2.27 shows the bearing loading schematically.

$P1$ = 70 kN (estimated by manufacturer)

P2 = 14 kN (provided by *BLC* maintenance staff)

Taking moments about the centre of the machine turntable (refer to Figure 2.27) gives

$$P2 \times 2480 = M = FB \times 2490,$$

$$FB = 14 \times 2480/2490 = 13.9 \text{ kN}.$$

As a result the load carried by half the balls is (70/2 + 13.9) kN or approximately 49 kN. This load is distributed over 100 balls and hence the load per ball is 490 N in the vertical direction. The actual load normal to the bearing surface is found from the vector diagram indicated on the exaggerated view of the bearing in Figure 2.27, yielding

$$FR = 490/\cos 45^\circ = 686 \text{ N}$$

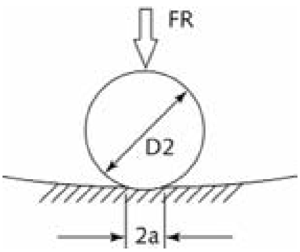
Contact loading of the balls on the bearing surface is evaluated using the appropriate equations from Young (1989), Article 13.1 (also referring to Figure 2.28). E_1 and E_2 are elastic moduli for the sphere and the substrate respectively and for the bearing and ball $E_1 = E_2 = 210$ GPa, μ_1 and μ_2 are Poisson's ratios for the ball and substrate respectively and for the bearing $\mu_1 = \mu_2 = 0.3$. K_D is a factor to allow for the relative curvatures for the two surfaces in contact. Here the conservative estimate for $K_D = D2 = 0.035$ m. Evaluating the various constants and the maximum local stress results in

$$a = 0.43 \text{ mm}$$

$$\text{Maximum } \sigma_1 = 1.77 \text{ GPa}$$ [2.13]

Accordingly, the worst contact stress occurs at the edge of the contact area and is approximately 0.133 (Max σ_1) = 235 MPa. This is well below the allowable stress for the bearing material in its hardened state.

The estimates above were made with the simplistic approximation that bearing load is distributed evenly on all the balls in the bearing. Although this would be the case under normal circumstances, if the machine were to experience a substantial vertical impact, then a severe localised load could



$$a = 0.721\sqrt[3]{FR \times K_D \times C_E}$$

$$Max\ \sigma_1 = 1.5\frac{FR}{\pi a^2}$$

$$C_E = \frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}$$

Figure 2.28 Contact loading of sphere on surface (dimension units in these equations are measured in metres)

2.13 See also Timoshenko and Goodier (1983), Article 140; Samuel and Weir (1999), Section 2.7.

indeed cause local yielding of the bearing surface. The originally supplied machine levelling pads were made of solid polypropylene. These pads were replaced with the metal pads (see Figure 2.24) when one of the plastic pads collapsed during commissioning of the machine. From the evidence of the bearing damage it appeared highly probable that the failed bearing was indeed subject to this type of damage.

Brinell hardness is obtained by indenting a surface with a small hard metal sphere and measuring the size of the indentation. Hence the permanent deformation of the bearing surface locally due to the imposed contact stress of a ball is referred to as Brinelling. Once Brinelling damage has been imposed on a bearing surface, total deterioration of the bearing soon follows. As balls pass over the damaged section, the bumps experienced under load leads to further Brinelling and the type of overall bearing deterioration seen in the machine at the Tattaly plant is commonly experienced.

Bearing life calculations were made using the life formula and life data available from the maker's catalogue.[2.14] Using the life formula for Rothe Erde bearings the estimated life of the bearing under investigation was estimated at approximately 200 years. Based on these results it is certain that the bearing in the filling machine at Tattaly was very much under-loaded and life overload was unlikely to have been the cause of the observed bearing failure. Naturally there are several other technical issues which could shorten the life of a bearing and these are all noted in the Rothe Erde bearing catalogue. Only one of these issues was considered as relevant to this investigation, namely the behaviour of the companion structure on which the bearing is mounted.

(b) *Was the bearing properly mounted or had the companion structure of the bearing suffered some unexpected or unacceptable deflection during installation or commissioning?* – This last question is an expression of the commonly observed engineering paranoia when an unexpected machine failure is encountered (i.e., "*Was it dropped?*", or "*Has someone hit it with a hammer?*"). As it happened, in this case there some was cause to be concerned about a random accident event. The machine did suffer some vertical jarring when a plastic support pad under one leg of the machine collapsed during installation.

The design of the Rothe Erde Series KD600 slewing-ring bearings is based on a relatively frail structural component being supported on a stiff companion structure. Considerable care needs to be taken to minimise bearing deflection under load and the companion structure of the bearing needs to withstand the loads without undue deflection. The flatness of the supporting surface under the bearing race is to be paid particular attention, as is the tension in the hold-down bolts which ensure conjoint operation of the bearing race with its companion structure. From my discussions with

2.14 *Rothe Erde Large-Diameter Antifriction Bearings*: Hoesche Rothe Erde.

maintenance staff at the BLC plant, none of these issues was discussed or advised by *GBF* engineers when the new bearing was installed in January 1997.

Figure 2.25 is a photograph of the underneath of the machine turntable. This photograph indicates the robust ring beam structure used to support the machine bearing. Simple calculations of deflection in the ring beam under normal loading showed these deflections to be negligible.[2.15]

(c) *Was there something peculiar about the bearing material that may have resulted in this failure? – Intico Pty. Ltd.*, an independent approved testing authority had carried out hardness testing on the failed bearing. However, these tests were all conducted on the parent metal and the actual bearing surface was not tested for hardness. As noted earlier, the bearing was supplied by Rothe Erde, a company supplying bearings to the heavy lifting industry. This bearing was designated as a series KD600 slewing -ring bearing, and it is the type of bearing commonly used in large machinery such as cranes that rotate intermittently.

There were samples available from the bearing to be tested for material properties. This was performed by a metallurgical testing laboratory, *STS*, who identified the material as a *45-Cr-2 steel* (a European designation for this chromium steel, designed specifically for surface hardening). The Society of Automotive Engineers (*SAE*) designation for the same steel would be *SAE-5147*. *STS* advised that the desirable surface hardness for this steel should be in the range 55-61*HRC*, corresponding to a Brinell hardness of 550, when induction hardened.[2.16]

The bearing surfaces of the turntable bearing had been induction hardened, a widely used process for the surface hardening of steel in which components are heated by means of an alternating magnetic field to a temperature within or above the transformation range followed by immediate quenching (rapid cooling). When steel is heated above its transformation temperature (720°C), the carbon changes the steel's crystalline structure to an austenite, one of the allotropes of iron, also known as gamma iron. The harder, more brittle steel is then quickly cooled or quenched. The core of the component remains unaffected by the treatment and its physical properties are those of the bar from which it was machined, whilst the hardness of the case can be within the range 350 to 550 Brinell hardness. Carbon and alloy steels with a carbon content in the range 0.40–0.45% are most suitable for this process. The author used a *Churchill* portable Brinell hardness tester to measure the hardness of the bearing surfaces. Both inner and outer race surfaces were tested for hardness, with the results showing that both surfaces had a hardness below 400 Brinell.

2.15 These simple deflection calculations are presented in Appendix 2.

2.16 A primer on material properties and testing methods is given in Appendix 2.

Table 2.2 Potential failure mode evaluation

Potential failure mode	Description	Information available	Likelihood
1	*Overloading; design fault*	*Life curves for given application shows the bearing is underloaded; highly reputable supplier*	*very low*
2	*Lubrication failure*	*Inspection showed the lubricating grease on the bearings was clean and contained no metal particles*	*very low*
3	*Material fault; manufacturing error*	*Material softer than expected, strength still well above levels of concern (refer bearing life calculations)*	*Medium to low*
4	*Bearing mounting fault or companion structure deflection*	*Mounting examined and structure deflections estimated (refer to calculations of companion structure deflections)*	*Very low*
5	*Random accident during installation or commissioning*	*A machine support disc that collapsed during commissioning may have imposed a undue local load on the bearing surface (Brinelling action)*	*High*

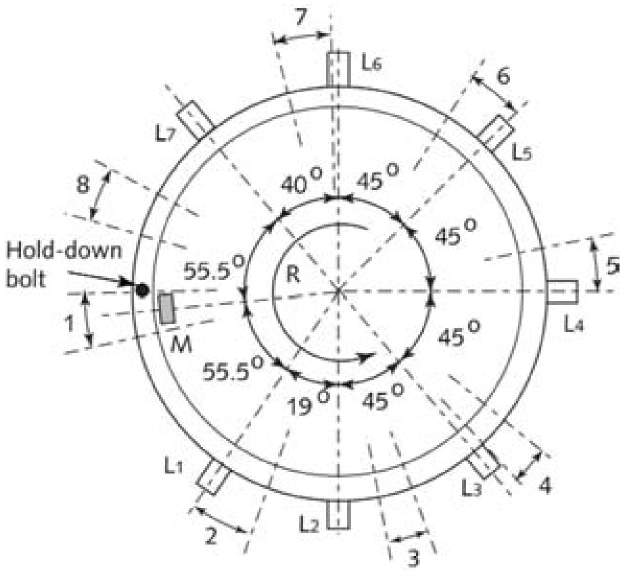


Figure 2.29 Layout of machine showing leg and bearing damage locations on the inner (fixed) bearing surface. The L numbers represent support leg locations, M is the motor drive location, R is the machine rotation direction and the damage locations numbered 1 to 8 were measured arbitrarily from the first hold-down bolt near the location of the motor drive

Table 2.3 Bearing damage survey (refer to Figure 2.29)

Location	Position relative to hold-down bolt 1 (Degrees of arc)	Arc length (mm)	Description
1	11	240	Severe chatter, with very deep indentations; this is the most extensive damage; heaviest near hold-down bolt 1
2	54–73	415	Very light, sporadic surface damage; mostly surface discolouration
3	99–107	175	Very light discolouration; few minor surface indentations
4	137–147	218	Very severe damage, with deep indentations; most severe near hold-down bolt 15
5	179–189	218	Severe chatter; less than at locations 1 and 4, but quite deep grooves; worst near hold-down bolt 19
6	223–232	196	Relatively light damage; less than at location 5; heaviest at midpoint between hold-down bolts 23 and 24
7	257–266	196	Very light damage; mainly surface discolouration, with a few shallow indentations
8	310–320	196	Heavy damage; deep indentations of the same order as at location 4

Table 2.2 is consolidated statement of the probabilities of potential failure modes considered in the initial investigation. Table 2.3 gives a detailed description of the inner (fixed) bearing surface damage. In detailing the damage, the locations are referenced with respect to the hold-down bolt holes in the inner race. There were 36 hold-down bolt holes spaced at 9.73 degrees of arc apart. Hold-down bolt hole designated number 1 was located arbitrarily just ahead of the drive motor location. This survey was conducted in order to establish if there was any pattern of correlation of damage with machine leg or hold-down bolt locations. There was no such correlation apparent in the damage locations observed.

Notably, the outer race is not loaded as heavily as the inner race, since both principal curvatures of the surface are of the same sign as the curvature of the bearing ball. Moreover, the ball path on the outer race is considerably longer than the ball path on the inner race (larger radius to contact surface). Hence the number of load cycles seen by the outer race are smaller than those seen by the inner race during the bearing life. These observations were supported by the fact that the outer race surface appeared to be less damaged than the inner race surface.

Finally, if the failure of the bearing were due to an event such as material, lubrication or companion structure failure, the outer race would be uniformly damaged rather than in a few specific locations. It is this set of randomly spaced failure locations on the outer bearing surface that presented the strongest evidence for a jarring initiated failure event, leading to the eventual failure of the whole bearing at series of specific locations.

2.4.5 Lessons Learnt and Recommendations

Probably the most salutary lesson from this case was the rather complex and convoluted nature of the relationship between the two protagonists BLC and GBF. As noted earlier, it was very much in the interest of both companies to ensure the continued and reliable operation of these filling machines. The investigation did not uncover any sinister underlying flaw with the Tattaly machine. As a consequence the following recommendations were offered to BLC technical staff:

1. Advise GBF about the nature of the failure event together with the information about this investigation. In my opinion BLC were entitled to some compensation in addition to the replacement bearing, considering that GBF were responsible for supplying the machine originally with unsatisfactory leg supports. The resulting jarring and damage initiation of the bearing appeared to be a machine design fault which should be borne by the machine supplier. This issue is supported by the evidence that the surface hardness of the bearing appeared to be 34% lower than expected.
2. Examine all other GBF machines for possible jarring which could initiate early failures similar to that experienced in the machine at Tattaly. This could be done in the following ways. A vibration survey of these machines could reveal unaccounted for resonances in their vibration spectrum. Continuous monitoring of lubricant grease could reveal unacceptably high metal content, providing a warning of incipient failure. In both of these cases the bearing could be removed and examined for surface damage during scheduled maintenance shut-down. Alternatively, and this is probably the most expensive option, the bearing of each machine could be scheduled for removal and full inspection for surface damage or brinelling at scheduled maintenance shut downs.
3. Request full life design data information for all the slewing-ring bearings in GBF designed filling machines operating at BLC plants. This design data should be available from GBF or Rothe Erde. It would provide some degree of appreciation of probable machine reliability for BLC technical staff if they had a clear understanding of how design decisions associated with these bearings were reached.
4. Monitor bearing lubricant for metal content on a regular basis to ensure that the bearings operate without shedding metal particles.

2.5 A Milk Tanker Takes a Spill

In many countries the bulk processing of food products is carried out in plants separated by substantial distances. The transportation of food products is a service requiring specially designed transport equipment. This case concerns the transport of raw milk collected from dairy farmers and transported to a food processing plant using a large milk tanker truck.

2.5.1 The Case Culture

Port Campbell is a small picturesque seaside township located on the doorstep of the world-famous Apostles along the Great Ocean Road in Victoria. *Cherry Transport* has its centre of operations there because it is in the heartland of the Victorian dairy industry. The town of Cobden is located in central Victoria where a large milk processing pant is operated by *YumYum Foods*. Milk tankers are used to transport large volumes of milk from dairy suppliers to milk product processing plants such as *YumYum*. Collection of milk is performed by the tanker driver using a stainless steel tanker trailer driven by a prime mover. The connection between the trailer and prime mover is achieved by the operation of a device called a *greasy plate hitch*. The tanker used in this case was carrying approximately 23,000 litres of milk to be processed into cheese by YumYum Foods. Cherry Transport was the tanker operator and Alan Cherry, a nephew of the owner, drove the tanker when it rolled over while negotiating a bend in the Port Campbell to Cobden road and spilled its milk contents, as well as sustaining some damage to the tanker and the greasy plate hitch.

2.5.2 Defining Event

In large transport accidents the police are routinely asked to attend the scene, interview the driver or any witnesses and subsequently file a report of the accident. The police report of this accident simply stated that the tanker trailer had rolled over and the prime mover was facing in the direction of travel. Road conditions at the time of the accident were described as slightly wet with clear visibility. As would be expected, the driver claimed to be travelling at 25 km/h, a speed he deemed appropriate to take the bend with the tanker.

