Supreme Court of Victoria

Timelink Pacific Pty Ltd v Major Engineering Pty Ltd [2006] VSC 288 (4 August 2006)

Last Updated: 4 August 2006

IN THE SUPREME COURT OF VICTORIA	Not Restricted

AT MELBOURNE

COMMERCIAL AND EQUITY DIVISION

ADMIRALTY LIST

No. 4122 of 2005

TIMELINK PACIFIC PTY LTD — Plaintiff

(ACN 063 714 303)

V

MAJOR ENGINEERING PTY LTD — Defendant

(ACN 005 432 397)

JUDGE:	BYRNE J
WHERE HELD:	Melbourne
DATE OF HEARING:	10-14, 17-21 July 2006
DATE OF JUDGMENT:	4 August 2006
CASE MAY BE CITED AS:	Timelink Pacific Pty Ltd v Major Engineering Pty Ltd
MEDIUM NEUTRAL CITATION:	[2006] VSC 288

SALE OF GOODS – sale of hydraulic cylinders for racing yacht – whether a term that cylinders capable of withstanding a force in compression of 262 kN – whether breach of term – whether breach caused damaged to yacht.

APPEARANCES:	Counsel	Solicitors
For the Plaintiff	Mr G H Golvan QC and Ms My Anh Tran	Vadarlis & Associates
For the Defendant	Mr A Herskope and Mr Daniel Clough	Kalus Kenny

HIS HONOUR:

1 At approximately 1.15am on 28 December 2004 the racing sloop *Skandia* was engaged in the Sydney to Hobart yacht race when it suffered an incident which caused it to withdraw from the race.

2 At the time of the incident *Skandia* was on a port tack heading in a south westerly direction in the direction of the Tasmanian coast. It was situated some 70 nautical miles east on the Eddystone Lighthouse at the north east tip of Tasmania. The wind was then at 30 to 40 knots from a little west of south and the prevailing swell from the port bow, that is from the south. The depth of the water was about 3,000 m. The evidence of the navigator, William Grieve Oxley, was that the waves were 4-6 metres high and that the yacht was following a very conservative sail plan. He said the conditions were not extreme; they were common in that part of the world.

3 What then happened was this. The yacht came off a wave which was larger than those then prevailing and the two pistons which held the keel in place failed and broke off. This meant that the keel head, being unrestrained laterally, swung within the hull and damaged various surrounding components notwithstanding the efforts of the crew to lash it down. The vessel was abandoned when it appeared that the yacht might capsize when the keel came away from the hull altogether and the crew were rescued. Their apprehensions were fulfilled; the keel became detached, the yacht capsized and it suffered considerable damage.

4 In this proceeding the plaintiff, Timelink Pacific Pty Ltd, as owner of the yacht, seeks damages from the defendant, Major Engineering Pty Ltd which supplied the pistons. This trial and this judgment is concerned with all questions in issue other than quantum.

Background

5 The circumstances which give rise to the claim are, to a large extent, not in dispute. In 2001 Grant Russell Vincent Wharington, an experienced and enthusiastic ocean racing yachtsman and a director of Timelink, decided to build a new super-maxi yacht with the intention of sailing it in the Sydney to Hobart yacht race. The proposed yacht was to be named *Wild Thing*, but this was changed to *Skandia* when a company of that name became its sponsor. Mr Wharington engaged for the purpose a local yacht designer and design engineer, Donald Allen Jones, to assist with the engineering aspects. Soon

after, in 2002, he engaged Mr Jones to undertake all aspects of the design of the new yacht and this was his role at all material times.

6 Mr Wharington’s instruction to Mr Jones was to design a "cutting edge" yacht that would be capable of winning line honours in the Sydney to Hobart race and of competing in other major ocean yacht races around the world. The designer had therefore, to satisfy the requirements of the Cruising Yacht Club of Australia for the Sydney to Hobart race and, generally, those applicable for ocean sailing elsewhere. These included the American Bureau of Shipping Guide for Building and Classing Offshore Racing Yachts 1994 ("ABS Guide").

7 It was decided that the yacht should have a canting keel. This was a fairly innovative design feature in Australia at that time and had first appeared in yacht design generally in or about the mid 1990s. This was a keel which was fixed to the bottom of the hull on a pin running fore and aft, so that the bulb at the bottom of the keel could be swung laterally. This was achieved in this case by running the keel through the hull into the yacht and applying force to the side of the keel head so as to move it sideways to the desired angle and holding it there. The pin then acted as a fulcrum with, on the underside, about 14,000 kg of ballast in the bulb whose centre was about 4,000 mm from the pin. The length of the keel head above the fulcrum was about 1,400 mm. The forces acting at the top of the keel head, therefore, were very considerable. For the purposes of moving and restraining the keel head, the designer selected a pair of hydraulic pistons which were fixed in parallel to the head of the keel at one end and to the hull on the port side at the other. These are the cylinders which failed.

8 At the time of the incident, the yacht was on a port tack. This meant that the keel was canted to port and the keel head to starboard, held in place by the two extended pistons. What then happened was that, when the yacht came off the large wave, the forces applied to the keel were transferred to the keel head and then to the pistons which were in compression. Both pistons failed; the piston shafts buckled and snapped off, leaving the keel head unrestrained.

9 Returning to the design process, Mr Jones said that in the course of his design work, he made calculations of the forces which might be expected at the keel head when the keel was canted to its maximum of 19.5deg. in either direction. Unfortunately these no longer exist. Nor has he replicated them after the incident for the purposes of assessing the loads on the cylinders which caused them to fail.

10 In response to my direction of 31 May 2006 that the plaintiff file Mr Jones’ calculations made in or about 2002 of the applied forces that would be expected on the keel and on the hydraulic cylinders when fully extended to 19.5deg., Mr Jones submitted a calculation, but it was not the one which he now says he then made.

11 Mr Jones said that he also undertook the calculation of lateral forces on the keel in the circumstances required by the ABS Guide. These circumstances are that the whole keel, including the bulb, when canted to its maximum is horizontal and out of the water. These forces, as at the keel head, he calculated to be a maximum compression force of 43,529 kg (427 kN; 95,947 lb)[1] when the keel was canted 19.5deg. to port, and a maximum tension force of 37,514 kg (368 kN; 86,690 lb) when the keel was canted 19.5deg. to starboard. Since these forces are applied to a pair of hydraulic cylinders, the load on each is respectively 213.5 kN (compression) and 184 kN (tension).[2] This is the calculation which was provided in response to my direction.

12 Some five years earlier, in 1997, Mr Jones had had dealings with Reginald Petty, the sales manager of Major Engineering’s Hydraulic Division which traded as Major Sales and Service ("MSS"). This

was with respect to the provision of a retractable hydraulic propeller drive for Mr Jones' ocean racing yacht *Cadibarra 7*. Mr Jones described the business of Major Engineering as "a well established engineering firm which designed and manufactured hydraulic equipment". He said that his dealings were with Mr Petty whom he described as "an engineer employed by Major [Engineering], who I came to know well. I greatly respected his knowledge and experience in hydraulic engineering".

13 Mr Petty was far more modest in his witness statement and, I fear, not altogether candid. He said he was but a fitter and turner by trade. He said that "Neither I nor MSS offer design services for a fee". He said, too, that MSS supplies and repairs hydraulic systems but does not manufacture them. He said that MSS stocks and sells proprietary brands of hydraulic cylinders and engages specialist companies to manufacture them.

14 Under cross-examination he agreed that he had been involved in the business of the supply of hydraulic equipment since 1964. In 1970 he was the director and sales manager of his own company, All Power Hydraulics Pty Ltd, a company specialising in hydraulics equipment. He was in 2002 familiar with hydraulic equipment and, in particular, with the Parker range of this equipment.