When the service histories of the tanker and prime mover were routinely examined by the loss assessor for Cherry Transport's insurer, it was discovered that at some time prior to the accident event the kingpin bolt of the greasy plate hitch was replaced by *Blunt Engineering*. The replaced kingpin bolt was of a non-standard design, manufactured by Blunt Engineering to expedite the service of the greasy plate hitch when a standard bolt was not available during service. The damages claim was issued by the solicitor acting for the insurer of Cherry Transport against Blunt Engineering, claiming that the non-standard kingpin bolt was the major cause of the accident. The quantum of the claim included AU$ 9000 for repairs to the Louiswille prime mover, AU$ 33,000 for repairs to the tanker trailer and AU$ 3000 for the lost milk. The total claim, not including costs of litigation, was AU$ 45,000.

2.5.3 Parties to the Dispute

Although this was a *small* claim that under normal circumstances would have been handled by Cherry's insurer, the failure of the non-standard kingpin bolt gave the impression that an engineering failure had contributed to the accident. Consequently, Cherry's insurer sought to distribute their losses by suing Blunt's insurers. It is a sad fact of life that almost invariably these cases devolve into a loss sharing fight between two or more insurance companies.

2.5.4 The Expert's Role and the Investigation

Defence counsel for Blunt's insurers sought my services as a consultant and asked me to respond to the following:

(a) What was the most likely scenario for the accident resulting in the roll over of the Cherry Transport tanker?;

(b) In what way did the kingpin bolt influence the outcome of the accident?

(c) Finally, could it be established whether the non-standard kingpin bolt may have adversely contributed to the outcome of the accident?

In what follows, the operation of the greasy-plate hitch will be reviewed and an accident scenario constructed. Figure 2.30 is a general view of the tanker. Figure 2.31 shows the trailer hitch mounted n the back of a prime mover. A schematic cross sectional view of the greasy-plate swivelling arrangement is shown in Figure 2.32. The kingpin bolt in the centre of the greasy-plate ties the rotatable top plate to the fixed bottom plate. The spring permits some limited vertical movement between the two plates. The trailer quick-fit hitch is fixed to the top plate and the tanker hitching pin is

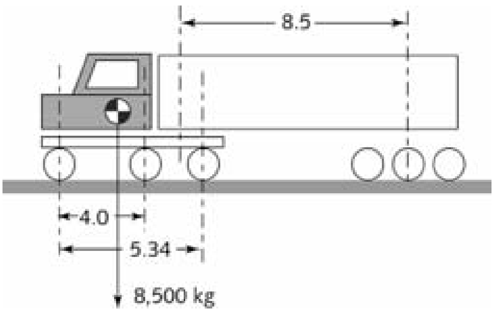


Figure 2.30 General view of the tanker

Figure 2.31 The trailer hitch mounted on a prime mover

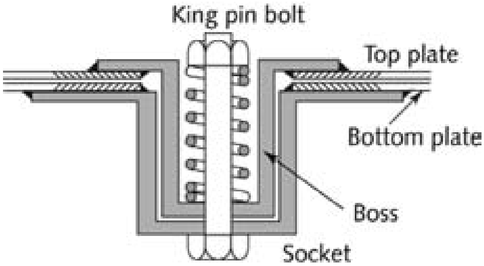


Figure 2.32 Greasy-plate arrangement (schematic, not to scale)

Figure 2.33 Hitching pin mounted on tanker trailer

received in the vee-shaped entry during the trailer hitching process (Figure 2.33). Shear loads are carried during transport by the two cup-shaped components of the top and bottom plates. Rotation of the top plate and cup on the bottom plate and inside the bottom plate cup by grease lubrication. The only time the kingpin bolt would experience any substantial loading beyond the force developed in the spring is when, due to unexpected vertical displacement the spring coils become fully compressed. Figure 2.34 shows the damaged kingpin bolt recovered after the accident. Also shown in this figure are schematic sketches of a standard kingpin bolt and one like that manufactured by Blunt as a replacement for a standard bolt. The major difference between these two types of bolts is that in a standard bolt the head of the bolt is integrally made with the body of the bolt, while in the manufactured replacement the body is machined and the head is welded on as indicated in Figure 2.34. Note that the damaged kingpin bolt has its bolt head missing and it is also slightly bent.

After the accident the loss investigator found the kingpin bolt to be a non-standard type of bolt manufactured with the head of the bolt welded on rather than formed normally. Metallurgical examination of the damaged bolt confirmed the suspicion of the loss investigator that the bolt head welded on by Blunt had been torn off the bolt stem in the accident. In subsequent statements by Blunt Engineering it was admitted that they replaced the worn original kingpin bolt with one of their own manufacture. This action was taken as a means of speeding up the delivery of the reconditioned trailer hitch. A report of the examining metallurgist alleged that the manufactured bolt had only about one third the strength of the replaced kingpin bolt.

One aspect of constructing this particular traffic accident scenario was that of modelling the behaviour of the tanker in the turn. In this case the initial modelling was conjectured as indicated by the model pictures in Figure 2.35. The conjectured chronology of the tanker rollover was as follows:

(a) Driver takes the turn faster than appropriate to the road conditions;

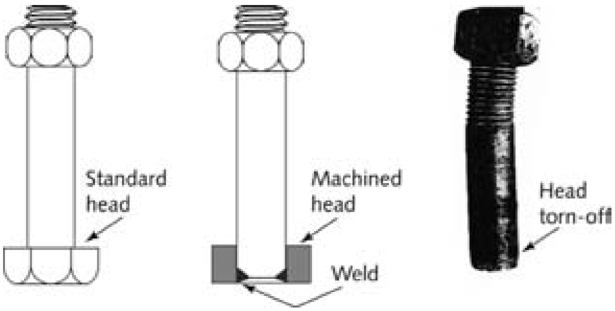
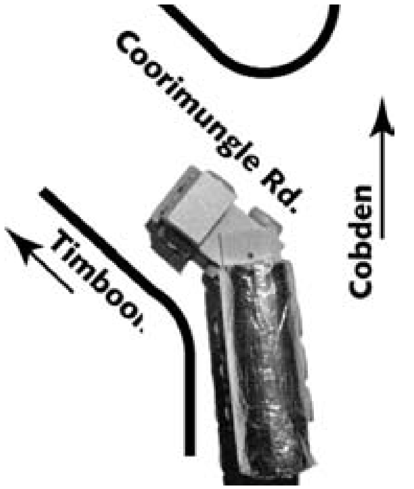
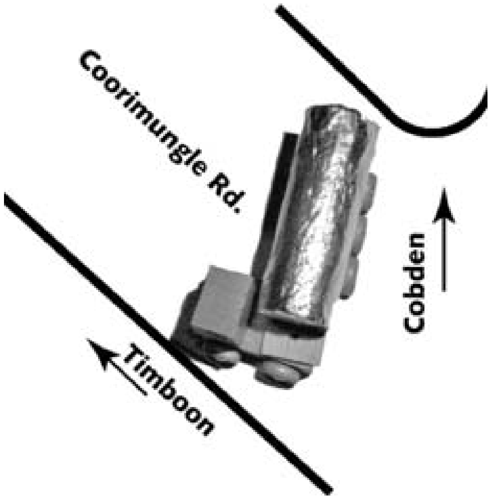


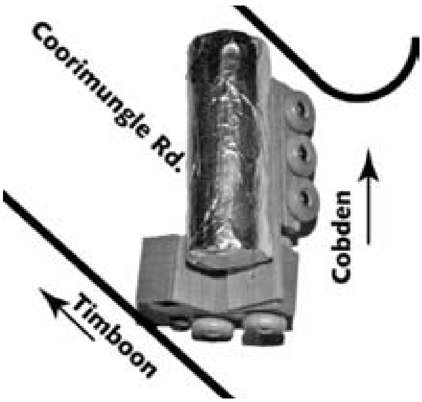
Figure 2.34 The damaged king-pin bolt and schematic sketches of a standard bolt and one similar to that manufactured by Blunt



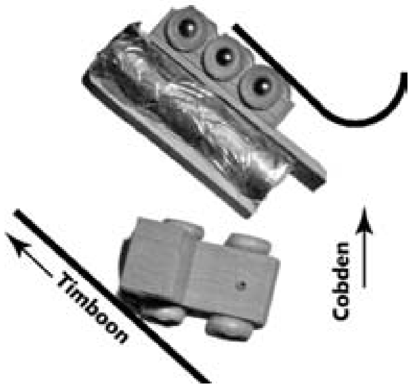
(a) Tanker approaching turn



(b) Tanker near jack-knife condition



(c) Commencement of rollover



(d) Final locations of tanker and prime mover

Figure 2.35 Model demonstration of the tanker rollover process (not to scale)

(b) Tanker trailer begins to slide on wet road;

(c) Inexperienced driver applies brakes;

(d) Tanker trailer rolls and tears out kingpin bolt from the bottom plate of the hitch. Trailer and prime mover settle in locations indicated in the photo of Figure 2.35 (d). This was confirmed by the police report of the accident scene.

A professional road traffic modelling expert, *Peter Bitterman*, was consulted about the conjectured scenario depicted in Figure 2.35. In his report Bitterman stated:

"Under conditions of good tyre/road surface friction the most likely form of instability is rollover. This involves large centrifugal forces acting laterally at the centre-of-gravity of the trailer and exceeding the stabilising influence of the vertical loads between the trailer tyres and the road surface. The University of Michigan Transportation Research Institute (UMTRI) Roll Model was used the simulate the transport vehicle and to estimate its rollover limit."

The results of this simulation showed that at at 25 km/h the trailer wheels would lift but the vehicle would remain stable. The same analysis showed that at 30 km/h the vehicle would roll over. Bitterman went on to conclude:

"Under conditions of poor tyre/road friction, and combined braking and cornering, the most likely forms of instability are jackknife and trailer swing.
Trailer swing involves the rotation of the trailer about the turntable while the prime m over continues in its path. In a left turn the trailer would always swing to the right and may swing far enough to damage the right side of the cab.
… the police report … clearly shows damage to the right hand corner of the cab only …"

Figures 2.36 and 2.37 show schematically (again conjectured) the two phases of the rollover. In the first phase the tanker trailer will lift the rear of the prime mover producing substantial tension on the kingpin bolt. Once the spring (see Figure 2.32) had been fully compressed, the edge of the top plate would have become the fulcrum about which the rollover moment acted on the bolt. This second phase of the rollover was the most probable reason the kingpin bolt was torn from its location. The two phases were most likely only a fraction of a second apart.

Taking moments about the ground reaction in Phase 1 of the rollover

$$F1 = (8555 \times 9.81 \times 2.5)/4.7 = 44.4 \text{ kN}$$

In Phase 2 of the rollover the moment acting on the top plate due to the rolling over of the trailer is $F2 \times 0.45$ Nm. $F2$ could have been estimated from the conjectured centrifugal forces acting on the rolling trailer at various road speeds and turn diameters. However, a simpler approach was to recognise that the bolt had indeed been torn out from its location and the twisting moment responsible for this was as indicated in Figure 2.37.

$$F2 \times 0.45 = 8{,}500 \times 9.81 \times 2.5$$

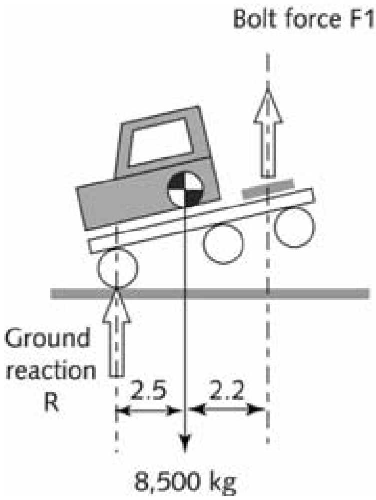


Figure 2.36 Phase I of the roll

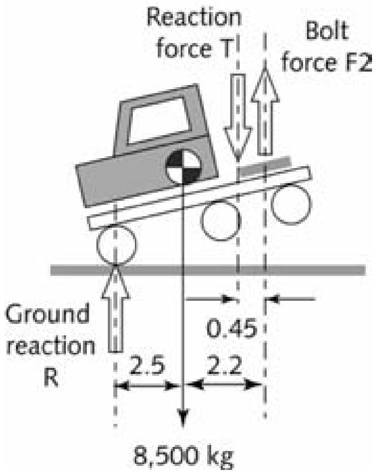


Figure 2.37 Phase 2 of the roll

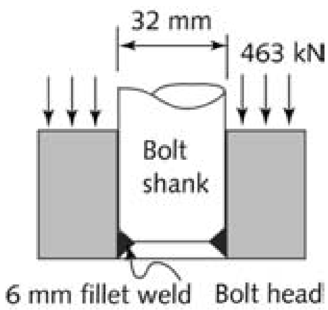


Figure 2.38 Schematic view of bolt head under load

$F2 \quad = 463$ kN

Figure 2.38 is a schematic view of the bolt head under load during Phase 2 of the roll. The stress developed in the weld is given by[2.17]

$\tau \quad = (1.21 \times F2)/(\mathrm{L} \times \mathrm{h}),$

$L \quad =$ weld length $= \pi \times 0.032$

$h \quad =$ weld height $= 0.006$

and $\tau \quad = (1.21 \times 463 \times 10^3)/(\pi \times 0.032 \times 0.006)$

$= 0.93$ GPa.

This is a very large shearing stress that would have easily failed the weld holding the machined head of the bolt onto the shank. However, the standard bolt originally used in the hitch was a *Duraflex* CS1045 steel bolt with a tensile strength of 440 MPa.[2.18] In the event that a standard kingpin bolt had been used in the hitch during the accident the failure criterion for this standard bolt would have been the tensile strength of the bolt shank.

Maximum tensile load on the bolt:

$$\sigma_{\mathrm{Max}} = F_{\mathrm{Max}}/(\pi \times D^2/4) = 463 \times 10^3/(\pi \times 0.036^2/4) = 455 \text{ MPa}$$

2.17 See for example Samuel and Weir (1999), Section 4.4.

2.18 This information was provided by Cherry technical staff.

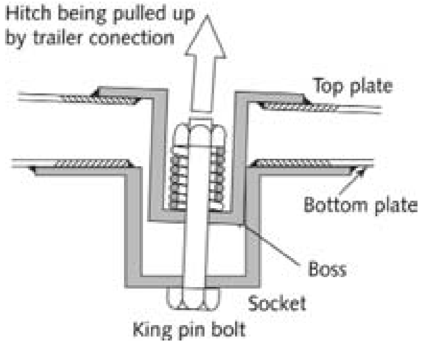


Figure 2.39 Greasy-plate hitch at its extreme displacement

This calculation does not take into account the failure of the bolt at its thread, where the root diameter is smaller than the shank diameter, and stress concentrations are introduced by the thread elements. The significance of the calculated stress on the bolt is that it would have failed in any circumstances whether it had been a standard bolt or a modified manufactured bolt. This bolt is not expected to carry substantial loading at any time other than when the hitch spring (refer to Figure 2.39) is fully compressed. In this condition the load is taken by the kingpin bolt head and nut as well as the metal bases of both boss and socket components. Under sufficient loading the thread on the kingpin bolt may shear off, the head of the bolt might be pulled through the base of the socket, or the kingpin bolt might fail in tension. These failure modes are listed in order of greatest to least likelihood, estimated from the engineering mechanics of the greasy-plate assembly. In the Cherry Transport accident, the kingpin bolt head was welded on to the bolt stem, as indicated in Figure 2.38, and it was the fillet weld of this construction that became the weakest part of the greasy-plate assembly.

2.5.5 Lessons Learnt and the Outcome

In this case as in many similar examples, the careful reconstruction of a failure scenario was an essential part of the defence for Blunt. When the large load acting on the kingpin bolt was applied to a standard bolt, it became evident that no bolt of the size used in this application would have been able to withstand the load acting in Phase 2 of the rollover. In fact, had the bolt been able to withstand this load, it is entirely feasible that the momentum of the tanker trailer would have carried the prime mover with it in the rolling process. If that happened, the prime mover would have suffered significantly greater damage than was the case in this accident. In fact, one might consider the failure of the bolt head acting like a fuse in an electrical circuit, preventing a more substantial damage.

In this case the winning line was that of recognising the fine detail of the accident scenario. This case was (as in most cases) settled out of court as a result of the above evaluations and the additional evidence provided by the transport dynamics model.

2.6 A Paper Coating Machine is Damaged in Transit

Paper with a clay or other coating applied to one or both sides is coated paper. The coating is intended to fill in all the micro-scale irregularities produced when randomly distributed cellulose fibres are compressed into the basic paper web. Coated paper generally produces sharper, brighter images and has better reflectivity than uncoated paper. The coating can be dull, gloss, matte, or other finishes. Many coaters use an *airknife* to aid the coating process where the coating is applied to the substrate and the excess is 'blown off' by a powerful jet from the airknife. Figure 2.40 shows a schematic view of the airknife coating process.

2.6.1 The Case Culture and the Accident Event

Sometime in the early nineties *Busyboard*, a paper maker, set up a new paper coating line in their preprint business at Coolaroo in the state of Victoria, Australia. Paper coating lines consist of several special types of machinery with purpose-built transfer mechanisms for the paper to be coated. In this case, Busyboard decided to out-source the manufacture and installation of the whole line to the *Holker Corporation* of Ohio. Holker had substantial expertise in paper coating lines and they contracted to supply, and install the several components of the line including an airknife coater. The total contract cost for the supply delivery and commissioning of the coating line was US$ 2.18 million. The airknife coater included in this contract was costed by Holker at US$ 253,000.

The machinery for the coating line was packaged and delivered as marine cargo to the port of Melbourne in late 1994. One package, that containing the airknife machine, was found be seriously damaged on delivery to

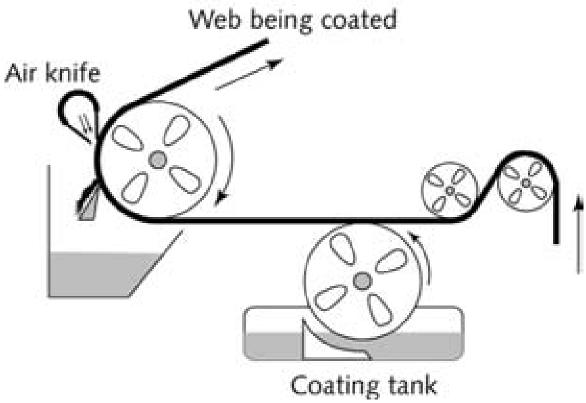


Figure 2.40 Schematic view of the coating process and the role of the airknife

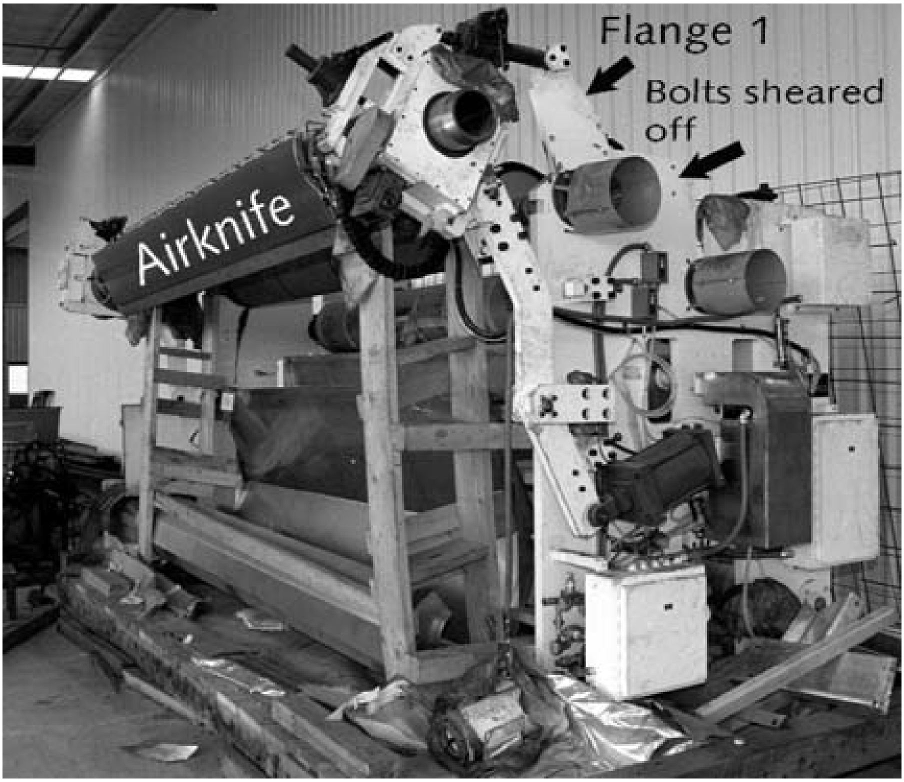


Figure 2.41 General view of the partially unpacked airknife machine in its damaged packaging

Coolaroo. Marine surveyors examine ships' cargoes, investigate accidents at sea and prepare accident reports for insurance purposes. Because the airknife package was damaged somewhere in transit, either on the docks during handling, or on board the ship that delivered it, *Captain Joseph Porter*, a marine surveyor, was appointed to inspect the damage and to report on it to Holker's insurers.

In addition an independent assesor, the engineering firm, *Telfer Ltd.,* was appointed to inspect the damaged machine and advise Holker about the nature and extent of the damage. Based on Telfer's assessment, Holker would then estimate the cost of repairing the damaged machine. Interestingly, the repairs to the damaged machine were estimated by Holker at US$ 520,000. This incredible cost figure included US$ 133,000 for packaging and return air freight (to Ohio and back to Melbourne), on the basis of Holker's insistence that the machine could be properly repaired only in their own works in Ohio. Moreover, once repaired, Holker disavowed any warranty or guarantee of performance for the repaired machine.

2.6.2 The Nature of the Dispute and Stakeholders

It is useful to put in context the claim on Holker's insurers resulting from the damage to the airknife coater. Reviewing the above figures, the supply and air freighting out a brand new airknife coater with full warranty from Holker would have cost US$ 200,000 less than the claimed repairs to the damaged machine. Of course, there may have been some issues of timing and possible liquidated damage implications in the Busyboard contract with Holker that motivated the incredibly expensive claim. Recall here that Holker's complete contract with Busyboard included full commissioning of the coating line. Consequently, one could interpret the denial of warranty and guarantee of performance for the repaired airknife coating machine as impinging only on Holker themselves rather than Busyboard. After all, Busyboard were entitled to receive a fully commissioned coating line as agreed by the original contract suggesting immunity from the implied lack of warranty or guarantee of performance.

The major stakeholders in this prelude to a dispute were the marine insurers of the transport company and Holker's liability insurers. There may have been a subsidiary party to the dispute, namely Telfer Ltd., the independent damage assessors. Their assessment of the damage would require evaluation, considering that they may have been influenced by the time and ultimate performance constraints imposed on the repairs and return to service of the machine.

2.6.3 The Role of the Expert and the Investigation

Early in 1995 Porter commissioned me to investigate the damage to the machine and to report on the possibility and implications of repairing the damaged airknife coating machine locally. My task was to review the available evidence and to report on the assessed need to air freight the machine back to Ohio for repairs.

Figure 2.41 shows the partially unpacked airknife machine. Figures 2.42 through 2.45 show closer views of various elements of the airknife machine. On first inspection it appeared that the machine in its crate had suffered a substantial bump. The most easily apparent damaged items were found to be as follows:

1. The airknife support system at the front of the machine appeared to have suffered the most serious dislocation. This is identified in Figure 2.41, where Flange 1 is seen to be displaced from its intended location. The four holding screws, designed to retain the airknife support system in its intended location, had been sheared off due to the impact received in the accident event. Figure 2.44 is a close up photograph of the airknife support head, indicating the original location of Flange 1. In fact, all the eight (four on each side) 15 mm diameter cap screws holding these support flanges on both sides of the machine had been sheared off flush with the machine frame.

2. Various support rollers had been displaced from their bearings and some had bent shafts. Almost all bearings supporting rollers had been shattered due to the impact on the machine.
3. Pneumatic actuating cylinders had been damaged (see for example 2.43).
4. The air delivery chamber (seen in Figure 2.42) had impacted on one of the support rollers and may have become damaged. This needed careful evaluation in the repair of the damage.

There was no doubt that the machine had suffered substantial damage, but its repair prognosis had to be be seen in the light of:

(a) The type of event that could have caused the damage (hypothetical accident scenario)

Figure 2.42 View along air delivery nozzle, with airknife blades removed

Figure 2.43 View showing roller dislodged from its bearing and damaged air cylinder

Figure 2.44 Airknife support head and its various adjustments. The arrows show where the flange bolts sheared off

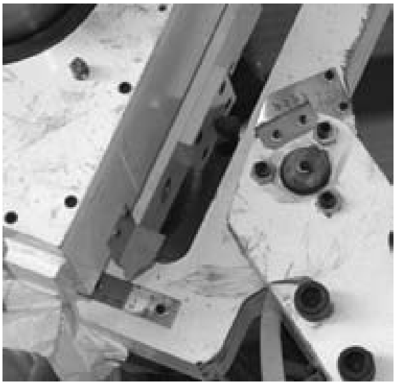


Figure 2.45 Close up of support head indivating gap and attitude adjustments

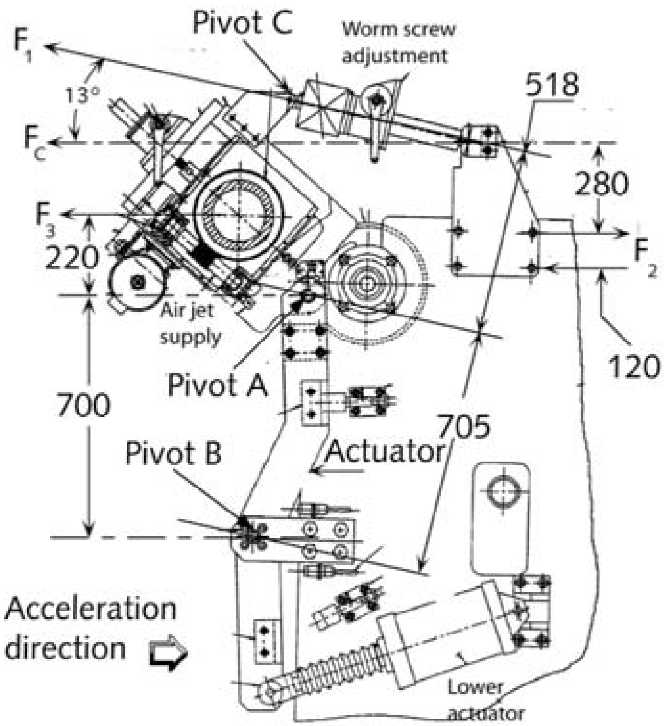


Figure 2.46 Side view of the airknife coater indicating some of the major features of adjustment to the machine

(b) The likely consequences of the accident for the overall machine assembly

(c) The likely risks for the machine operator, working with a repaired machine in a critical position of a continuous production facility.

(a) The Accident Scenario – Figure 2.46 shows a side elevation of the airknife machine taken from a copy of drawings supplied by the Holker Corporation. The measurements were scaled from the drawing, using comparative dimensions measured on the actual Holker machine on site. Figure 2.47 is a simplified mechanism view of the moving parts of the airknife machine. The lower actuator is a pneumatic cylinder and in principle the air in this cylinder will act as a pneumatic spring until the lever arm pivoted at B hits a machine stop (see Figure 2.47). The following nomenclature is assigned to the analysis:

F_1 = force acting on Flange 1 eventually causing it to shear the 15 mm diameter mounting cap screws;

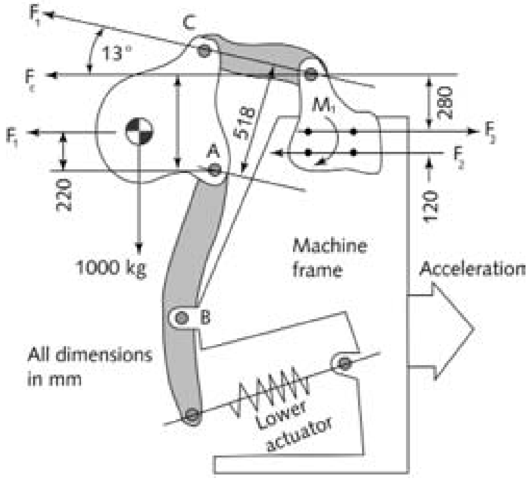


Figure 2.47 Simplified mechanism schematic of the major moving parts of the airknife machine indicating acceleration of the crated machine

F_2 = resisting force provided by bolts during shearing off;

= timp x 2 x A (noting that F_2 is provided by two bolts acting in line), where

t_{imp} = impact shear strength of the 15 mm bolt. From markings on the bolt head this has been estimated to be an SAE (Society Automotive Engineers) grade 3 medium carbon steel bolt with an approximate tensile strength of 700 MPa. The resulting shear strength under impact loading is estimated at 200 MPa;

A = the section area of the bolt material. This is estimated from the thread root diameter of the 15 mm thread (14 mm root diameter) as 1.54×10^{-4} m²;

F_2 = $2 \times 200 \times 10^6 \times 1.54 \times 10^{-4} = 61.6$ kN

The moment provided by the impact shearing of the 4 x 15 mm bolts is

$$F_2 \times 0.12 \text{ Nm} = 7.4 \times 10^3 \text{ Nm}$$

This moment is generated by the inertia force of the airknife assembly. Referring to Figure 2.46, the assembly will commence rotating about pivot C, under the action of the applied rearward acceleration on the packaged machine. This rotation will cause the actuating lever to move out to its outboard position about pivot B, where it will be stopped by the mechanical stops. When the lever comes to a sudden stop, the airknife assembly inertia load will be experienced in the screw jack connecting the pivot C to the actuator mounts. The magnitude of this impact load will depend on how suddenly the actuating lever is stopped. Judging by the sturdy design of this lever, I estimate that when it hits the stops the inertia load of the airknife can experience several times the acceleration of gravity ($1g = 9.81$ m/s^2). At this point the airknife inertia load, F_3 will act about pivot A. This force is generated by the inertia of the airknife assembly when a sudden acceleration a is applied to it by the impact on the machine crate. The moment due to the inertia of the airknife assembly, acting about pivot A, is hypothesised to be of sufficient magnitude to shear the bolts in the support flanges on both ends of the airknife machine frame. The airknife assembly mass was estimated at 1 ton. Hence we get (referring again to Figure 2.47)

$$\begin{aligned} 1000 \times a \times 0.220 &= 2 \times M_1 = 2 \times 7.4 \times 10^3 \\ a &= 67.3 \text{ ms}^{-2} = 6.9g \end{aligned}$$

This is a relatively low level of acceleration due to an impact load. It could well have been produced by the front of the machine (in its packaging) receiving a severe blow. Moreover, the damage at the rear of the machine is also consistent with a rearward acceleration of the machine package. The idler roll is resting substantially forward of its original mounting position and all the indications of its damage suggest that the machine was accelerated suddenly (hit) in a rearward direction. The four bolts holding the machine to its package base seem to be intact. If the machine were dropped on its front, one would expect the rear pair of these bolts to be, at least partially, dislodged from the package base. The only conclusion one can draw from these estimates is that the machine package either hit something during loading or unloading, or a heavy load impacted on it while stationary.