15 As to Major Engineering, he said in cross-examination that it was incorporated in about 1912 and that, during the ensuing 90 years, has carried on business in various engineering fields. One such field in 2002 was hydraulics, pneumatics and lubrication and this was the field which was the responsibility of MSS. The internet website of Major Engineering shows that MSS offers services in the nature of design of hydraulic systems and sells hydraulic equipment including the Parker range.

16 Little further information was provided as to the dealings in 1997 between Major Engineering and Mr Jones other than that Mr Petty did design a hydraulic system to drive the propeller in *Cadibarra 7*. This system worked satisfactorily and, in 2002, when a hydraulic system for *Skandia* was under consideration, Mr Jones turned to Mr Petty and to Major Engineering.

17 The two men had a number of meetings at which they discussed hydraulic aspects of the proposed canting keel system. This system involved pumps to deliver oil into the hydraulic cylinders at the required pressure as well as the cylinders including the pistons and connections themselves. The discussions appear to have focussed on the hydraulic cylinders.

18 At this point the accounts of the two men as to what was discussed at this early stage differ considerably. The discussions usually took place in the kitchen of Mr Jones' home and it does not appear that notes, if taken, have survived. It seems that these discussions focussed upon a hydraulic cylinder manufactured by a substantial US corporation specialising in these products, Parker Hannifer Corporation. Parker publishes a catalogue which contains specifications, dimensions and technical data relating to the Parker cylinder products. It is entitled "Industrial Cylinder Products Catalog" and numbered 0106-6. It is a substantial document: Section C, which is reproduced in the Court Book runs to 124 pages and the whole catalogue appears to contain about 350 pages. Mr Petty said that he gave a copy of this catalogue to Mr Jones early in their discussions. Mr Jones was adamant that he was not given this document and, indeed, that he was never shown it at any time. He said he was given some pages from another Parker catalogue, that relating to its Infinity Series cylinders.

19 In any event, in March 2002, Mr Jones spoke by telephone with Mr Petty. He asked him to provide cylinder sizes on two alternative bases. One, that there would be a single hydraulic cylinder with a capacity for a force of 60,000 lb; and the other, a pair of cylinders each with a 30,000 lb capacity. 30,000 lb is the equivalent of 133.5 kN and 13,604 kg.

20 Mr Petty responded by a fax dated 27 March 2002 proposing for the single cylinder a five inch bore which would operate at a pressure of 3,050 psi; or for each of the dual cylinders, a 4 inch bore operating at 2,400 psi. According to the fax cover sheet there are five sheets in the transmission. The inclusions are four pages from the Parker Infinity Series catalogue, not from the Parker Catalogue. Of present interest, these show the dimensions of the cylinders including the piston shafts. The fax invites Mr Jones to adopt these dimensions and concludes with a statement that "These we can make locally".

21 Mr Jones later formed the view that there should be a pair of cylinders and that their capacity should be greater than that he had previously given to Mr Petty. He discussed this increased capacity with Mr Petty and said that, nonetheless, he did not want the cylinders to exceed a maximum of 3,000 psi pressure in operation.

22 On 4 June 2002 Mr Jones sent to Mr Petty a critical email which is in the following terms:

"The cylinder arrangement I have decided on for WILD THING's canting keel includes two 5" x 930 mm stroke cylinders, one with a cylinder end as in the 'Infinity Pin Eye Series Style R' with pin diameter 1.25" and a spherical bearing rod end for a 1.75" diameter pin. Oil connections are to be in the 'Position 3' (Parker Hannifin). The other cylinder is to have spherical bearings for 1.75" diameter pins at both ends and oil connections at 'Position 1'.

Based on the data you gave me I am assuming 2" rod diameter and a retracted length for both cylinders of 1241 mm.

The maximum static loads (i.e. in the locked condition) shared by the two cylinders will be 44500 kg push force and 38267 kg pull force. Would you please check that my assumptions are OK and forward a quotation."

It will be seen from this that the load on the two cylinders has been greatly increased from a maximum 60,000 lbs (267 kN; 27,210 kg) to a maximum push force of 44,500 kg (437 kN; 98,127 lb). The new load in tension is 38,267 kg (376 kN; 84,383 lb). This appears to approximate the results of Mr Jones' ABS Guide calculation.[3] These loads are to be shared between the two cylinders so that each would be required to bear one half, that is, 218 kN (compression) and 188 kN (tension). The second is that Mr Jones had determined upon a stroke of 930 mm. It is not clear how this was arrived at other than that, given that the length of the cylinder assembly when retracted was 1,241 mm, this was the space available between the keel head, when the keel was canted at 19.5deg. to starboard, and the fixing place at the port side of the hull. In any event, this was the dimension provided to Mr Petty and he accepted it. Third, Mr Jones says in the email that he has made two assumptions which he asked Mr Petty to check. The first assumption is that the overall length of the cylinder assembly when retracted would be 1,241 mm. The second assumption is that the piston rod would have a diameter of two inches (50.8 mm). These assumptions might have been taken from the Parker Infinity Series Catalogue pages which had been enclosed in the Major Engineering fax of 27 March. The dimensions given in these pages for a five inch cylinder included a two inch (50.08 mm) diameter shaft and a total retracted length of 311.15 mm plus 930 mm for the stroke, a total retracted length of the whole assembly of 1,241 mm. These pages do not include another diameter for the piston rod in a five inch cylinder.

23 Mr Petty said that he made a calculation of the capacity of the suggested cylinders, each to deliver a force of 22,250 kg and 19,134 kg, as requested. This involved a fairly straightforward calculation based on the compressive force of 3,000 psi in the cylinder and the area of the piston. This calculation showed that the forces which might be produced were 58,920 lb (26,719 kg; 262 kN) in compression

and 46,920 lb (21,278 kg; 209 kN) in tension. He also applied the same formula to determine the sufficiency of the suggested 3000 psi operating pressure under the loads specified by Mr Jones in his 4 June fax. This exercise showed that the operating pressure of the cylinder was comfortably less than 3000 psi. In fact he made no response to Mr Jones' email, presumably on the basis that his silence would be taken as confirmation of the information provided.

24 Mr Petty did not, however, address the two assumptions which he was required to check. His silence was accepted by Mr Jones as confirmation. There may have been something further said between the two men about the retracted length, however, because consideration was given in June 2002 to reducing the length of the connections at the piston shaft end. Nothing, however, was said to cast doubt upon Mr Jones' assumption that a two inch piston shaft would suffice.

25 Following the transmission of the 4 June email, the two men continued to meet to discuss aspects of the hydraulic system. Mr Petty's paper copy of the 4 June email bears Mr Jones' handwriting which appears to raise the possibility that the cylinder diameter might be increased from five inches to six inches, but this came to nothing. Mr Jones said that there were further discussions regarding the required capacity of the hydraulic system. These appear to have focussed upon the pressure which the cylinders were to bear. This was fixed at a maximum of 3000 psi. Mr Jones said, and this was not challenged, that he made it clear to Mr Petty that this was to be the maximum pressure required and that this was the equivalent of a maximum compression force in a five inch cylinder of approximately 59,000 lb (approximately 262 kN) and a maximum tension force of 49,500 lb (approximately 220 kN).

26 On 25 July 2002 Mr Petty submitted Major Engineering's price for the cylinders in the sum of $1,664 per cylinder nett plus GST. On 29 August he sent to Mr Wharington care of Wildthing Yachting, a quotation in similar terms. It is –

"... for the supply of the following.