(b) Consequences of the impact on overall machine assembly – Considering the analysis of the events leading to the damage, the following repair scenario was recommended:

1. All bearings and bearing mounts in the main frame side plates must be replaced. They would have taken the brunt of the impact load.
2. All rotating elements of the machine must be checked for run out. There is clear evidence that some rollers may be bent at their mounting into the side frame of the machine.
3. All damaged and broken parts must be replaced with equivalent new components.
4. The airknife assembly gap (refer to Figure 2.42) must be checked for gap size consistency (± 0.025 mm recommended by Holker) and if necessary replaced. The lips of the airknife were packaged separately and they were undamaged.
5. The machine must be disassembled and the frame checked for any misalignments of the various bearing housings and mountings. Due to the sturdy design and construction of this machine it is entirely

possible that the machine frame is intact.

6. The machine stops will need to be readjusted to ensure proper operation. These stops are of a relatively simple and robust design, and they did not appear to be damaged.
7. The sequencing of the programmed logic controller is unlikely to be affected in any way by the damage to the machine. However it should be checked out and put through its programme according to the operating manual, once the machine is reassembled.

Taking the above analysis into consideration, it was estimated that the total damage to the airknife machine was relatively superficial and easily repaired either on site, or at a local repairer's workshop.

(c) The likely risks for the machine operator, working with a repaired machine in a critical position of a continuous production facility – The Holker airknife machine is not some delicate instrument with a complex program of operation. In general its complexity could be compared favourably with simple mining or earth moving machinery. The Holker machine handbook notes:

"Please keep in mind that this coater uses a jet of air travelling at speeds in excess of 200 m/sec. to uniformly meter coating across a 2.9 m wide substrate, which is itself travelling at 8 m/sec. We have observed air stream non-uniformities caused by internal scratches in the airknife as small as .025mm, that are capable of effecting coating uniformity. Our airknife coaters are the product of 35 years of painstaking development. Each piece of the equipment is manufactured to tolerances at either plus or minus 0.127 mm for standard parts, or plus or minus 0.025 mm for critical parts."

The standard tolerance of ±0.127 mm (±0.005 inch) is not particularly accurate and corresponds to the general tolerances given on most engineering parts. This corresponds to general machining tolerance on part sizes up to 3 m in size (refer to *Australian Standard 1654–1974*, *Table 5 – Numerical values of standard tolerance grades 6 to 12*) This type of general tolerance can be easily achieved in most machine shops.[2.19] The tolerance of ±0.025 mm (±0.001 inch) is a fine machining tolerance on part sizes up to 800 mm. These would be again easily achieved by any competent machine shop operator.

The most critical element of the whole airknife metering operation is the setting up of the airknife jet location and attitude. After describing the nature of coating inconsistencies resulting from poor adjustment of the airknife, the Holker manual notes mysteriously *"… adjust the airknife position and gap according to accepted practices …"*.

Clearly, all the information available indicates that the operator has significant influence on the proper operation of the machine and the ultimate

2.19 See also ASME DImensional Standard B4.3-1978; http:// isotc213.ds.dk/standard.htm (Published standards of ISO Technical Commitee 213).

quality of the coating performance. There are in fact two adjustments provided for the operator on the airknife at both sides of the machine (refer Figures 2.43 and 2.44), one to adjust jet angle and the other to adjust the jet location relative to the paper surface. As can be seen in Figure 2.44 these adjustments are robust and the scales of adjustment are marked in relatively coarse increments. From the above analysis of the operation of the machine, it may be concluded that for the Holker machine repaired to its original specifications and properly checked out for mechanical operation the risk of diminished performance below the original manufacturer's guarantees is extremely low.

2.6.4 Lessons Learnt and Outcome

The Holker airknife machine is a relatively simple device requiring considerable skill and care by the machine operator to provide a high quality consistent coating on paper board. The relevant adjustments on the machine are simple and robust. The programme is relatively simple and in any case it has not suffered any damage. The mechanical components suffered serious damage and the machine required to be thoroughly checked out prior to recommissioning.

Given the precautions of checking and replacing irreparable parts, as well as checking out the operation of the assembled machine, there appeared to be no reason why this machine should not be returned to satisfactory service. The repairs and checking required are well within the capacity of any competent machine shop operator, of whom there are several in Melbourne.

In this case the winning line of argument rested on the degree of robustness and level of observed precision in the operation of the airknife machine. The warranty issue was a *"red-herring"* since the delivery and successful commissioning of the coating line (including the airknife coater) were Holker's responsibility. Their contract with Busyboard called for Holker to be responsible for the satisfactory operation of the complete coating facility. In attempting to collect substantial marine insurance and disawowing responsibility for the operation of the repaired airknife, Holker may have been trying to cover themselves against their own possible lack of due diligence in repairing the machine to its new condition. Ultimately the operation of the complete facility rested with Holker. Possibly, they may have been able to claim liquidated damages against the marine insurer for the time delays involved in repairing the airknife. But, in the event, the on site repairs to the machine did not delay the installation of the coating line.

This is a case of identifying the elements of the repair process and assessing the risk of operating the repaired machine based on technical expertise of a suitable engineering machine shop operator. This case was resolved by repairing the machine on site.

2.7 A High-speed Compressor is Damaged by Faulty Bearing Manufacture

2.7.1 The Case Culture and the Defining Event

Air conditioning units are designed to operate unattended for long periods of time. In general, the most robust part of such systems is the compressor unit. This case concerns a modest size (60 horsepower, or approximately 45 kW) industrial air conditioning unit. This unit was supplied to *Hercules Pty. Ltd.* in about 1980 by *Electra Pty. Ltd.* a local importer of industrial air conditioners. The unit was commissioned and installed by Electra, who agreed to provide regular servicing for the unit. The unit ran successfully for nine years, at which time it was stripped down and a bearing replaced. After restarting the unit ran for a few hours and then stopped. This maintenance process was repeated several times, with the machine failing only after a few hours of running time. After about the third failure, Hercules threatened Electra with a suit to cover complete replacement of the unit as well as loss of production resulting from unacceptable working conditions without the air conditioner.

2.7.2 Parties to the Dispute and the Client

Nominally this was a dispute in planning between Hercules and Electra's liability insurers. In the ensuing process of collecting the evidence for this case it was discovered that Electra outsourced the repairs of its air conditioning compressors to an engineering firm *Flybynight Ltd.*. Had the case come to court, it is probable that Flybynight would have been enjoined in the dispute.

My client was Electra's insurer, and I was asked to broadly investigate the failure and provide opinion on the matter.

2.7.3 The Expert's Role and the Investigation

This type of general investigation is often referred to as a *"fishing expedition"* in case the expert can find an appropriate failure scenario and some way in which the insurer might share out their financial loss. My first task then was to collect the failed components and to examine all the maintenance documentation available.

Figure 2.48 is a sectional drawing of the compressor. The shaded component is the high-speed pinion running at approximately 20,000 rpm. This pinion is hollow and carries a smaller diameter shaft inside it. The impeller of the compressor is attached to this inner shaft. The apparently strange design of this impeller drive system was dictated by manufacturing economy and weight saving considerations. The pinion gear cut on the outer shaft is expected to last considerably longer than the simple sliding bearings on the unit. Simple sliding bearings were used in the design to save on space and inertial loads that might be imposed by rolling element bearings. The most critical part of this construction is seen to be the thrust bearing

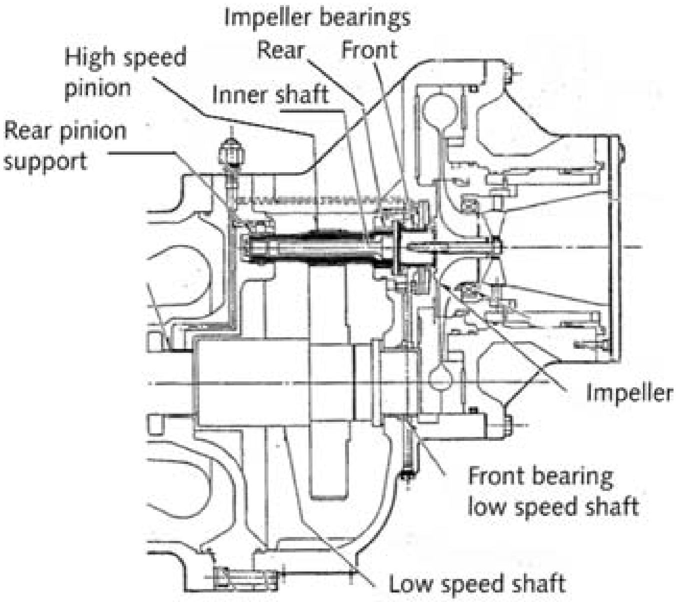


Figure 2.48 Sectional drawing of airconditioning compressor

at the rear of the impeller. Figure 2.49 shows the arrangement of the thrust bearing and the thrust face on the high-speed shaft schematically. Figure 2.50 shows the assembled high speed pinion and journal as well as the inner shaft that carries the journal and impeller.

Just prior to this investigation the compressor unit was found to have seized and the inner shaft of the high-speed pinion fractured in torsion. On inspection, the Babbitt[2.20] alloy bearing shell was found to have shattered (see Figure 2.52) and the thrust face of the shaft was found to be heavily worn (Figure 2.51). In the first major overhaul of the compressor the original inner shaft of the high-speed pinion together with its thrust face and journal was replaced by what was thought to be an equivalent component The original design is seen in Figure 2.53 and one of the replacement designs is seen in Figure 2.51. In the several following maintenance episodes various alternative thrust faces were tried by Electra (Figure 2.54). Apparently they realised that something was grievously wrong in their design of the replacement thrust face.

It is evident from Figure 2.54 that when overhauling the original bearing Electra chose to replace the original hydrostatic bearing with an angled pad design. There was no evidence suggesting the reasoning behind this decision, but it may be conjectured that obsolescence of the unit or inability to get replacement parts could have been responsible for the decision. From

2.20 Soft, white metal, an alloy of tin, lead, copper, and antimony, used to reduce friction in bearings, developed by the US inventor Isaac Babbit in 1839.

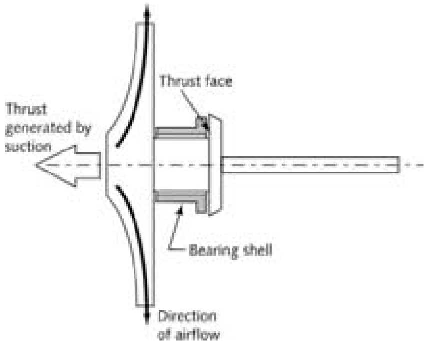


Figure 2.49 Schematic arrangement of the high-speed shaft and bearing

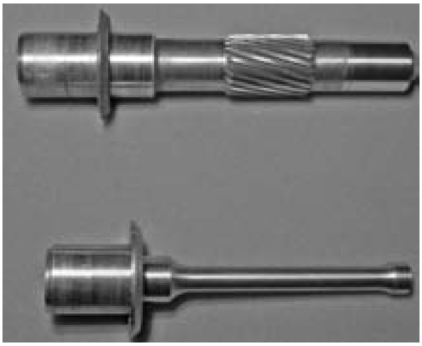


Figure 2.50 High-speed pinion assembly and separated inner shaft

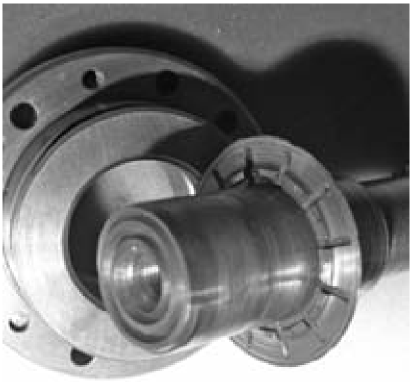


Figure 2.51 Heavily worn thrust face on high-speed journal

Figure 2.52 Components of bearing shell recovered from seized compressor

Figure 2.53 Nominally worn thrust face of high-speed shaft replaced in overhaul

Figure 2.54 Original thrust face compared to various alternatives tried by Electra

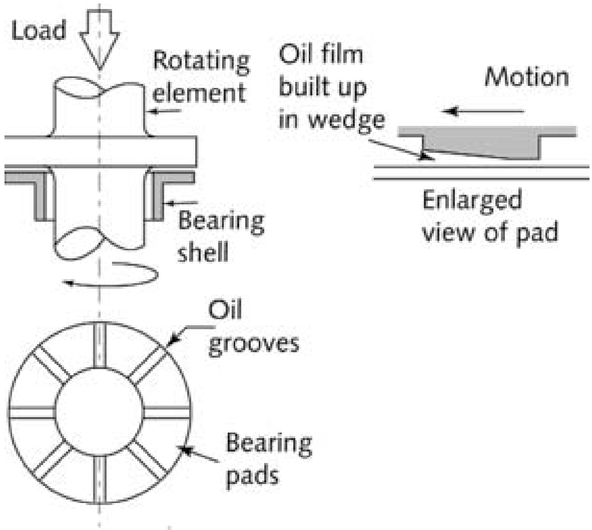


Figure 2.55 Angled pad thrust bearing geometry (schematic only)[2.21]

the several failed attempts at replacing the bearing it is also clear that Electra's design approach was one of trial and error rather than informed analysis. A reference to Neale's handbook on thrust pad design would have enlightened Electra to the real nature of angled pad thrust bearing design (refer to Figure 2.55). Moreover, tables in that publication would have advised them of the proper geometry for the loads and speeds encountered in the compressor. In the event the rather crude attempt at concocting such a bearing resulted in the ultimate seizure and failure of the compressor.