Item 1

2 only 5 inch Bore × 2 inch Rod × 930 mm Stroke Cylinders fitted with 40 mm Diam Spherical Bearing Both Ends."

This quotation was accepted on 3 September 2002.

27 The cylinders were duly manufactured by a subcontractor of Major Engineering and supplied to Timelink in November 2002. No complaint is made of the manufacture or of their installation. In September, they were subjected to a tension test at 220 kN. This was to test the end connections and the hydraulic assembly and they did not fail under that load.

28 What is put on behalf of Timelink is that the hydraulic cylinders had insufficient capacity to cope with the specified working force of 262 kN in compression and that, had they had that capacity, they would not have failed in the conditions encountered by *Skandia* on 28 December 2004.

Contractual Issues

29 The claim of Timelink, is pleaded as a breach of contract and as a breach of duty of care. When pressed, counsel for Timelink accepted that its central case was put in contract, so that, if that case failed, there was little point in the negligence claim.

30 It was pleaded by Timelink[4] that the contract between Timelink and Major Engineering was for the design, manufacture and sale of the hydraulic system, incorporating the cylinders[5] and that there were terms of this agreement that the hydraulic system:

"(a) would be reasonably fit for the purpose of controlling the canting keel on 'Skandia' in ocean racing conditions;

(b) would have a working capacity to withstand a compression force of up to 262kN in ocean racing conditions;

(c) would be designed and supplied so that the piston rods would not buckle or fail under compression when they were subjected to a working compression force of up to 262kN in ocean racing conditions;

...
(e) would comply with the guide contained in the Parker Industrial Cylinder Products (Hydraulic and Pneumatic Cylinders) Catalogue ('the Parker Catalogue') in relation to required piston rod size selection for metric hydraulic cylinders based upon the required thrust (push) application."[6]
I omit the term pleaded in paragraph 4(d) because no breach of this term is alleged.

31 The position of Major Engineering, as appears from its defence[7] is that the contract required it to supply cylinders to meet the specification set out in the quotation of 29 August 2002 and that it did so. It denies any design responsibility, saying that decisions of Timelink as to what were to be the dimensions of the cylinders and their performance capacity were the product of the professional judgment of its designer Mr Jones.

32 The four suggested terms are said to be partly oral and partly to be implied. The implication is said to arise in any or all of the following manners.

(i) by operation of law in order to give business efficacy to the agreement;

(ii) by reason of trade custom and usage;

(iii) by the *Goods Act* 1958 s 19; and

(iv) by the *Trade Practices Act* 1974 s 71(2) and/or s 74(2).

I reject immediately the contentions in part (i) and (ii). The contract for the sale of the cylinders is perfectly efficacious without any of the suggested terms. There was no evidence of industry custom. The three suggested statutory bases are all expressed in terms of a warranty of fitness for a disclosed purpose. As pleaded, only suggested term (a) could be seen as such a warranty. If the other terms are to be part of the sale agreement they must arise from the conversations which are referred to in the particulars.

33 In determining whether any of these terms should be imported into the contract it is necessary to bear in mind who were the contracting parties. Much of the evidence and argument was directed to the relative expertise of Mr Petty and Mr Jones. While this is an important factor, it must not distract attention from the fact that the parties involved were Timelink and Major Engineering. Nor am I concerned to resolve rights which Timelink might have against its designer, Mr Jones. Timelink, in its dealings with Major Engineering, had access to the engineering expertise of its consultant designer, Mr Jones and to the practical expertise of its director, Mr Wharington. Major Engineering was a

company which held itself out as a long established enterprise with capabilities including that of providing technical consulting services in hydraulic systems, and a range of services in this area of engineering. It was an agent for Parker and other major suppliers of hydraulic equipment. Its spokesman in its dealings with Timelink's engineer was a man with decades of experience in selling this equipment.

34 I conclude from this that, when Timelink sought or obtained advice as to technical aspects of a proposed hydraulic system, it was entitled to accept that this was accurate notwithstanding that it had its own design engineer. This is particularly the case where the product which Major Engineering was offering was a standard Parker cylinder or its locally made equivalent. The critical piece of information for present purposes was the diameter of the piston rod. The evidence to which I shall refer shows that the calculation which is required to determine this dimension from first principles is beyond the capacity of all but the most qualified specialists in the field of buckling failure and, perhaps, even beyond their capacity. The purchaser of a cylinder is, in these circumstances, entitled to accept the expert advice of the supplier, particularly where this is given by reference to standards recommended by the manufacturer.

35 The issue of some significance in this context arose from the disputed evidence of Mr Petty that he provided to Mr Jones a copy of the Parker Catalogue. Mr Petty was unable to recall when this was done. He said that at an early stage in their discussions he dropped it off at Mr Jones' house. Counsel for Major Engineering said that Mr Petty's account of this delivery is confirmed by the reference in Mr Jones' email of 4 June to the port positions in the cylinders. These are shown in a Major Engineering sketch which was in existence on 17 July and referred to in a fax of that date as being "Port position 3", that is, at 6 o'clock or at the base of the cylinder. It is in the Parker Catalogue, which Mr Jones said that he never saw, that port position 3 is shown as being at 6 o'clock. In the pages from the Parker Infinity Series catalogue, which he did have, port position 3 is at 3 o'clock. And so, it was put, Mr Jones must have had the Parker Catalogue at that time.

36 Unfortunately for Major Engineering, the evidence leading to this conclusion was a little uncertain. Mr Jones said that he did not see the sketch and that the port positions were not shown in the drawings in his possession. In fact the ports were to be at 3 o'clock in June 2002 but, when pressed, he said he was uncertain. When it was put to him that this demonstrated that he must have had the Parker Catalogue at this time, he maintained his categorical denial. He said that he thought that the design called for ports at 3 o'clock and that this had to be changed later at the time of installation. In these circumstances, I will not conclude that Mr Jones was in error when he denied that he received the Parker Catalogue.

37 Before I leave this, I make mention of a matter which was brought about by the manner in which the witness statements and Court Book were prepared. In fact there are two Parker Catalogues: that entitled "Industrial Actuator Products Catalog, 0106-5 (05/99)"; and that entitled "Industrial Cylinder Products Catalog, 0106-6 (06/02)". I shall refer to the former as version five and the latter as version six. Version five was discovered by Major Engineering and its solicitor requested the solicitor for Timelink to include it in the Court Book. This was not done. For some reason, only two pages were inserted, and these were from version six. This escaped the attention of counsel for Major Engineering, who conducted their case on the basis that it was a copy of version six that Mr Petty gave to Mr Jones in March 2006. Their error was compounded by the terms of Mr Petty's own witness statement which refers to the document which he gave to Mr Jones, simply as the "Parker Catalogue" without identifying it further or referring to it by its Court Book reference. Mr Jones' witness statement is no more helpful. He, too, simply refers to the Parker Catalogue specifications and his contradiction of Mr Petty's evidence as to its delivery is in these terms: "I strongly deny that Reg

Petty ever gave me a complete copy of the Parker Catalogue Specifications, including Charts and Graphs as he now alleges. Nor did Reg Petty ever show me a Parker Cylinder Catalogue". He does not identify the catalogue by its Court Book reference or otherwise. His denial may be taken as a reference to version six, so that, in fact, there is not here a true contradiction.