2.7.4 Lessons Learnt and the Outcome

This was not one of those large quantum cases that are fought out by insurance companies, occasionally dramatised by the media. Nevertheless it contained substantial lessons for the engineering litigator. The repair of the air conditioner could have dragged on with substantial manufacturing losses incurred ultimately by insurers of Hercules or Electra. If the air conditioning unit was not easily replaceable, or parts were no longer available, then the almost insignificant cost of contracting an expert designer would have been the simplest solution to the problem. In the event nobody would admit to the root cause of the problem and mediation was sought. The advice given to Electra was that they should use a three-dimensional coordinate measuring machine to record the geometry of the original bearing system and manufacture an exact replica. This geometry functioned satisfactorily for several years originally and there was no reason why it would not continue to do so.

2.21 Neale (1995).

2.8 Two Large Vehicles Roll Over

Two cases and their associated investigations are described here. Although they both involve the rollover of large industrial vehicles, both due to steering failures, neither involves any human injury. They are presented here as cautionary tales to illustrate the importance of care in advising maintenance procedures to mechanics and machine operators when inspecting and repairing vehicles exposed to harsh operating conditions. Both cases concern only the technicians involved in maintaining or repairing the vehicles. The cases were fought out between insurers for these technicians and the insurers of the vehicles damaged. Although neither case involves a dramatically large quantum (both claims were less than AU$ 300,000), both are interesting because they are the results of poor engineering design decisions.

2.8.1 Case Culture (a) – A Flipped Fertiliser Spreader

Modern mechanised farming makes use of computer-controlled spreading of fertilisers due to the cost of fertilisers, as well as the need to apply carefully controlled amounts of farming chemicals for optimum production. In some cases large-scale farming involves satellite thermal imaging to assess where and how much fertilising is needed.

The *Bigbrother Co.* of the United Kingdom make an excellent range of tractors for use in mechanised farming. Figure 2.56 shows a type of tractor manufactured by Bigbrother for the attachments of various farm implements. In the background a typical trailer may be seen in this figure. The procedure used in controlled spreading of fertilisers is to attach such a trailer to the tractor and control spreading by programmed opening of distribution gates on the trailer. *Fillary Engineers* of Coleraine in Australia manufacture and market a range of computer automated fertiliser spreading trailers. Farmers interested in purchasing these types of machines need to apply to Fillary, who will import the Bigbrother tractor and attach the Fillary trailer to it, usually subject to the purchasing farmer's specifications of size and various optional fittings.

2.8.2 The Accident Event and Parties to the Dispute

In 1999 *Old-McDonald Farm* purchased a fertiliser spreader system from Fillary, and then proceeded to use it for a period of about nine months. During this period the Old-McDonald farm had experienced an unusually wet season and the tractor was used in very muddy conditions. It was during a normal fertiliser spreading operation that the driver lost control of the tractor and it flipped over, causing substantial damage and completely destroying the fertiliser trailer. Fortunately the driver was not hurt in this accident.

Initial evaluation of the accident indicated that a mechanical steering failure caused the flipping of the tractor and trailer. Consequently, Old-McDonald made a warranty claim against Fillary, whose liability insurers

Figure 2.56 Typical four-wheel-drive RS232 Bigbrother tractor

sought to distribute their loss and asked for an independent evaluation of the steering failure. Their loss assessor recognised the mechanical failure as a possible design fault with the Bigbrother tractor's steering mechanism.

2.8.3 The Role of the Expert and the Investigation

When the steering linkages of the flipped tractor were examined, it was found that the steering knuckle had been pulled out of the main steering track rod. This track rod is a heavy tubular component that connects the steering arms of the two front wheels on the tractor (see the item marked Z in diagram (b) of Figure 2.57). Three-degree-of-freedom ball joints on the steering knuckle (item marked E in diagram (a) of Figure 2.57) allow for the necessary motion freedoms required for steering the front wheels. The screw adjustment of the steering knuckle permits the camber of the front wheels to be adjusted. Camber is the angle the plane of the wheel relative to the vertical plane and is used to control steering forces.

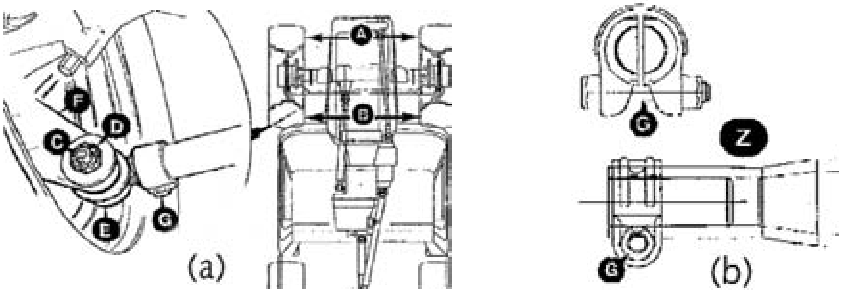


Figure 2.57 Maintenance instruction diagrams for the RS232 tractor

Figure 2.58 Original design of track rod (worn sample from Fillary)

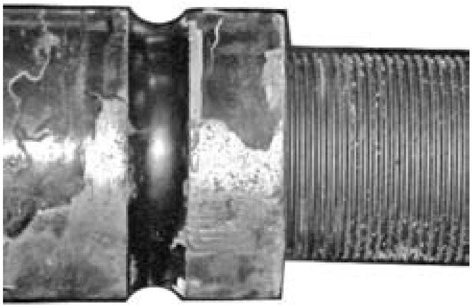


Figure 2.59 Redesigned track rod end as found on flipped tractor

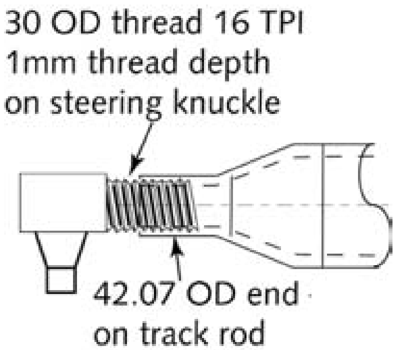


Figure 2.60 Sketch of track rod and knuckle arrangement (schematic only)

Figure 2.61 Worn knuckle (top) and new knuckle (bottom)

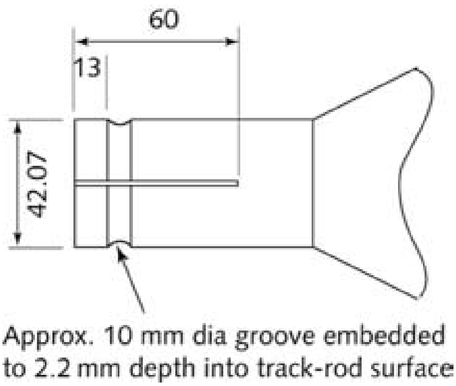


Figure 2.62 Sketch of track rod end as found on flipped tractor

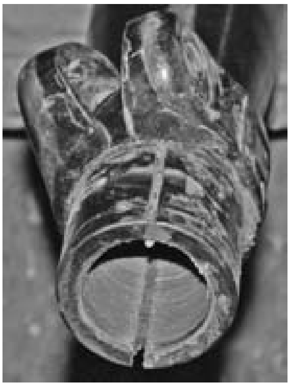


Figure 2.63 Modified track rod design with improved clamping bolt locator

Tractors, as well as other farm implements are regularly redeisgned to meet customer requirements. No doubt, the Bigbrother design team responded to a need for greater transmission power, probably for driving the tractor in heavy mud, when they substantially increased the size of the front differential on the RS232 tractor. This design change called for a redesign of the originally straight track rod, to permit it to pass around the now much larger differential. The new design also made use of substantially heavier tubing with increased wall thickness. To accommodate the original steering knuckles the tubing was swaged[2.22] down to the original smaller diameter tube at the end. However, the wall thickness of the heavier tubing was retained. Figure 2.58 is a photo of the original design for track rod and steering knuckle. This photo is of a worn sample track rod, found in the Fillary scrap box. It was intended for comparison with the new track rod design. Figure 2.59 shows the track rod end taken from the flipped tractor. On inspection, this track rod end had one transverse slot cut in it for clamping the steering knuckle thread and it had a circumferential groove for the clamping bolt. Figure 2.62 is a schematic sketch of this new track rod end.

The advice offered to maintenance staff for adjusting the camber on the front wheels of the tractor is taken from the Bigbrother maintenance handbook for the RS232 tractor (refer to Figure 2.57).

"2 Check Wheel alignment (see Figure 2.57 – diagram (a))

2.1 Ensure that the wheels are pointing straight ahead. Measure the distance ***A*** *between the leading edge of the front wheel rims at hub height. Measure the corresponding distance* ***B*** *at the trailing edge.*

Distance ***A*** *should be less than distance* ***B*** *by 0 to 5 mm. If required, adjust as detailed below.*

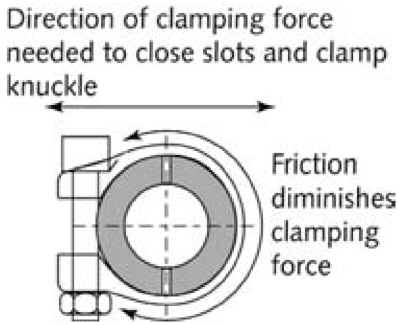


Figure 2.64 Schematic view of tube clamp with the clamping bolt oriented in the least desirable direction

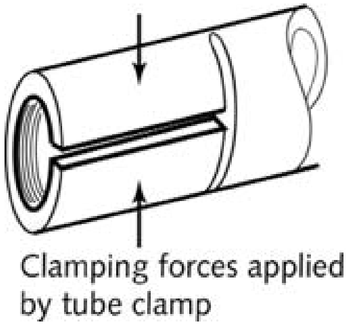


Figure 2.65 Schematic sketch of a pipe clamp designed to permit better grip on the steering knuckle thread

2.22 A mechanical deforming process is a process that is used to reduce or increase the diameter of tubes and/or rods.

2.2 Disconnect the left hand end of the track rod ***E*** *from steering arm* ***F****.*

2.3 Position the wheels so that the wheel alignment is in the middle of the limits specified at step 2.1.

2.4 Screw the left hand end of the track rod in or out as required until the taper rod end ***E*** *aligns with the taper of the steering arm* ***F****.*

2.5 Refit track rod end and firmly tighten nuts ***D*** *and nut/bolt* ***G*** *which must be positioned horizontal as shown at* ***Z*** *and not vertical (see Figure 2.57 – diagram (b)).”*

From Figure 2.59 it is evident that the locking bolt of the tube clamp used on the new track rod end is constrained to be oriented at right angles to the tube axis by the bolt groove. However, the bolt is not constrained to be located at any specific orientation relative to the slot in the tube end (refer to Figure 2.62). It is this slot that should permit the clamping together of the tube end onto the steering knuckle thread. If the clamp bolt is not at right angles to these slots, the clamping force will be partly dissipated in clamp friction, the worst case being when the bolt is parallel to the plane of the slots (refer to Figure 2.64).

In addition, due to the heavier gauge tube wall thickness of the new track rod, the tube clamp required substantially larger clamping force to clamp the tube onto the steering knuckle screw end. Even when modified (as seen in Figure 2.63) with two transverse slots and a more carefully designed clamping bolt locator, the design was inadequate to hold the steering knuckle screw firmly along its full length. The effect of the poor clamping arrangement was that dirt was able to enter the track rod end and lodge in the thread of the stearing knuckle. There was some evidence of dirt embedded in the slots on the track rod end.

The failure of the flipped tractor's steering was attributed to the following failure scenario.

(a) When the steering knuckle was clamped into the heavy track rod tube end, only the few screw threads near the ball joint end of the knuckle were clamped firmly. This meant that the end of the thread distant from the ball joint was permitted some slight movement inside the track rod end.

(b) Abrasive soil and dirt embedded in the clamping slot was fed into the small space between the screw thread and the track rod end. Eventually the motion of the screw and associated abrasion wore away the thread to such an extent that it was capable of being pulled out of the track rod end under the steering loads. Figure 2.61 shows the nature of wear experienced by the steering knuckle thread in this process.

A more appropriate design for clamping the track rod end to the steering knuckle thread is seen in the sketch of Figure 2.65. This design permits clamping of the thread along substantial part of its length and would have avoided the problem experienced with the RS232 tractor. Interestingly, the design shown in Figure 2.65 is well known and used in many engineering applications, though perhaps not in the farm machinery industry.

2.8.4 Case Culture (b) A Tip Truck Tips Over

Tip trucks or *"tippers"* are commonly used in the earth-moving and construction industry. The *Safedrive Company* of Sweden makes a well-known range of trucks, including tippers. *Mr. Soprano*, a hire company proprietor purchased a Safedrive tipper in 1985, for the purpose of hiring it out with a *Bobcat*[2.23] to earth-moving contractors. Approximately thirteen years later in 1998 Soprano decided to replace the tipper with a newer truck. In the event the truck was sold to a *Mr. Baritone*, an earth-moving contractor. As is usual in such cases the sale was accompanied by a roadworthiness inspection. This is a mandatory inspection for all used vehicles in Australia. When completed, and any faults found have been corrected, a roadworthy certificate is issued. Inspections are carried out by authorised mechanics. The mechanic in this case was a motor vehicle repairer operating a business called *Two-Ten Autos*.

2.8.5 The Accident Event and the Client

Approximately two months after the transfer of the tipper from Soprano to Baritone, it was involved in an accident while carrying a full load of earth fill from a building site. In the accident it was rolled over and suffered so much mechanical damage that it had to be completely written off by Baritone's insurance company. Baritone, who was the driver, was also injured in this accident, but this case is only concerned with the issue of the rolled tipper. This issue was fought out between Baritone's and Two-Ten Autos' insurers. When reporting on the accident, the driver claimed that at some point he lost control of the vehicle due to a loss of steering, drove off the sealed road onto the soft shoulder and there the vehicle rolled over. When the steering was inspected it was found that one of the ball joints on the steering arm had been pulled out from its socket. Moreover, it was also

Figure 2.66 A type of tipper produced by Safedrive

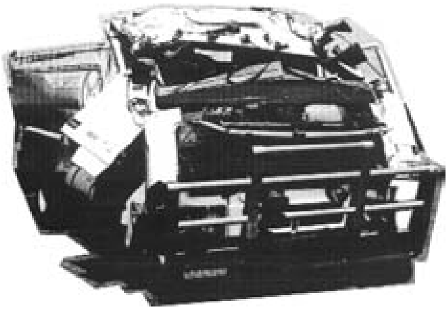


Figure 2.67 The damaged tipper owned and driven by Baritone

2.23 A trade name for a compact skid loader used in earth moving and construction.

found that this ball joint was so badly worn that it could be easily pulled out of its socket. Baritone's insurers claimed that Two-Ten Autos were negligent in the roadworthy inspection and they should have been able to detect the wear in the ball joint.

My initial involvement in this case was through *Dogwood and Dogwood (DD)*, the counsel acting for Two-Ten Autos' liability insurers. DD provided me with the background to the accident and three expert's reports reviewing the matter and offering contradictory opinions about causal links in the accident as well as the culpability of Two-Ten Autos as a result of their issue of the roadworthy certificate on Baritone's tipper. I was asked to offer opinion that might resolve the conflict between these reports.