38 The discrepancy first came to light on day seven of the trial after Mr Jones had left the witness box. When Mr Petty was confronted with the two catalogues it was pointed out to him that the date on version six, which he had identified as having been given to Mr Jones in about March 2002, suggested that it had not been published until June of that year. He agreed that the copy which he had given to Mr Jones was probably not version six, but he was not able to produce a copy of the document which he said he actually gave to Mr Jones. At the very end of the trial, the significance of all of this became apparent to counsel for Major Engineering. They asked me in the circumstances, to treat the evidence as to the delivery of the Parker Catalogue as a reference to version five which they then tendered as Exhibit 53. Counsel for Timelink resisted this, pointing out that Mr Petty's evidence was given by reference to version six.

39 I will not put Major Engineering to the suggested disadvantage by reason of this confusion. It is clear that it is a product of the conduct of the lawyers and that Mr Petty's error is understandable given the way his evidence was led. I make no finding that the substitution in the Court Book by the solicitor for Timelink was due to anything more than oversight. I will treat Mr Petty as having given evidence that he provided Mr Jones with a copy of the then current Parker Catalogue. The catalogue then current was probably version five. I conclude this digression by observing that the pagination and content of those parts of the two versions of the Parker Catalogue which were referred to in evidence are identical. I now move on.

40 I make now some general observations upon the reliability of these two principal witnesses. Mr Petty struck me as defensive and without very much recollection of the detail of the events which he described. On many occasions he had to modify his oral evidence when it was demonstrated to be inconsistent with his witness statement. Mr Jones, on the other hand was forthright, confident and intelligent but with a tendency to arrogance.[8] There were, however, a number of matters that caused me to approach his evidence with some caution. I have already referred to his evidence of the calculations made by him in 2002 of the actual forces which he expected to be applied to the keel. His evidence that he made "lots of calculations", I found unconvincing. He said that he no longer had these calculations and he has made no attempt to identify them or to replicate them. In this way he denied to Major Engineering the chance to check his calculations. When directed by Court order to produce them, he produced only an ABS Guide calculation, which calls into question his evidence that me made "lots of calculations". My criticism of him or his working methods goes further. It is apparent that his initial calculation in terms of the ABS Guide contained errors of detail. Moreover, in any event, it is normal for an engineer to prepare and preserve the calculations which underlie the structure which has been designed. These are to permit another to undertake a check calculation and to expose the theoretical basis for the structure in case further or remedial work is required. Even accepting that Mr Jones is retired from his profession and that the *Skandia* project may not have been undertaken with the formality of a professional retainer, it is surprising that there is no set of design calculations in existence. All we have is a notebook which he referred to, perhaps accurately, as a "scrap book". All of this suggests that this project was handled by him with a degree of informality which is not consistent with the image of the careful and cautious design engineer which he sought to present.

41 Notwithstanding this, I am satisfied that, on the balance of probabilities, his evidence of not receiving the Parker Catalogue is to be preferred to that of Mr Petty. As I have mentioned, Mr Petty's

recollection was shown to be suspect on the issue of the delivery of the Parker Catalogue. If he was correct on this issue, I would have expected his version to have been put forward at an early stage in this litigation or even before it commenced. I conclude that Mr Jones was not given and did not see the Parker Catalogue. This conclusion means that Mr Jones did not have from Major Engineering any way of verifying the appropriateness of the two inch piston rod which is specified in the Parker Infinity Series Catalogue for a five inch cylinder which, in turn, is capable of operating under a pressure of 3,000m psi.

42 I can now set out shortly my conclusions with respect to each of the suggested terms of the contract referred to in paragraph [30] above. The question whether any of the suggested terms should be imported in the contract involves a consideration, based on what passed between the two men, of what the reasonable bystander would infer was the obligation offered and accepted as part of the contract between their principals. What Timelink must show is that Major Engineering assumed a contractual responsibility for the matters contained in the suggested terms.

43 On this basis, I reject term (a). It was, of course, obvious to both men that the yacht, incorporating the hydraulically driven canting keel, was to be used for ocean racing. Nevertheless, the dealings between them show that Major Engineering did not assume a general responsibility that the cylinders would cope with whatever forces might be imposed during this activity. It was for the designer to make this kind of decision.

44 Term (b) depends upon the requirement of Timelink that the cylinders be capable of withstanding a compressive force of 262 kN. Major Engineering recommended to Timelink’s design consultant that Parker cylinders would be appropriate and provided him with material from the manufacturer which suggested that the piston rods should have a two inch diameter. Mr Jones made known to Major Engineering his requirement as to the capacity of the cylinders to withstand a certain force. The email of 4 June gives this requirement as a total of 44,500 kg in compression. This translates to 218 kN for each cylinder. The higher figure of 262 kN is derived from the capacity of the cylinder to withstand 3,000 psi which Mr Jones required.

45 It is apparent from my summary of the dealings between the parties that both the maximum pressure within the cylinder and the maximum working load to be imposed on the rod were discussed. In the abstract, the two are not related because there are many factors which might bear upon the thrust which is produced by a piston in a cylinder subjected to a pressure of 3000 psi. In the present case, however, these factors, or possibly the more significant factors,[9] were known to the two men. For practical purposes, Mr Petty knew that the five inch cylinder at 3000 psi would generate a thrust of 262 kN through the piston so that, in operation, the hydraulic cylinder assembly including the piston shaft must be capable of operating in this environment. His conduct prior to the delivery of the Major Engineering quotation and its acceptance shows that he accepted that this was so and, further, that Mr Jones understood this and that he relied upon it.

46 I was for a little while troubled by the thought that, properly construed, the email of 4 June, which is on any view a very significant document, fixed the performance capacity of the cylinders in compression at only 218 kN. On reflection, I think this is not correct. The subsequent conversations make this clear. Furthermore, Major Engineering, both at this time and on 5 January immediately after the incident and, indeed, at trial, appeared to accept that, if there was a performance specification, it was by reference to a 262 kN working load.

47 I therefore conclude that the agreement between Timelink and Major Engineering contained a term that the hydraulic cylinders including the pistons would, in operation, have a capacity to withstand a compression force of 262 kN. I accept the term pleaded in paragraph 4(b) of the statement of claim.

48 For similar reasons I accept suggested term (c).

49 I do not accept suggested term (e). The Parker Catalogue was not given to or shown to Mr Jones at any relevant time. It was put in final address that this is beside the point. The suggested term was intended to mean that Major Engineering in making its recommendation as to a suitable cylinder assembly, would have regard to and apply the recommendations of the Parker Catalogue. The short answer is that this is not the term pleaded.

The Breach

50 As I have mentioned, it was accepted before me that the cylinders supplied by Major Engineering conformed to the dimensional specification contained in the accepted quotation. The concern is whether they conformed with the performance specification relating to maximum working load. In essence, it was put on behalf of Timelink that the two cylinders did not have a working capacity of 262 kN in compression.

51 The parties essentially addressed this issue in two ways: by theoretical analysis and by experiment, and I shall deal with each in turn. The evidence as to this was presented through Mr Jones and through a number of eminent experts retained by the parties: Peter Numa Joubert, Russell Horwood Keays, Andrew Harry Baigent and Peter Colin Raymond. These men had extensive and impressive qualifications and experience in engineering, particularly in buckling theory, and in sailing, and in most cases in both.

52 Professor Joubert was formerly professor of mechanical engineering at Melbourne University with a particular interest in ship resistance and ship motions. He is also an experienced yachtsman and yacht designer, having had over 100 yachts built to his designs. His yachts have enjoyed racing success including in the Sydney to Hobart race. He himself has competed in no less than 27 Sydney to Hobart races.

53 Dr Keays is a structural engineer with a particular interest in buckling. He is also an amateur sailer having completed two Sydney to Hobart races and three Melbourne to Hobart races.

54 Dr Baigent is a structural and materials engineer who practises as a consultant engineer with an involvement in an impressive list of major projects. Following his graduation from the University of Sydney in 1979 he undertook postgraduate research in which he worked with inelastic buckling analyses. He is not a sailor.