2.8.6 The Role of the Expert and the Investigation

Figure 2.66 shows a typical tipper and Figure 2.67 is a photo of the damaged Safedrive tipper. Figures 2.68 and 2.69 show the steering linkage used on the tipper. Standard steering mechanisms use a simple Ackerman-type steering linkage based on the kinematics of the four-bar chain.[2.24] This linkage is designed to permit the outer steered wheel in the steering circle to roll further than the inner steered wheel, thereby eliminating tyre slip and wear. Figure 2. 70 shows a schematic sketch of the Ackerman linkage. On right-hand-drive vehicles the input to the steering linkage is provided by the steering link, normally a short link with two ball joint ends (see the link marked A in Figure 2.68). It is this link that failed on the Safedrive tipper, causing the whole steering linkage to become free to move under the steering forces generated by the ground reaction on the tyres. Once the vehicle left the sealed road and moved onto the soft shoulder of the road, the front wheels would dig into the soft soil and turn at some steep angle to the direction of motion. The wheel dug into the dirt and became a pivot about which the vehicle rolled over. Police photographs at the scene showed the

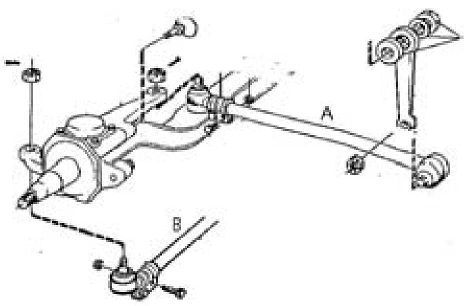


Figure 2.68 A schematic view of the steering linkage of the tipper. The link marked A is the steering linkage and B is the drag link

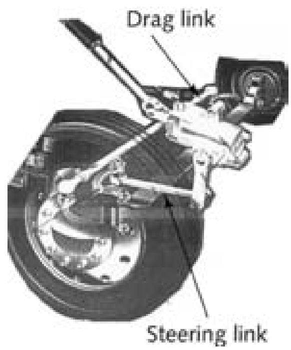


Figure 2.69 Artist's impression of the steering linkage showing both steering and drag links

2.24 See for example Uicker et al. (2003).

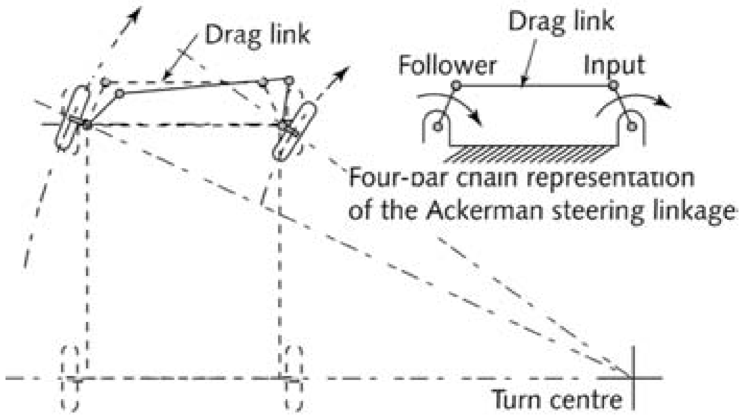


Figure 2.70 Schematic sketch of the Ackerman steering linkage

heavy gouging left in the soft shoulder by the wheels during the accident. In this case there was sufficient evidence to support the above accident scenario. The rolling occurred in a matter of seconds and the driver did not recall the technical detail of the accident chronology, other than losing steering control just before the vehicle rolled.

Figure 2.71 shows the two major parts of the failed ball joint. The socket was found to be completely filled with local clay and had to be cleaned out for full metallurgical examination. Figure 2.72 shows the ball joint at the other end of the failed steering link. Figure 2.73 sows a schematic sketch through a typical ball joint used on Safedrive vehicles. In the investigation of the accident the nylon bushes were not recovered. Figure 2.74 shows some parts of the sealing boot recovered from the failed joint. Sealed ball joints are generally expected to last for many years. If the sealing boot remained intact then it would be reasonable to expect such joints to last the full lifetime of the vehicle. The dispute in this case resolved into the following two issues:

Figure 2.71 The failed ball joint on the steering link. The socket had been cleaned out for this photo

Figure 2.72 The ball joint at the other end of the steering link. Note the damaged grease boot

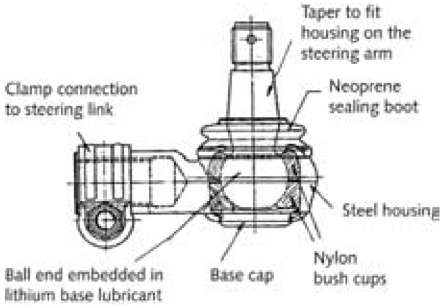


Figure 2.73 Typical ball joint (not to scale) The nylon bushes were missing from the failed joint

Figure 2.74 Sealing boot parts recovered from the failed joint

(a) Was the ball joint faulty at the time of the roadworthy examination by Two-Ten Autos? If this was the case, then could they have detected this fault during the roadworthy inspection?

(b) Did the ball joint fail as a result of some random event in use by Baritone?

In this case I was asked to resolve the above issues based on the reports of three other experts, identified here as *E1*, *E2* and *E3*.

The E1 Report

This report examined the accident site and drew conclusions from gouge marks left on the soft shoulder of the road, from examining the damaged vehicle after the accident and from interviews with the driver Baritone.

The most significant issues in this report were:

1. The allegation that
"Based on Mr. Baritone's statement that he was totally without steering whilst the truck was still completely on the sealed part of the road it is evident that this loss of steering is responsible for the vehicle veering to the left."

2. A figure in the E1 report showed clear marks and measurements of the gauges in the soft shoulder part of the road where it was alleged that the skewed front wheels of Baritone's truck gouged the soft shoulder (these figures were taken from police photographs of the accident scene).

3. Figures of other skid marks in the E1 report allegedly indicated that the wheels were still straight (aligned with the axis of the vehicle) as it left the road.

4. The report went on to estimate the wear on the steering linkage ball joint in the range 0.6 to 1.6 mm on the diameter.

From the above information I estimated that the vehicle may well have been veering off the road, but that the steering may or may not have been fully lost at the point where the vehicle left the sealed road. There is no clear evidence in favour of accepting the loss of steering on the sealed part of the roadway. Moreover, I also estimated that any standard inspection of the steering ball joint would not be able to detect the level of wear indicated in this report.

The E2 Report

This report dealt, albeit briefly, with the dynamics of the overturning process experienced by the vehicle during the accident. I had no disagreement with the factual content of this report. However, this report alleged that

> *"Had the driver lost concentration momentarily, and the vehicle commenced to veer off the sealed road, then there would be some evidence (in tyre track marks on the road) of the driver attempting to recover from this. Since there was no such evidence of attempted recovery, the veering off the road had to be the result of lost steering."*

I asserted that had the vehicle lost steering while on the sealed road, there could not be any evidence of an attempt to recover direction. Moreover, I also noted that there was absolutely no evidence (other than the report of the driver) as to why the vehicle had commenced to leave the sealed road. In fact I suggested that an equally compelling scenario for the accident might well have been as follows:

A steering ball joint is badly worn and requires only a relatively small but finite force to dislodge the ball from its socket;

The vehicle veers off the sealed road and hits the soft shoulder – for whatever reason;

Leaving the sealed road with a heavily laden truck, almost any rut or bump or other surface irregularity can provide the mechanical shock necessary to dislodge the ball from the steering knuckle socket;

The wheels begin to skew heavily and the vehicle rolls.

Since the ball joint required some force to fit the ball into the joint during assembly, it seemed doubtful that the ball simply *"fell out of its socket"* and caused loss of steering. The recollection of the driver was likely to be influenced by many factors. At the speed he was travelling the accident may have taken some seconds. Consequently it may not be clear as to which came first, the loss of steering or the skewing of the front wheels causing the rolling of the vehicle.

The E3 report

This report dealt with the wear on the steering knuckle ball joint and noted that:

> *"… the level of wear observed in the E1 report could well have taken place during the interval between the time the roadworthy certificate was issued and the date of the accident (estimated as 73 days)."*

This report went on:

"The vehicle was approximately 13 years old and had done over 118000 km. If the wear on the ball joint was excessive enough to be a hazard for steering, the tyres of the vehicle would have indicated inappropriate wear and normal service would have alerted the driver to this problem during normal service.

If a nearly 2 mm wear had existed in the steering link ball joint then the movement translated through the mechanism to the steering wheel on the Safedrive tipper would have been approximately 46 mm. A steering wheel requires approximately 30 mm to activate the servo valve on the power steering. Adding these two measurements we see that under these conditions the overall movement required to activate the servo valve and begin turning the wheels would be 76 mm. It is inconceivable that an experienced truck driver or a roadworthy tester could have overlooked such a slack."

Since there was no such warning mentioned in any of the reports, I had to concur with the findings of the E3 report in relation to the responsibility of the roadworthy test two months prior to the accident.

2.8.7 Evaluation of the Available Information

(a) The Flipped Fertiliser Tractor

In this case the failure resulted from both poor design as well as inappropriate maintenance management. The maintenance manual provided by Bigbrother failed to alert the uninitiated mechanic to the dangers presented by the improper clamping bolt location. This was seen as an example of failure to communicate. The design of the tube clamp was seen to be based on faulty reasoning. It neglected to account for the increased wall thickness in the track rod end. Moreover, the design failed to allow for appropriate deflection of the tube end to permit proper gripping of the steering knuckle thread. As noted earlier, Figure 2.65 shows a schematic sketch of a pipe clamp design that would have resulted in a better grip on the steering knuckle thread. Bigbrother design staff should have been aware of such a design and the consequences of the poor clamping action offered by their adopted design. This failure to appreciate the influence of a relatively minor design change on performance was seen as a result of insufficient field testing by Bigbrother.

(b) The Rolled Tipper

1. The steering ball joint was indeed worn as would be expected for a 13-year-old tipper with 118,000 km on its odometer. The wear was not sufficiently excessive to be easily detected by any other than a detailed examination of this component.

2. Normal servicing would have detected undue or inappropriate wear on tyres if the steering ball joint wear was of concern. This condition may have led to a detailed examination of the steering ball joint. Since no such concern had been expressed during the roadworthy test or any subsequent servicing, I suspected that this condition was not of sufficient concern to initiate an investigation.

3.The accident scenario proposed by the E2 report was in some doubt. There was no evidence to support the allegation that the steering was lost prior to the vehicle leaving the sealed surface of the road.

4. The assembly conditions of the steering ball joint would not easily permit the ball to simply fall out of its socket. I accepted that should the neoprene sealing boot be destroyed, the abrasive wear on the nylon bushes in the ball joint would have destroyed it very rapidly. Under such conditions the nylon bush may well fracture and dislodge, permitting the ball to fall out of its socket or be pulled out by a very small force.

5. In the type of use that the vehicle experienced in its life, damage to the sealing boot protecting the grease packing lubricant of the ball joint could be relatively easily sustained. That type of damage would have caused the destruction of the nylon bush very rapidly. Had the rubber boot been worn or damaged at roadworthy inspection time, that would have been easily detected by Two-Ten Autos. Since it was not, I had to agree with the E3 report that the damage, if indeed that was the prior cause of the accident, was sustained during the 73 days following the roadworthy examination.

7. Based on my experience with tip truck and general dirt handling operations in the building industry I considered it entirely reasonable to find wear of the nature experienced in the failed ball joint within the time period of 73 days. The heavy scoring on the surface of the failed ball element (see Figure 2.70) was consistent with very erosive material being incorporated into the joint during tipping operations. This type of heavy localised wear was also consistent with erosive clay soil with embedded small rocks being incorporated into the exposed ball joint.

8. Steering link ball joints are serious safety-critical components in any vehicle. In tip trucks operating in rough ground operations consistent with soil tipping and concreting operations it should be mandatory to carry out regular examinations of such joints. It would seem that Baritone did not carry out such examination in spite of the highly erosive nature of the work to which the Safedrive tipper was exposed.

9. The failed ball joint is oriented in the vertical direction with the sealing boot uppermost (see Figure 2.67 – it was the joint nearer the wheel that slewed and rolled the tipper). Once the boot had been damaged the ball joint would become a convenient receptacle for any dirt landing on its surface. In the conditions of operation to which the Safedrive tipper was exposed it is not surprising that failure of this joint occurred so rapidly. It was my considered opinion that roadworthiness evaluation of a vehicle would be related to operational history of the vehicle. That type of inspection can only evaluate the operational care with which the vehicle was used in the past. It is certainly not able to predict the safety of a vehicle under any future operating conditions. That had to be the ultimate responsibility of the vehicle operator.

2.8.8 Lessons Learnt and Outcomes

The Bigbrother tractor failure case was resolved in an out-of-court settlement. Both parties to that dispute agreed that some faults existed on both sides. Specifically there were serious design problems with the new track rod and its clamp mechanism. Also, there appeared to be a breakdown in communication between Bigbrother and Fillary. However, Fillary should have been aware of the problem with the new track rod design and should have alerted Bigbrother to their difficulties with it. There was some evidence that this type of communication between Bigbrother and Fillary had taken place. Hence the various modifications to the design of the clamping arrangements observed.

By contrast, the Two-Ten Autos case was eventually fought out in the High Court of Victoria in front of a judge and half jury of "six good persons and true". Having one's day in court is a powerful stimulant to anyone feeling a sense of unjust injury. As it transpired, Mr. Baritone was unwisely advised by his lawyers to make such an appearance in court. The case ran for two weeks with barristers and support staff in attendance as well as the other official complement of the court. Experts were called and eventually I was placed in the dock to give evidence in support of my report on this matter. The opposition barrister, *Mr. Alto*, commenced his questioning by almost sycophantically running through my list of qualifications. This is a common gambit used by barristers intending to lull the expert witness into some false sense of security. As a rule this approach usually terminates in what the cross examiner considers to be a *"hard question"* that might dislodge, or partly discredit the expert's opinion. The main point of Mr. Alto's cross-examination may be summarised in the following exchange:

Mr. Alto:

"In your report you state that:
The failed ball joint is oriented in the vertical direction with the sealing boot uppermost. Once the boot is damaged the ball joint becomes a convenient receptacle for any dirt landing on its surface. In the conditions of operation to which the Safedrive tipper was exposed it is not surprising that failure of this joint occurred so rapidly.
Why is it then that we do not see large numbers of Safedrive tippers rolling over?"

AES:

"That is, I presume, because there are not too many thirteen-year-old Safedrive tippers operating in the specially rough conditions experienced by Mr. Baritone's tipper."

This case was won by Two-Ten Autos' insurers and Baritone's insurers were also asked to pay costs. These costs added to the eventual cost of compensation for Baritone should be enough to discourage anyone from such litigation.

2.9 A Large Paper Machine Dryer is Damaged and Discarded Prematurely

Paper making is a process in which wet pulp (a mixture of cellulose and water) is compressed and dried. Drying is achieved by passing the wet blanket of pulp, roughly the consistency of very thick and wet blotting paper, between successive sets of rollers, much like the process used on old fashioned clothes wringers. Each successive set of drying rollers removes some proportion of the moisture from the paper web. In some special cases a very large roller is used to remove the last bit of moisture and to add a glaze to the paper surface in contact with this roller. This special roller is known as a *Yankee dryer* or *machine glaze* roller.

When the paper sheet enters the paper machine dryer section, it contains about 50% water. It must be dried to less than 10% water for a finished product. The dryers are rotating steam-heated cylinders approximately 1.3 to 1.8 m in diameter and slightly longer than the width of the paper sheet. A typical paper machine has 40 to over 100 such steam cylinders, depending on the line speed; the faster the line speed, the longer the drying section. Typical machines are over 100 m long and in excess of 10 m in height. The Yankee dryer is about 4.5 m in diameter and it is expected to remove about 30% of the moisture from the paper. Using a Yankee dryer, a paper machine can be shortened and the drying sections reduced in size. With the appropriate operating parameters, a Yankee dryer can permit substantial increases in paper production.

2.9.1 The Case Culture

Figure 2.75 shows a typical paper machine installation. The *Primrose* paper company operates a number of paper mills in Australia. Their Marigold plant operates two large machines, one of which is equipped with a Yankee dryer section. Marigold No. 1 machine dryer, referred to as *MG#1*, is a large grey cast-iron pressure vessel running at a temperature of about 110°C. This temperature is continuously controlled by the steam pressure inside the vessel. Two *service* rollers press onto the dryer at chosen locations

Figure 2.75 General view of a typical paper machine installation

Figure 2.76 The discarded Yankee dryer stored on a field float at Marigold

on its circumference. The surface pressure between the service roll and the dryer surface *"nips"* the wet paper sheet and wrings out some of the moisture from it. In paper machine parlance the wringer action of the service roll on the dryer surface is referred to as the *nip*. Stringent quality control of the paper sheet thickness demands that the pressure in the nip should be uniform across the width of the paper sheet. The dryer surface is regularly ground to remove any longitudinal (parallel to the dryer axis) irregularities that result from wear on the dryer. Service rolls are covered with a hard polyurethane coating, providing some compliance for the paper passing through the nip. Service roll coatings are also regularly serviced by the *Inca* Rubber Company.

2.9.2 The Accident Event and Initial Evaluation

Sometime late in 1990, the surface of MG#1 was reground to remove some minor surface irregularities and crowning resulting from wear. While this maintenance work was in progress the dryer section was bypassed and the paper machine continued to produce unglazed paper, albeit somewhat slowed by the dryer section having been removed from service. At the same time the service rolls were also recoated by Inca to match the by now pristine surface conditions of MG#1. Shortly after restarting the dryer section, the elastomer coating on one of the newly coated service rolls delaminated, peeling back a large section of its 25-mm-thick coating, all of it passing through the nip. Although the machine was shut down within seconds of this disaster, MG#1 runs at 500 rpm and the peeled back service roll coating was passed through the nip several times during this shutdown.

Careful examination of MG#1 and the whole Yankee dryer section revealed the following damage:

1. MG#1 suffered surface indentations and run out (axial deformation of the surface) to a maximum of 1.4 mm. This measurement was taken by *Tictactoe* Inc., a regular maintenance contractor to *Primrose* Paper. Although this is a small amount, considering the scale of the dryer drum, it is significant in paper making terms where permissible dryer surface errors are measured in microns.

2. Bearings and bearing support structures suffered substantial damage including permanent deformation of the bolts holding the bearing blocks on MG#1.

3. Metallurgical examination of the dryer in or near the vicinity of the surface damage showed no visible cracks or incursions into the dryer surface. Using a penetrant dye (a normal non-destructive testing method for surface cracks[2.25]) the metallurgist, Andrew Maurel, found some porosity and small surface plugs that had been in the vessel since manufacture. MG#1 was manufactured in England in 1949 by Antigua Ltd. as a gravity casting in grey cast iron. Then it was normal practice to

2.25 See for example Stanley et al. (1995); Grandt (2004); Mix (2005).

weld small surface plugs into casting defects before machining the outer surface.

4. Acoustic examination by an expert, *Dennis Mesa*, found that there was substantial sensor response in the damaged vessel when it was pressurised during testing. This was a regular testing procedure used in normal operational maintenance of the vessel by Mesa. On this occasion he found unusually high acoustic emission responses emanating from the damaged region of the vessel. Acoustic emission testing is a specialised process requiring skilled interpretation of results. It is commonly used for locating incipient failure in large pressure vessels.[2.26] In the case of MG#1, incipient failure appeared especially threatening because the vessel was made of a highly brittle material and evidence from other Yankee dryer failures suggested an explosive failure as a distinct possibility.

5. Magnetic particle and penetrant dye testing of the vessel surface found many indications of surface damage. The *Discovery* Corporation, an independent testing authority called in by Primrose Paper to evaluate the damaged vessel, found about 20 such indications on the surface of the damaged vessel, all of the order of less than 5 mm in length. In metallurgical terms, a *crack* has a specific meaning. It has length and depth. An *indication*, on the other hand, signals the possible presence of a crack without actually identifying its depth or significance. *Discovery* reported that none of the indications was aligned with the axis of the vessel.

6. Mesa also used penetrant dye and magnetic particle testing on the vessel surface, albeit by *long-distance* (in subsequent reports it was noted that Mesa was on the phone in the USA while an unidentified technician carried out the tests). Mesa's tests alleged that there was

"at least one substantial indication about 10 mm long parallel to the axis of the vessel, within the area of the surface damage. This indication was identified as Indication A."

7. Yet another independent investigator, *Niblick*, reported that the passage of the hard elastomer sections of the service roll through the nip corresponded to the damage found on the vessel surface.

8. The *Joyfoot* company has a long history of supplying paper machinery to the paper industry. One of Joyfoot's specialists in Yankee dryers, *Edward Holst*, was asked to examine the evidence for damage suffered by the MG#1 vessel. Holst strongly advised Primrose Paper operations staff to discard the vessel and replace it with a new one from Joyfoot.

9. On the evidence and advice offered to them, Primrose Paper operations decided to discard the vessel and replace it with a new dryer vessel. Figure 2.76 shows the discarded vessel at the Marigold plant.

2.26 See for example ASTM (1989); Sachse et al. (1991); Dimentberg et al. (1991).

10. Internal inspection of the vessel found no bulge in material corresponding to the external indentation on the vessel surface. This measurement was taken with a straight edge inside the vessel and a light shining onto the straight edge on the vessel surface. The report noted that there was
"… no visible light through straight edge contact with inside surface of vessel. Straight edge was placed on the surface at a location corresponding to the external deformation of the vessel."

2.9.3 Parties to the Dispute and the Client

Primrose paper replaced the MG#1 allegedly advised by various experts. they sued Inca for the cost of replacement as well as substantial loss in production incurred during the time the Yankee dryer section was idle. Inca's insurers appointed counsel, *Messrs Flat and Hound*, to investigate the loss and the history leading up to it.

My appointment to investigate the case came to me with a somewhat cursory enquiry from briefing counsel in the form of *"have a look at these papers and see what you think"*. I did not find this unusual. In the most complex of cases I am regularly approached by briefing counsel in this way. I will elaborate on the reasoning behind this cursory approach later. For now, let me say that my initial response to the Primrose Paper MG#1 dryer accident may be summarised in my assertion to counsel that *"this dryer vessel is built like a brick dunny"!* This response was partly a result of my initial *"back-of-the-envelope"* calculation of shell strength and a brief survey of the documentation provided to me by counsel. Using back-of-the-envelope calculations to appraise the investigation-worthiness of a case is a useful ploy for weeding out cases eminently unworthy of further investigation. On initial reading of the voluminous literature provided to me I formed the opinion that there appeared a case to be answered by Primrose and their advisers. This opinion was supported by the apparently hasty action by Primrose to replace the MG#1 with a new one. Moreover, the specifications of the new MG#1 showed it to be a substantially improved dryer over the discarded one.

- Primrose appeared to have a legitimate claim against Inca, on the grounds that they caused the accident in the first place.
- Inca's insurers, on the other hand, were entitled to a full disclosure of how the decision to replace the old MG#1 with what appeared to be a much better new one.
- Ultimately the various advisers to Primrose would be also drawn into the dispute.

2.9.4 The Role of the Expert and The Investigation

Primrose Paper ordered a new vessel from Joyfoot and had it installed in 2002 at the Marigold Plant. The replacement cost of the vessel was estimated at AU$ 5 million. In addition, the loss of production for Primrose while the Yankee dryer was out of action was estimated at a further AU$ 5 million. This sum of AU$10 million was the approximate claim faced by the insurers of Inca Rubber for the delamination of the failed service roller.

My brief was to advise and offer opinion on the following matters:

(a) Engineering protocols followed by a responsible paper product manufacturer and paper machine operator in the position of Primrose Paper, in evaluating the damage to MG#1.

(b) Whether the opportunities for repairing the MG#1 as an alternative to replacing it was a serious option for consideration by a responsible paper machine operator in the position of Primrose Paper.

(c) Whether Primrose Paper have followed protocols appropriate to a responsible paper machine operator in their position in their investigations of the opportunities for repairs to the MG#1 rather than choosing to replace it.

The voluminous reports supplied to me demanded that I draw up a simplified case chronology. In addition to the immediate events preceding and subsequent to the accident event, Primrose documentation showed evidence of a substantial history of concern with MG#1. This historical material is also included in the case chronology.

Case Chronology

The sequence of events leading up to and immediately subsequent to the accident event in which the Yankee dryer MG#1, of number 1 paper machine at Primrose Paper's Marigold plant was damaged is shown inTable 2.4.

Background History of MG#1

The original vessel was manufactured by Antigua Ltd. in England in 1949. Various documents relating to its manufacture identify the material of the shell as a "Nickel Grey Cast Iron" with the following properties :

- Outside diameter = 14 ft; Inside diameter = 13 ft 8 inch
- Face width = 200 inch
- Ultimate tensile strength = 23.5 tsi (using UK tons = 363 MPa)
- Modulus of elasticity E = 18.1 x 10^6 psi (125 GPa)
- Coefficient of thermal expansion α = 6 x 10^{-6} / °F
- Poisson's ratio μ = 0.26
- Temperature difference from inside to outside shell surface during normal operation at 500 m/min surface speed = 90°F
- Approximate weight of structure = 80 tons

Regulatory and statutory background

The original design of the vessel was according to the standards operating in 1949, namely the SAA[2.27] Boiler Code. The requirement for unfired grey

2.27 Standards Association of Australia.

Table 2.4 Event chronology

Event number	Date	Event description[2.28]
1	1949	MG#1 manufactured by Antigua Ltd in England
2	1960	Primrose operations concerned about throughput of No. 1 paper machine and an investigation is launched to discover if MG#1 pressure may be increased to 60 psi (414 kPa). Antigua design specification was for 50 psi (345 kPa) tested hydraulically at 100 psi (690 kPa). Makers, Antigua and DLI[2.28] agree to this, but recommend hydrostatic testing to 120 psi (828 kPa) and maintaining centreline shell thickness at 2inch (50.08 mm) Thickness established by acoustic testing and is found to be 2.14 inch (54.4 mm) Hydrostatic testing declined by Primrose operations.
3	1970	MG#1 reground to correct crowning errors using abrasive belts on the service roll. Approximately 0.035 inch on diameter removed (i.e. 0.5 mm on radius)
4	1980	More concern about the possibility of increasing MG#1 pressure to 60 psi. DM of University of Melbourne contracted to carry out fatigue degrading investigation of vessel due to service roll nip pressures (500 pounds per lineal inch – pli). This investigation finds MG#1 to be sound in this regard and safe to operate. Again, due to requirements of hydrostatic testing to formally approve operations at the increased pressure MG#1 remains operated at 50 psi.
5	1999	Primrose operations concerned about Yankee dryer losses incurred due to in-service failure of such dryers. Data from Arkwright International provides some information. No lives lost, but some machine damage and downtime.
6	9/99	MG#1 profile measurements suggesting a regrind is needed.
7	10/99	MG#1 acoustic emission tested using only four sensors. Found to be sound.
8	3/01	MG#1 ground to correct profile. Again approximately 0.5 mm removed on shell thickness.
9	3/01	Newly coated service roll is installed on No. 1 machine to provide nip on MG#1.
10	4/01	Service roll delaminates and allegedly irreparably damages MG#1.
11	4/01	Post-accident acoustic emission test performed and results suggest incipient failure in shell near alleged damage imparted by service roll delamination.

2.28 A guide to units of measurement is provided in appendix A3.

2.29 DLI, Department of Labour and Industry, was in 1960 the Australian Government authority for the acceptance testing of pressure vessels.

Table 2.4 Event chronology (continued)

Event number	Date	Event description
12	4/01	*Metallurgical examination of shell surface by Maurel. Some surface porosities found, all relating to repairs on the original shell due to casting imperfections only. In its conclusion Maurel's report states that:* *"The cylinder and plug microstructures were found to be in good condition, with no evidence of microstructural degradation being identified.* *The cracking defects were found to be entirely casting related, with no indications consistent with in-service crack propagation being identified.* *Examination of regions remote from the crack revealed the presence of other linear crack-like indications (albeit on a microstructural scale). It was considered that indications of this type may well be found to be present over much of the surface of the cylinder.* *..., it follows that it may well prove unreasonable to condemn the cylinder as a consequence of primary crack indications."*
13	4/01	*Magnetic particle and radiographic examination of shell reveals no cracks, just some indentations. Surface examination also reveals the depth of these indentations to be, at worst 1 mm. The total out of roundness allegedly attributed to the accident event is no greater than 1.4 mm (Tictactoe report).*
14	4/01	*Joyfoot prepares quotation for replacement MG #1. The test pressure is specified for the proposed new dryer is 148 psi, corresponding to an operating pressure of 74 psi (510 kPa).*
15	4/01	*Letter from Ed. Holst of Joyfoot condemns damaged MG#1 based on acoustic emission results as well as Mesa's identification of crack-like indications and especially Indication A.*
16	7/01	*Niblick report identifies service roll delamination as the main culprit for damage to MG#1.*

cast-iron vessels was that the shell be designed for dominant, "maximum" stress only. This rule is in line with the normal failure criterion for brittle materials of which grey cast iron is one. The maximum stress operating in MG#1 was circumferential stress (or *hoop* stress) pertaining to the stress due to internal pressure. This rule neglects other stresses such as thermal stresses due to moderate temperature differences across the vessel shell and centrifugal loading due to modest rotational speeds. Consequently, the factor of safety implicit in the permissible working stress was 10. It was also a requirement that the vessel be hydrostatically tested at twice the operating pressure. In the case of MG#1 this was at 100 psi (699 kPa). The vessel was tested at 100 psi hydrostatic pressure at Antigua's works in November 1949.