55 Mr Raymond is a marine mechanical engineer who has had experience as a ships engineer between 1956 and 1968. More recently he has been engaged in the investigation of marine accidents and associated insurance claims. Mr Raymond has a very long experience with sailing and racing yachts both in the UK and in Australia.

Theoretical Analysis

56 The failure with which this case is concerned may be described as a buckling failure. When a slender column is subjected to a longitudinal compressive force it will resist this increasing force until

the material reaches a failure point. It is at this point that the column distorts laterally and buckles. I shall refer to this point as the critical failure point and the force or load at which it occurs as the critical failure force or load. In the present case, the force was transmitted from the keel through the keel head to the end of the cylinder assembly which was nearly fully extended. This was a compressive force which the cylinder rods were unable to withstand. They failed in a buckling mode, bending with the force to a point where they broke away from the connections at the keel head.

57 It was generally agreed by the experts that the amount of compressive force which would cause a column to fail can be easily calculated by reference to a formula which was developed by an eighteenth century Swiss mathematician, Leonhardt Euler. The critical point of failure is less as the column is lengthened or when the ends in the column are not restrained. The Euler formula, however, does not contemplate the case where the column has different dimensions as is the case where it is a stepped column or a cylinder in extension. The 2171 mm column in the present case comprised a hollow cylinder of five inch (139.7 mm) diameter for part of its length (1162 mm including connection) and the balance (1009 mm including connection) a solid steel shaft of two inch (50.8 mm) diameter. Moreover, the connection between these two parts is not totally rigid and there was, too, a degree of overlap, inasmuch as the shaft ends at the piston which lies within the cylinder. This presents a very complex case for the application of a theoretical failure analysis, a case which has been addressed in the work of RJ Roark upon the elastic stability of a stepped straight bar in compression.

58 Professor Joubert applied the Roark formula to a column with differential diameters representing the cylinders in this case when extended to 2,171 mm, as they would be when the keel was canted to 19.5deg. to port. The theoretical critical failure point so produced was at 194 kN compressive load. He warned, however, that this was not a precise calculation and that, if more accurate dimensions of the cylinder be adopted, the failing load would be greater.

59 A more sophisticated theoretical analysis was undertaken by Dr K C Brown, a colleague of Professor Joubert. This analysis, which was adopted by Professor Joubert, produced a critical failure force of 260.6 kN. Professor Joubert said that, by reason of the connection between the cylinder and the shaft, this force should be reduced, but he did not say to what extent.

60 The Roark approach was also adopted by Dr Baigent to produce a critical failure point at a load of 320 kN. He produced a table of values calculated by Young and Budynas[10] from Roark's work based on the assumption that both ends of the multi-dimensioned column are pinned. He used the table to calculate the critical failure force in three circumstances. First, where the ratio of the moments of inertia of the less slender part and that of the more slender is 1. This occurs where the diameters of the two parts are equal. Other values of the ratio of the moments of inertia adopted from Roark were 1.5 and 2. Dr Baigent also undertook a calculation of the critical failure force on the basis that the ratio was infinity. The ratio for the cylinders here in question is 18.12 on the basis that the piston is fully extended. Dr Baigent then prepared a graph on which he plotted the coordinates from the three ratios from the Roark table as abscissas, that is on the horizontal axis, and the critical failure forces in kilo newtons as ordinates, that is on the vertical axis. He extrapolated from these three coordinates an asymptotic curve tending towards his infinity ratio calculation. He then determined from this curve the force required at the critical failure point for a column which had a ratio of 18.12. This produced an estimated critical failure force for the cylinders of 320 kN.

61 Professor Joubert strongly disagreed with this approach. He said that Dr Baigent's calculation for the ratio of infinity was erroneous. Dr Keays also took issue of this calculation. Professor Joubert said also that there is a great risk of error in extrapolating from three coordinates based on ratios not

exceeding 2.0 to a coordinate based on a ratio of 18.12. He concluded that it is not legitimate to extend the curve as did Dr Baigent. By way of illustration of this, Professor Joubert drew another curve from Dr Baigent's three Roark-based coordinates which produced a critical failure point of 261 kN at the ratio of 18.12. To my mind, these criticisms are justified. I do not accept Dr Baigent's estimate of 320 kN arrived at in this way.

62 There is no Australian standard applicable to the capacity of a cylinder in compression. I was, however, referred by Dr Keays to a US standard dealing with this matter[11]. The standard provides a formula what is described as a valuable aid for designers of fluid power cylinders showing results comparable with known test results and computer generated results. An application of this formula to the cylinders by Dr Keays produced a critical failure point at a load of 236 kN.

63 Next, Dr Keays undertook a calculation of the critical failure load of the cylinders using a computer program, "Strand". This produced, at a cant of 17deg., a theoretical maximum buckling load of 240 kN. To arrive at a critical failure load, he said, this figure should be adjusted downward to cater for the crookedness and yields that are referred to in AS3990, Mechanical equipment - Steelwork. His allowances for these produced a critical failure load of 195 kN. If the initial crookedness were reduced from 3.2mm to 0.8mm, however, this becomes 233 kN. In his second report Dr Keays agreed with Dr Baigent that 3.2mm was excessive, but he would not accept a crookedness figure of less that 0.67mm and I agree that 0.8mm is appropriate. A further downward adjustment to the figure of 233 kN must be made to cater for the fact that the computer calculation was undertaken with the cylinder extended for the keel at a cant of only 17 . The specified length of the cylinder at 19.5 cant is greater than that at 17 and the point of critical failure correspondingly lower. In his second report, Dr Keays estimates the allowance for this upon the buckling load to be 8%. If this figure were applicable to the load of 233 kN, it would produce a critical failure point of 216 kN. If the figure of 240 kN were applicable, the critical failure point at full extension becomes 222 kN.

64 It is not easy for me to evaluate these various theoretical analyses, depending as they do upon complicated and technical concepts and difficult mathematics. I have rejected Dr Baigent's estimate of 320 kN for reasons which I have given. I will not accept as accurate Professor Joubert's figure of 194 kN, given his warning about its unreliability. The remaining calculations produce results which vary from 218 kN to a figure less than 260.6 kN. All of them are likely to be excessive because they assume fixed connections. In the *Skiandia* the cylinders were connected at each end by a spherical joint which is less rigid than a fixed connection. I assume, too, that the ANSI based calculation has a factor of safety built in. I conclude that, on this basis, the critical failure point of the cylinders would be no higher than 230 kN.

65 I pause at this point to remind myself that the contractual requirement in this case was for a maximum static working load. This will be somewhat lower than the critical failure load. How much lower is a matter for judgment and the application of a factor of safety.

66 I come now to the last basis for estimating the critical failure point. The Parker Catalogue provides a chart for calculating the size of the piston rod in their cylinder range. It will be recalled that Mr Petty had this catalogue and chart but, as I have found, Mr Jones did not. The chart[12] is designed to enable the reader to select a piston rod for given thrust applications. The starting point is to calculate the basic length of the piston by multiplying the net stroke length by a stroke factor. In this case it was accepted that the stroke length was 930 mm and the stroke factor was 2, given the end connections which Mr Jones had selected. The basic length was, therefore, 1860 mm, which can be plotted on the vertical axis of the chart. The horizontal axis shows the thrust. This is calculated by multiplying the full bore area of the cylinder by the system pressure.[13] Parker's instruction, then, is to select the

piston rod size from the curve marked "rod diameter" immediately above the co-ordinates representing this basic length and thrust respectively. This intersection for a system pressure of 262 kN occurs between the 70 mm and the 90 mm rod diameter curves at a point which approximates 85 mm (3.35 inches). The Parker instruction, therefore, would lead to the selection of a piston rod diameter of 90 mm (3.5 inches). This is almost twice that of the two inch rod which Major Engineering supplied. On the basis of a basic length of 1860 mm and a piston rod of diameter two inches (50.8 mm) the chart would recommend a thrust not exceeding about 45 kN. This is considerably less than a 262 kN performance specification.

67 I conclude from this that, upon the manufacturer’s recommendation, the specified working load for the hydraulic cylinders far exceeded that recommended by it.

68 Compared with the theoretical critical failure loads which I have summarised, it would seem that Parker adopts a very conservative approach in its recommendations. This not surprising since its cylinders may be required to operate in a great variety of circumstances. The Parker factor of safety in these circumstances, appears to be of the order of 5.8.

69 It was generally accepted before me that, where a working load is specified for a column, this is not met by providing a column whose theoretical critical failure load is equal to that working load. Put another way, a column is not, in ordinary engineering practice, expected to work at its ultimate point of failure. The difference between the critical failure load and the maximum working load will depend upon the circumstances, including the experience and conservatism of the designer. These circumstances will include the confidence of the designer in the estimated maximum working load, the reliability of the structure and the consequences of failure. This was referred to as the factor of safety but, for present purposes, it may be seen to be a factor to cope with the unknowns or uncertainties inherent in the operation of any structural member. In AS3990-1993 it is referred to as the load factor[14]. Dr Baigent, on the other hand, said that the factor of safety represents the ratio of the buckling load to the maximum calculated load that can be applied to the cylinder. I am not confident that this is the correct use of the expression except insofar as the factor of safety includes an allowance for the difference between the buckling load (which is a theoretical load) and the known critical failure load. The factor of safety is a matter for the designer’s judgement rather than a mathematical ratio.

70 Views as to what is an appropriate factor of safety will differ. What was agreed in this case was that some factor should be applied. Otherwise, the working load would be equal to the critical failure load. Moreover, it was accepted that multiple factors of safety may be applied in any design process. In a case such as the present, the designer will form a view as to the worst possible circumstances in which the keel must operate. Having arrived at this worst case scenario, the supplier of components will then provide its own safety factor. It is for this reason that the Parker Catalogue appears to provide a safety factor of over 5 and the expert witnesses spoke of factors varying from 1.25 to 4.

71 Returning to the facts of this case, Mr Jones assessed the worst case working compressive loads to be 218 kN. He added to this somewhat in his cylinder pressure specification which required that the compressive working load be 262 kN. I have concluded that, upon a theoretical analysis, the cylinders supplied did not meet this requirement because the critical failure load in each case was calculated to be less that 262 kN.

The Tests

72 In order to test the actual capacity of the cylinders under a compressive load, Major Engineering, under the direction of Dr Baigent, had a new cylinder made to the specifications applicable to the failed cylinders and subjected this new cylinder to two compression tests.

73 The first test was undertaken on 11 October 2005 at the premises of Major Engineering. The overall length of the cylinder was set at 2166 mm. This is 5 mm less than the length of the cylinders at 19.5deg. cant, but no point was taken of that. Load was applied to the test cylinder in 500 psi increments. It was loaded to 22,250 lbs (218 kN) under a pressure of 2,500 psi. The pressure was then increased to 3000 psi, producing a load of 262 kN. The cylinder, including the piston rod, remained stable under this load. The conclusion from this, drawn by Dr Baigent, was that the test showed, contrary to the theoretical analyses, that at 262 kN the cylinder had not reached its critical failure point.

74 On 21 June 2006 a second test on this cylinder was carried out, again at the premises of Major Engineering and under the direction of Dr Baigent. The piston was extended so that the total length of the assembly from connection to connection was 2095mm. This was done because this was the dimension at the time of the failure. It is the equivalent of the keel in a canted position of 16 to port. Load was applied to the point where the cylinder exhibited instability. This was taken to be the buckling load. At this point the pressure within the 5 inch diameter cylinder was between 3730psi and 3750psi. This translates to a compressive force on the piston shaft of between 326 kN and 328 kN. Dr Baigent concluded from this that the buckling load of the cylinder under test, and that of each of the cylinders in the *Skandia*, was 325 kN. Dr Keays was of opinion that the reduction of the length of the cylinder under test by 70mm, compared with that at full extension of 2171mm, would increase this load by 8%. This means that Dr Baigent's conclusion from this second test can be interpreted as showing that the buckling load for the cylinder at 19.5 port cant is 309 kN. This produces a critical failure point which, for reasons offered by Dr Keays which I have mentioned, must be somewhat less than this figure. If, however, it is taken to be the same figure, it shows a critical failure point which is greater than that required of Major Engineering. It is 1.41 of the load of 218 kN specified in Mr Jones' email of 4 June 2002 and 1.18 of the load of 262 kN specified in the agreement, as I have found it.

75 A number of criticisms were directed to these tests with a view to impugning the validity of the results reported. In considering these it is necessary to bear in mind that no representative of Timelink was present at the first test; at the second test both Professor Joubert and Dr Keays were present. They had the opportunity to inspect the rig and its set up as well as the progress of the test and the condition of the cylinder assembly under test.

76 Criticisms of the first test were directed to the design and set up of the apparatus. In his first report Dr Baigent provides minimal information as to these matters. In his second report this witness asserted that the cylinder under test simulated the actual fixing conditions which were on the yacht. I accept this to be the case.

77 With respect to the second test, Dr Baigent said that he adopted the same testing procedure. In particular, he said that the test rig contained clevis plates and pins that engaged the spherical bearings which were attached to each end of the cylinder. Dr Keays criticises this aspect of the test set up, supposing that there was friction which impeded rotation of the bearings. He said, too, that the spherical bearings, being new, would be stiffer than those on *Skandia* at the time of the incident. They had then been used for a year or more.

78 There is no substance in these complaints. I accept that the connections were as provided for the cylinders in operation. The age of the bearings is beside the point. The contract specification called for

a given capacity when new. There is no evidence to suggest any deterioration in their condition on *Skandia* or any want of maintenance.

79 The next criticism relates to the presence of lateral force during the test. Dr Keays said that the self-weight of the cylinder and its rod would cause deflection, particularly having regard to the connection between the cylinder and the piston rod. This may be so. It would also have been the case when the pistons were installed on the yacht, for they were not mounted vertically. I have no knowledge as to the difference between the lateral force on a cylinder in a horizontal position, as was the case in test, and that in positions of varying horizontality, as would have been the case on a yacht in racing conditions. I reject this criticism.

80 Professor Joubert's criticisms were directed to these matters and to some others. He criticised the first test for the lack of a Southwell plot of deflection against load. In the case of the second test, there is no reason why Professor Joubert might not himself have recorded the deflections as the load was increased and prepared his own Southwell plot. An examination of the text[15] to which the witness referred me does not support the conclusion that the test results should be rejected for this failure.

81 The professor then said that the test did not conform to AS 4100-1998, Steel Structures. He said with respect to the first test that there was no evidence that the loads were maintained for the required periods. Whatever might be said about this for the first test, there was no evidence by this witness of non-compliance with respect to the test at which he was present. He also said by reference to cl. 17.5.3 of this same standard that, since one cylinder was tested as a prototype, any test should be accepted only if the result was 1.5 times the required loading. I am not persuaded that this is applicable to this case. The mention of prototype suggests that what is contemplated by this paragraph is that there is a sample prepared and tested to determine the capacity of similar products in a production run. In the present case there were only two cylinders produced so that the vagaries in materials and manufacture do not play a part. I reject this criticism.