The current standard for pressure vessel design is SAA 1210-1997 and this standard also defines a safe working stress for low-temperature unfired grey cast-iron vessels, so that the implicit factor of safety is 10 (SAA 1210-1997, Australian pressure vessel code for cast iron 40 (40 MPa – Table 3.3.1(C) page 69). The current authority for the operation of low-temperature unfired pressure vessels is the Licensing Branch of the Victorian Work Cover Authority. They require the vessel to be inspected and reported on regarding safe operating conditions every three years. An inspection certificate must be filled out and copies sent to the Authority for registration. The two certificates available from this inspection identify the vessel as having an operating pressure of 0.414 MPa (60 psi).

Service History and Operation-Related Issues

The vessel was ground in 1975 and also in 2001, removing about 0.5 mm from its shell centre line thickness on each occasion. Due to concerns about the possibility of operating the vessel at 60 psi, to increase its performance, the thickness was monitored by acoustic means at regular intervals. These readings show the wall thickness varying over a modest range, the lowest value recorded as 47 mm.

2.9.5 Evaluation of Information by the Expert

Shell stresses and operating factors of safety[2.30]

Operating working stress at the time of design and manufacture of MG#1 was 3200 psi (22.4 MPa). In light of the actual tensile strength of the shell material this working stress represents a factor of safety of 363/22.4 = 16.2. This is a considerably in excess of the SAA Boiler Code operating factor of safety of 10 for unfired pressure vessels in grey cast iron. At the time of design of this vessel in 1949 the only stress evaluation used for shell thickness specification was hoop stress. Since this neglects any centrifugal loads and any thermal loads, the larger factor used by the makers suggests some suitable degree of conservative design. Moreover, the extra thickness in the shell due to the conservative factor may also account for the nip load imposed by the service roll during normal operation. An alternative view might be (without any formal evidence) that the makers in early discussion with Primrose Paper may have considered using the vessel in the upgraded mode with 60 psi internal pressure. In subsequent correspondence Antigua seemed quite comfortable about operating the vessel at the higher pressure.

In correspondence relating to upgraded pressure loads Antigua had advised the need to maintain central shell thickness at 2 inch There were several acoustic thickness surveys conducted on the shell throughout its service life, with considerable variation in thickness as measured by the acoustic surveys, the lowest value being 47 mm, or significantly less than the suggested 2 inch for safe operation at 60 psi. Since the original design

2.30 Formal calculations for this case are presented in Appendix 2.

had a wall thickness of of 2 inch (50.8 mm) some doubt must be cast on the veracity of the acoustic thickness measurements. In spite of these uncertainties, Primrose operating staff were prepared to operate the vessel at the higher pressure of 60 psi. Ultimately the documents and correspondence suggest that the only reason for abandoning this pressure upgrading of the vessel was due to the cost and risk involved in hydrostatic testing in situ.

I calculated the operating stresses prevailing in the vessel just prior to the time of the accident event (refer to Appendix 2). I estimated the stresses in the shell to be 18 MPa (assuming the thickness is 50 mm); this now represents a factor of safety of approximately 20 over the actual tensile strength of the shell material. It is also well below the tensile working stress permitted in the original design code (22 MPa) or even the current SAA 1210 Australian pressure vessel code for cast iron 40 (40 MPa – Table 3.3.1 (C) page 69 of the relevant standard). This simplified calculation and the above correspondence and documented interest of Primrose to operate the vessel at increased pressures might suggest a degree of opportunistic haste in choosing to discard the allegedly irreparably damaged MG#1.

Non-destructive Testing and Evaluation of the Vessel Immediately Subsequent to the Accident

Acoustic emission (AE) testing subsequent to the accident event showed that the vessel had some grain boundary movement initiated since the previous test in 1999. These results are fairly common in grey cast iron and particularly so when the structure has suffered some local deformation. All information relating to AE testing suggests that while it is certainly an indicator of internal structural events, it is by no means a reliable indicator of the scale without other forms of non-destructive testing. Neither the magnetic particle inspection nor the radiographic tests subsequent to the accident event showed up any cracks in the shell to support the *AE* test concerns expressed by Mesa. Moreover, the evidence from the Primrose visual inspection of the internal surface of the vessel found no bulges or deformation corresponding to the location of the external damage. Because the vessel is a thin-walled pressure vessel (defined as having shell thickness smaller than one tenth diameter – in this case $t = D/85$), this finding is inconsistent with any substantial damage to the shell.[2.31]

AE tests at best identify the location of possible incipient failure or material defects. However, they need to be supported by other forms of examination, in particular ultrasonic tests, to establish the scale of the damage if any. Since this was not performed in the case of the damage to the shell of MG#1 it is most imprudent to assign any significance to *AE* results alone.

Since the internal surface of the shell showed no deformation (bulge under the external indentations) it is most likely that the shell had suffered superficial surface damage only.

2.31 See for example Timoshenko and Woinowski-Krieger (1959); Timoshenko and Goodier (1983).

Alternative Courses of Action Open to Primrose Subsequent to the Accident

There were several additional courses of action which Primrose should have taken to investigate the consequences of the damage caused by the accident event in order to decide whether it was appropriate to replace or repair. A responsible paper machine operator in the position of Primrose would be expected to follow the following courses of investigations:

(a) Ultrasonic evaluation of the three-dimensional structure of the damaged area to determine the scale of the grain boundary movements identified by the AE tests. These types of tests are commonplace in the pressure vessel industry for the examination of large-scale porosities in welds.

(b) Grinding out the local indentations and re-testing with AE and other non-destructive testing methods.

The option of repairs to the vessel was very much on the agenda of Primrose operations staff. This intent is evident in the correspondence provided to me in the documentation. All of these correspondence documents deal with various repair options. The only document that makes direct reference to MG#1 being *"beyond economic repair"* is the one from Ed Holst of Joyfoot, noted in the event chronology above. Based on the scant information available about the health of the vessel at the time this opinion was offered, one cannot disregard the self-serving nature of this opinion.

Further Investigations and Mediation with Other Experts

Following the initial investigation of the soundness of the allegedly damaged MG#1, several follow-up investigations were launched by both Inca Rubber's and Primrose's insurers. Dennis Mesa performed further AE tests under pressure on site at Marigold. He also performed magnetic particle and penetrant dye tests. In his report of these tests he alleged that there were two *crack-like* indications, identified as indication A, approximately 10 mm in length and indication B, approximately 20 mm in length. It was also alleged in Mesa's second report that indication B was actually present in the tests performed shortly after the accident and that it has grown in time from 10 mm to its current size. Both indications were allegedly aligned with the axis of the vessel.

I was asked to help carry out non-destructive tests on the surface of the discarded vessel as a means of validating Mesa's findings. These validation tests were performed by a local NATA[2.32] approved testing authority, *ATTAR*. They were asked to find any surface indications on the damaged surface of the vessel. None could be found other than those reported by the metallurgist Maurel and the Discovery Corporation. The disputing parties agreed that a joint examination of the vessel should occur in the presence of Mesa and ATTAR.

2.32 National Association of Testing Authorities.

This joint investigation took place in 2005 and it too failed to find the indications A and B allegedly found by Mesa. Further evidence of opportunistic replacement of MG#1 by Primrose came from the operational records of the paper maker. It was found that the new MG#1 was not only capable of operating at substantially higher pressures and temperatures than the discarded vessel, but that it was actually being used in this enhanced operational mode. This new production schedule allowed Primrose operations to remove half of the steam dryers on machine No. 1 from service and use the new Yankee dryer in their place. In addition the throughput on machine No. 1 had been increased by about 10% due to the new dryer, yielding a healthy increase in profitability for Primrose Paper.

2.9.6 Lessons Learnt and the Outcome

It is worthwhile to review and elucidate on the rather casual approach taken by counsel in appointing me to investigate this matter (see Section 2.9.3). I should note that, in my experience, briefing counsels are, in general, very conservative people. Years of litigation experience has taught many of them to consider the opportunities of a case not so much from the optimistic view of maximising gains for their clients, but from a more pragmatic consideration of minimising the losses incurred. This approach is broadly based on my earlier assertion that, in a protracted litigation, much in the same way as in a war, neither side can hope to sustain a win, but each may minimise their losses. Technical experts, on the other hand, may see mainly the overwhelming value of their finely tuned technical argument for their side of the litigation. This narrow view often fails to see the bigger picture.

There are many mitigating factors, other than technical issues, that might intrude into a judgement in court. Judges, in general, favour the injured party, even when they may have exploited their injury in some seemingly opportunistic way. In the Primrose and Inca litigation, there was no doubt that Inca had injured Primrose by supplying a poorly coated service roll. As well, Primrose was duly advised, by the best technical advisors available to them at the time, that the damaged MG#1 should be replaced. The mitigating factors in awarding damages here could be seen to be the following:

In Favour of Primrose

(a) MG#1 was damaged by the improperly coated service roll delamination. For this damage Inca was clearly responsible. There were production losses, damage investigation costs and substantial engineering costs involved in possible repairs or replacement.

(b) There was considerable uncertainty about the nature and extent of the damage incurred by MG#1. In addition there was evidence from other failures in other plants of considerable risk from an explosive failure of grey cast iron vessels when operating under pressure. One can appreciate the conservative view that the risk of returning a repaired MG#1 to service may involve some operational risk.

(c) There would be operating losses incurred even if the damaged MG#1 could be repaired.

In Favour of Inca

(a) Failure by Primrose to seek additional advice to confirm and support Mesa's AE tests and his allegations about the existence of some significant cracks in the vessel shell.

(b) Trustingly accepting the undeniably self-serving advice of Ed Holst from Joyfoot, that the allegedly irreparably damaged MG#1 vessel should be discarded, bearing in mind that they (Joyfoot) would be the providers of the replacement vessel.

(c) Most significantly, disregarding the findings of the metallurgist Andrew Maurel about the apparently superficial damage to the MG#1 vessel.

(d) Disregarding the lack of a bulge on the inner surface of the vessel, corresponding to the damaged outer shell depression.

The level of any award against Inca would need to be adjusted for the improved operating features of the new MG#1. It is an accepted rule of insurance that one should not be able to gain profit from a loss. Insurance will not replace a written-off used tricycle with a new Lear jet. Operating records showed that the new MG#1 was being operated at substantially greater throughput rates than was available with the discarded dryer.

From the above discourse it is clear that, if a winning line in this dispute were to be found, it would not be based entirely on technical matters relating to the soundness or otherwise of the damaged MG#1. Consequently, technical investigations of the soundness and repair opportunities available for the discarded vessel were, at best, likely to result in diminishing returns for the defence. An offer of compensation was made to Primrose at an early stage in the dispute. This offer was rejected. Eventually the case went to mediation and a further, more attractive offer was made by Inca's insurers. This second offer was also rejected by Primrose's insurers on the advice of their counsel. As a matter of procedure, when a mediation offer is refused, and the case is pursued to court, should the court's award be less than the mediation offer, costs of proceedings are awarded against the plaintiff (in this case Primrose). Naturally this procedure is meant to discourage vexatious litigation. In an unprecedented legal move, Mesa and Ed Holst, for Joyfoot, were enjoined in a counter-suit by Inca's counsel, for providing inappropriate and incorrect information to Primrose about the soundness and possible repair of MG#1. The vessel was discarded on the basis of this inappropriate advice.

2.9.7 Yet More "Legal Foot Stamping" in This Case

When faced with the unassailable fact that a case is unwinnable, litigators and their expert advisors have been known to exhibit kindergarden-style behaviour of children arguing some point of difference. The procedure involves virtual foot stamping and making respective statements such as

" … my facts show your case to be failing, you are in the wrong …"

" … no I'm not …"; *"… yes you are …"* etc.

All of this is usually accompanied by more and more detailed reports and the invocation of more and more experts. This case of *Primrose v. Inca* had evolved over a period of 3 years into just this style of virtual foot stamping. The following steps in this process demonstrate this evolution:

(a) In 2001 Primrose's vessel was accidentally *"bumped"* as a result of Inca's inadequately surfaced service roll. Metallurgical examination immediately following the accident suggested that repairs were possible.

(b) Several *"experts"* were called in to examine the vessel and based on their advice, despite the advice of the metallurgist, the vessel was condemned and replaced.

(c) In 2003 Engineering Investigations & Associates (EI&A, the author's consulting company) do some back-of-the-envelope calculations and estimate that the vessel was heavily overdesigned (*"built like a brick dunny"*). In my opinion the vessel could indeed have been repaired and returned to service at a fraction of the cost of replacement. Attention was drawn to the fact that there did not appear to be any serious shell deformation.

(d) In 2004 Primrose's experts retort with a deconstruction of EI&A's estimates and spend substantial sums in mapping the inside of the vessel to find the *"bump"* that would signal serious shell deformation.

(e) In 2005 experts of both sides meet at the Marigold plant to locate the *"cracks"* in the shell that made Primrose's experts condemn the vessel. No cracks are found and the experts withdraw to write more reports.

(f) In late 2005 Primrose's main expert pronounces that *"… even if the cracks are not present, there would be substantial and unquantifiable residual stresses imposed on the shell of the vessel due to its substantial deformation."*

(g) In 2006 EI&A contracted a NATA-approved testing authority to carry out a mapping of the inside of the vessel as well as to measure residual stresses in the shell.[2.33] These tests show that surface variations inside the vessel are substantially greater in the undamaged parts of the shell. Moreover, surface variations on the internal surface of the shell are

2.33 Residual stress measurement is a regular feature of damage assessment. The method uses a strain rosette mounted on he damaged surface, followed by staged drilling through the centre of the rosette, while strains are monitored. See for example ASTM E837 Standard Test Method.

well within generally accepted machining tolerances. Residual stress measurements showed that the shell has only highly localised residual stresses in the outer 0.5 mm of the surface. Moreover, these stresses abate to insignificant values below that level. Specialised heat-treating authorities advise that even these relatively insignificant stresses may be relieved by on-site heat treatment.

(h) Primrose's experts request all data to be delivered to them in raw form. I had no doubt that this data would be used for generating further reports and more virtual foot stamping.

At the time of writing this case continues.

2.10 Chapter Summary

Eight examples of industrial accidents involving machinery failures were presented in this chapter. In reviewing the investigative threads common to all the cases presented perhaps the most compelling item was that each had an easily identifiable line of defence (*the winning line*). In these cases the expert is asked to respond to issues raised by counsel, who already had the benefit of having read through the documentation and case history. These issues raised by counsel would reflect their own well-considered assessments (their hunches) of where the weakness in the arguments of the other side might lie. The expert's role was then to investigate and wherever possible reinforce the hunches of the briefing counsel. Of course the expert must evaluate the evidence from an investigation in the harsh light of all factors, not just those that might support the client's case. These points were clearly drawn in the presentation of the cases in this chapter.

Another common experience with all the cases presented was the often confused collection of information provided to the expert. Some helpful tools for sifting through this information have been presented here. The construction of case chronologies provided essential background to the more complex cases presented. The preparation of an FMEA table helped to identify the most likely mode of failure in some cases where there were several alternative probable sources of failure. The construction of a hypothetical failure scenario helped in focusing attention on specific technical elements of these cases.