82 Two further criticisms of Professor Joubert may have greater force. It was that the set up of the rig and the application of the compressive forces lacked a significant aspect of the environment in which the cylinders were required to operate on the yacht. This environment involved the constant and often violent movement of the vessel as it travelled through the waves. This movement in a vertical direction imposed dynamic lateral loads on the cylinders which were not replicated in the test. Further, the compressive forces imposed on the cylinders in operation were not constant or constantly increasing as in the test. The changing forces acting on the keel were transmitted to the cylinders as impulse loads. I should add immediately that these differential forces were noted and recorded on the yacht some time after the incident and were not observed to be great.

83 I have no way of knowing whether these criticisms, particularly the last, would invalidate the reported results. I would prefer simply to bear them in mind when I come to compare the test results with the theoretical analyses and such other evidence as bear on these issues.

84 My attention now returns to the appropriate factor of safety or load factor. As I have mentioned, various figures for this have been given. At one end the Parker Catalogue figure appears to be over 5. Then there are Professor Joubert's 4, the ABS Guide's not less than 2, Dr Keays' 2 (assuming the keel assembly is an unconventional or complex structure),[16] his 1.67 (assuming dead load only applies),[17] and Dr Baigent's 1.25.

85 I reject as inappropriate for my purposes the Parker factor. This may be taken as an attempt by the manufacturer to cater for all manner of uncertainties which might affect the operation of a cylinder in

an unknown number of applications. It may well be appropriate for this purpose but it cannot be translated to the present case where some attention had been directed by the design to the expected circumstances of the application of the cylinder.

86 If it were necessary I would reject Dr Baigent's figure of 1.25. It seems that he arrived at it from an application of cl. 6.1.1 of AS3990 which gives a load factor of 1.67 for a concentrically loaded member. Dr Keays appears to contemplate that this is appropriate notwithstanding the reservations expressed by him and others as to the applicability of this standard to a cylinder. Dr Baigent then goes on to apply to this figure a further load factor of 0.75 derived from cl. 3.3.1. As Dr Keays points out, this is not appropriate where the original load factor is to be applied in the case where a multiplicity of load types are not to be allowed for. In the present case Mr Jones' design working load was to cater for the most adverse combination of static and dynamic forces that may reasonably be expected. Nevertheless, it is calculated on the basis of a dead load only, that contemplated for the ABS calculation.

87 This is sufficient to resolve the present case, assuming that the second test accurately reports the critical failure load of 309 kN. The difference between this load and the contractually specified working load of 262 kN means that the application of any of the suggested factors of safety would not show that there has been compliance with the contractual specification.

88 If it were necessary for me to reach a conclusion as to the critical failure point of the cylinders which Major Engineering ought to have provided under the contract, it would be arrived by the application of a factor of safety of not less than 2. This means that the cylinders in question should have had a critical failure load of not less that 524 kN or greater. The breach has been established.

Causation

89 Timelink must also prove that this breach of contract was the cause of the admitted failure of the keel assembly. It seeks to do this by relying upon the accuracy of Mr Jones' assessment of the likely loads on the keel head in a worst case scenario, and by eliminating competing hypotheses which would lead to a conclusion that the failure of the cylinders was not a cause, but a step in the process of failure which was the product of some other cause.

Mr Jones' worst case assessment

90 It is a curious feature of this case that the ABS Guide method of calculating lateral force on a keel which is assumed to be a horizontal static, and out of the water, is said to reflect the lateral force in a worst case imposed upon the keel which is more or less vertical, submerged and subject to unknown and perhaps unknowable forces from the water through which it travels. Curious it may be, but it was the evidence of three very experienced yacht designers and yachtsmen in this case. Professor Joubert accepted, as did Dr Keays, that Mr Jones was correct in using the ABS Guide to arrive at a worst case scenario. Mr Raymond said that static forces during a knockdown are simply the beginning of the design process. To these must be added the acceleration and deceleration forces which are likely to occur in bad weather. He then went on to discuss abnormal waves. If, indeed, Mr Raymond was contradicting the opinions of the other three witnesses which I have just mentioned, I must prefer theirs. It is supported by the weight of the ABS Guide which is accepted by the Cruising Yacht Club of Australia as a standard for classifying vessels as fit for offshore racing. The Guide does warn designers of keels to exercise their own independent judgment, but its value as a guide based on considerable experience of ocean conditions cannot be ignored.

91 The relevance for a designer of the calculation required by the ABS Guide was explained by Mr Jones in the following lengthy answer:

"What I have to do in order to work out what happens in the boat is take the operating conditions in the boat and do separate mathematics in the operating conditions, in other words with the boat sailing along through the water under different conditions and consider different wave patterns and so on, and I have to then calculate what is the maximum moment on the keel because the moment on the keel because it is the moment on the keel that transmits the forces on to the cylinders or on to other parts of the structure and that moment has several main components. One is the static component which, if you like, relates to the component like this, by multiplying that by the sine of whatever the angle is that the keel happens to be in relation to the vertical, not the vertical in relation to the boat but the vertical in relation to the earth. I will come back to the dynamic factor in a minute. You have got to then apply two other series of forces to that, one is there is the buoyancy force because when the keel is - the boat is normally sailing the keel is under water and by Archimedes principle there is flotation of both the fin and the bulb. They produce moments which are in the opposite direction from the static moment and then there is a rather more significant loading which is associated with the fin going through the water and that's in two parts. The hull is making leeway so that the hull is a certain angle of attack relative to the water so like a wing surface, the fin keel is creating lift and it's got a side force on it relating to that and that also operates to reduce the moment on the keel, and then there is a further one which is related to the fact if a boat is coming off a wave as we talked about, and the boat is coming down more solidly vertically then that water force is bigger than it would be than if you were just going through. On top of that there is the dynamic loading you apply which is really a dynamic loading applied as factor applied to the static loading of the keel at such and such an angle. Then you calculate the total loading by adding those four things together, some of them are positive and some of them are negative and you get a total figure. Now, experience is that because that total figure in my experience, and I've tried lots of examples and so on on boats, the total of that figure is always substantially less than the static one when the boat's out at an angle and seeing obviously the worst condition, I have to meet the ABS requirement, I also have to meet the practical requirement so obviously I have got to meet whatever is the bigger of them – I have to meet whatever is the larger of the two of those particular things and therefore as in every case that is relevant in my experience, the calculation for ABS turns out to be bigger by a substantial margin than the sum of the other four items, it is therefore of convenience, and that is why, I believe, ABS is generally the requirement for race organisers because it simply is the worst case and you consider that."
92 I reject the submission put on behalf of Major Engineering that the ABS Guide is inadequate on the basis that it does not contemplate a canting keel. The guide provides a measure of the lateral forces expected to play on the keel. These forces do not depend upon the method of fixing of the keel to the hull. This will bear upon the ability of the craft to withstand them, which is an entirely different matter.

93 When the ABS Guide calculation is undertaken, the result arrived at by Professor Joubert of the force applied in compression to the cylinders was 24.97 t (about 245 kN)[18]. It will be recalled that Mr Jones' calculation of this made in 2002 produced a force of about 214 kN. Dr Baigent criticises the detail of Mr Jones' calculation but arrives at a force of 44,503 kg (218 kN). Mr Raymond undertook a similar calculation, arriving at a force of 47.255 t (232 kN). Finally, in his supplementary written statement, Mr Jones presents revised calculations of this force which generally agree with Dr Baigent's figure of 218 kN compressive force on each cylinder. On the balance of probabilities, I accept the accuracy of Mr Jones' assessment of the worst case scenario which the yacht might be expected to encounter in racing conditions.

Competing Scenarios

94 I was then presented with a number of scenarios which Mr Raymond said possibly brought about the damage to *Skandia*. I mention at this point that I am concerned with probabilities rather than possibilities.

95 The collision with the sunfish was relied upon. This occurred at 0825 hours on 27 December 2004, some 18 hours before the incident. The entry by Mr Oxley in his log records the following: "Hit huge fish with rudder and came to complete halt thought rudder gone, back away". It is clear from this and from the evidence of the eye witnesses, that the impact was to the rudder and not to the keel. What was then hypothesised by Mr Raymond was that the rapid deceleration to the hull imposed on the keel a horizontal force from aft as the keel tended to continue forward as the hull slowed. This, it was suggested, might have damaged the housing for the pin holding the canting keel. I am not persuaded that this occurred for the following reasons. The pin was later seen to be in place without damage to its housing. The yacht was able to sail with the canting keel in operation for many hours after the collision. I accept, too, that the deceleration of the yacht due to the impact with the fish was not so great as to put undue load on the keel connection to the hull.

96 The second hypothesis offered by Mr Raymond was that the vessel had encountered what is called a rogue wave, that is, a wave of such extraordinary size that the force imposed upon the keel head and the cylinders was beyond what the designer expected. This hypothesis derived some support from the fact that the keel had operated satisfactorily for over 12 months in a variety of ocean racing conditions without failure. The difficulty with this hypothesis is that the two facts which underpin it were not established. It is true that the keel had operated satisfactorily from the time the yacht was first put into service. What is significant is that, during this period, it was operating at only a maximum of 17deg. cant. This meant that the cylinders were less extended than in the 2004 Sydney to Hobart race and their critical failure point was, accordingly, greater.

97 The second fact, the presence of an abnormal wave, was not supported by the evidence. The sea conditions prevailing were shown on the video of the rescue. The navigator, Mr Oxley, described the incident in his log thus: "On port tack, large wave, crashed off wave." In an interview with a journalist he is recorded as saying:

"I was sitting at the nav. station... and we just launched off this enormous wave and were mid-air. I got hit by someone who that came flying out of the top bunk and we both ended up by the leeward hull. And when we came down there was this terrible crunching sound of carbon."
When he was examined as to the incident, it was apparent that Mr Oxley was not on deck; he did not see the wave. Although he accepted that that the journalist’s account was accurate, it was apparent when he gave evidence, that he was not comfortable with the expressions which the journalist had adopted. He agreed, however, that the wave appeared to be much larger than the waves generally being encountered and that the sound of the failure coincided with the yacht hitting the bottom of a trough.

98 The sailing master, Graeme John Taylor, was on deck when the incident occurred. He described the wave as a large wave; not an abnormal wave or an enormous wave. He said that the wave was large but that it was no larger than he had experienced on that watch from midnight. These waves, he said, were about 4 metres from the top to the median. This was consistent with the waves shown on the video. I am satisfied that the waves were as Mr Taylor described them. There was no rogue wave. The yacht was sailing into a 4 metre swell coming from the portside, the wave conditions were no more severe than those which the yacht was expected to encounter in the ocean. The rogue wave scenario has not been made out.

99 The third possible hypothesis envisaged by Mr Raymond appeared to depend upon the expression attributed to Mr Oxley that the yacht was in mid-air. The Raymond constructed from this the scenario that the hull of the yacht was entirely out of the water when a large wave moved past it. It then fell under the force of gravity to the trough of the sea. In order to get an idea of the forces which would then be applied to the hull and the keel, Mr Raymond made some calculations on the assumptions that the hull fell in this way, vertically, for 2.4 metres alternatively for 5 metres. In the first case, the force applied to the rams in compression would be 150 t (1,472 kN); in the second case the force would be 350 t (3,435 kN). Although this was embraced by counsel for Major Engineering, it should be noted that Mr Raymond offered this, not as an analysis of what happened, but rather as an aid to "allow a reader to visualise what would be happening as the craft obeys the nature laws of motion".

100 Even with this qualification, the evidence shows that it cannot be correct. The witnesses who were familiar with the performance of the yacht all said that this scenario was fanciful. The way the yacht moves in heavy seas approaching from the port bow is that the bow clears the wave and descends as it loses the support of the water, so that the vessel generally rotates vertically about its centre. The centre is where the keel is situate. This means that it is only the bow which "slams" down on the water. None of these witnesses would accept that this centre point behaves as Mr Raymond postulated. They did not accept that the hull and the keel would fall as he supposed, so that it would be subjected to the forces which he calculated. I accept their evidence as to this.

101 I mention, finally, Mr Raymond’s criticism of the efforts of the crew after the failure to bash the unsupported keel head from aft and from the portside in order to restrain its movement. He described this as reckless. He would have preferred them to have attempted to lash the keel head to the hooks which were formed by the deformed piston rods. This advice ignores entirely the desperate predicament of the crew following the failure and the concern of the master to prevent the moving keel head from destroying the pad which prevented it from dropping through the hull and leaving the yacht with no keel ballast at all. The point was, not surprisingly, not pressed by counsel for Major Engineering.

Conclusion

102 None of Mr Raymond’s hypotheses has been established as providing a satisfactory explanation for the failure of the hydraulic cylinders on 28 December 2004. The conclusion which I reach is that the cylinders were unable to withstand the forces which were exerted upon them in what were not abnormal conditions. The precise extent of these forces cannot be known, but I have accepted the accuracy of Mr Jones’ design assessment of them. It has been demonstrated on a theoretical basis that the cylinders would not withstand the specified 262 kN force and, importantly, that their capacity was very much less than the manufacturer recommended. While it is true that the test cylinder, if fully extended, may be taken as having a critical failure point of about 309 kg, this would mean that the cylinders had a safety factor of about 0.18. This is very much less than any of the expert witnesses would consider sufficient. I conclude, therefore, that it was this insufficiency which was the effective cause of the failure.

103 I find, therefore, that Major Engineering is in breach of its contract with Timelink to provide hydraulic cylinders with the capacity to withstand a working force of 262 kN and that this breach was the cause of the failure of the cylinders and the consequent damage to the yacht. It follows from this that Major Engineering is liable in damages to Timelink. The quantum of such damages remains to be determined.

[1] For present purposes and, generally, I proceed on the basis that a force of 1 kN is equal to 101.9 kg and 224.7 lb.

[2] Other engineers who gave evidence confirmed the general accuracy of this calculation although their figures were slightly different.

[3] See para [11] above

[4] Amended statement of claim filed 22 April 2005.

[5] Statement of claim para 3.

[6] Statement of claim para 4.

[7] Amended defence filed 13 July 2006.

[8] By way of example I mention his reference in his calculation provided to the Court to a principle as being "school boy physics".

[9] Mention in evidence was also made of the impact of friction in the bearings and of the impact of an off-centre valve in this cylinder, but these were not said to be significant.

[10] W C Young and R G Budynas, *Roark's Formulas for Stress and Strain*, 7th Ed, McGraw-Hill at p 718.

[11] ANSI (NFPA) T 3.6.37-1991.

[12] For present purposes I speak of the metric tables appearing at page 109 of each version of the Parker Catalogue.

[13] This produces, for each of the cylinders here in question, a thrust of 262 kN.

[14] Cl. 1.6.3.8.

[15] S Timoshenko, *Strength of Materials*, 3rd ed, NY p 268.

[16] Assuming AS3990 cl 3.2.5 applies.

[17] Assuming AS3990 cl 6.1.1 applies.

[18] See para [11] above.