Airtrust (Hong Kong) Ltd *v* PH Hydraulics & Engineering Pte Ltd
[2015] SGHC 307

Case Number : Suit No 219 of 2013

Decision Date : 30 November 2015

Tribunal/Court : High Court

Coram : Chan Seng Onn J

Counsel Name(s) : Tan Chuan Thye SC, Avinash Pradhan, Alyssa Leong and Arthi Anbalagan (Rajah & Tann Singapore LLP) for the plaintiff; Daniel John and Kevin Cheng (Goodwins Law Corporation) for the defendant.

Parties : AIRTRUST (HONG KONG) LTD — PPH HYDRAULICS & EENGINEERING PTE LTD

Commercial Transactions – Sale of Equipment – Breach of Contract

Contract – Remedies – Punitive Damages

[LawNet Editorial Note: The appeal to this decision in Civil Appeal No 234 of 2015 was allowed while the appeal in Civil Appeal No 96 of 2016 was dismissed by the Court of Appeal on 11 April 2017. See [2017] SGCA 26.]

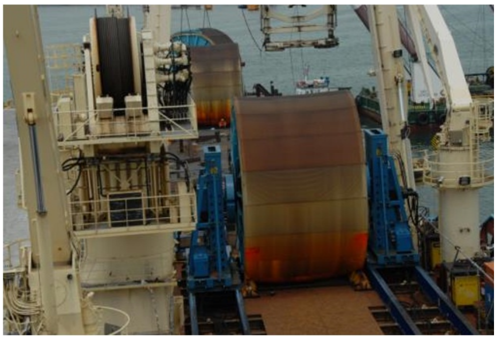
30 November 2015 Judgment reserved

Chan Seng Onn J:

Introduction

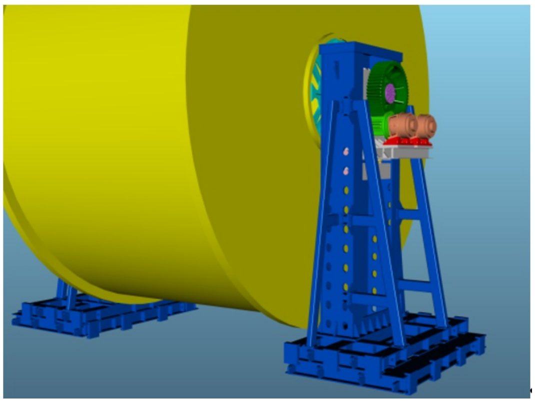
1 The Plaintiff is a company incorporated in the Special Administrative Region of the People's Republic of China, Hong Kong. The Defendant is a company incorporated in Singapore and is an established supplier, designer and manufacturer of heavy machinery (including tensioners and winches) for offshore use in the marine and oil and gas industry.

2 In 2007, the Plaintiff purchased a 300 tons reel drive unit ("RDU") from the Defendant. Attached below is a photograph of the RDU that was fabricated by the Defendant. The photograph shows a 300 tons reel (brown in colour) sandwiched between the two towers of the RDU (blue in colour) mounted on skids (blue in colour) on the deck at the aft of the ship, the "Maersk Responder".



3 For a better understanding of the overall structure of the RDU, I have also attached below a

colour schematic diagram of the RDU.



4 The RDU was delivered to the Plaintiff on 10 April 2008 and thereafter mounted on board the "Maersk Responder" for the laying of undersea umbilical for Nexus Energy at the Longtom field in the Bass Straits of Australia. After having laid one complete reel of umbilical and in the course of laying the second reel, a major failure of one of the gearbox assemblies occurred. The hydraulic drive motor and gear assembly on one of the towers of the RDU (*ie*, Tower A) came off its mounting and fell down. This incident caused the Plaintiff to investigate into whether there were any inherent manufacturing or design defects in the RDU.

5 In this action, the Plaintiff claims that the RDU already suffered from manufacturing and design defects at the time of delivery. Broadly, these defects pertain to the following:

(a) bolts;

(b) gears;

(c) bearings;

(d) structural strength of the two RDU towers and other structural components;

(e) excessive deflection under load of the sub-frame carrying the drivetrain;

(f) brakes; and

(g) the failure to manufacture the RDU according to the specifications and drawings.

6 The parties have agreed that the court is to determine this claim under Singapore law notwithstanding the choice of Western Australian law in cl 21 of the Sale and Purchase Agreement ("Sale and Purchase Agreement") for the RDU.

7 As the trial progressed and with fresh evidence emerging, the parties felt that amendments to their pleadings were needed. Instead of piecemeal amendments, the parties waited until the completion of the trial to consolidate all their intended amendments and attached them to their closing submissions. I think this is an efficient and cost-effective approach. No party has been prejudiced as the parties were clearly aware from the beginning of the trial what the main issues and areas of dispute were and they were not taken by surprise. Accordingly, I grant them leave to amend their pleadings.

8 After a technically challenging trial over several days where 17 factual witnesses testified and seven expert witnesses from different fields of engineering gave their expert opinion as a group in a "hot-tub" instead of sequentially, and after having carefully considered all the relevant evidence and the detailed and comprehensive submissions from both parties, I find on a balance of probabilities, for the reasons stated in this judgment based on the pleadings as amended, the facts that I have found and the totality of the factors that I have evaluated and taken into consideration, that the RDU:

(a) is not of merchantable quality;

(b) is not fit for the purpose for which it is intended to be used;

(c) is not free from defects in design, manufacture or workmanship; and

(d) does not meet the relevant industry standards and certifications, or the specifications and certifications stipulated in the contract between the Plaintiff and the Defendant.

9 For ease of reference, I have attached at Appendix 1 a "Glossary of Witnesses" with a brief summary of the focus of their evidence at the trial. In the course of this judgment, I shall refer to the evidence of some of these witnesses.

Sale and Purchase Agreement

10 It is not disputed that the Defendant was aware that the RDU was purchased by the Plaintiff specifically for lease to Trident Offshore Services ("Trident") for the laying of undersea umbilical in the Bass Straits of Australia for Nexus Energy in the Longtom Project.

11 The Plaintiff and Defendant entered into the Sale and Purchase Agreement dated 7 September 2007 which provides, *inter alia*, that:

(a) the Defendant was to sell a 300 tons RDU and deliver it on 14 January 2008 to the Plaintiff;

(b) the total purchase price of the RDU was $895,000 inclusive of "*ABS [American Bureau of Shipping] Full Certification*" which was itemised at a cost of $20,000 (See Schedule 1 of the Sale and Purchase Agreement at paragraph 2 - Scope of Supply);

(c) the RDU was to be fully certified by ABS;

(d) the RDU would be of merchantable quality, fit for the purpose for which it was intended and free from any latent or apparent defect in material or workmanship (See cl 15 of the Sale and Purchase Agreement); and

(e) the Defendant would perform all work diligently, carefully, in a good and workmanlike manner and in accordance with accepted industry standards (See cl 24 of the Sale and Purchase Agreement).

12 When it transpired that ABS does not provide certification for machinery such as the RDU, the Defendant suggested in an email dated 12 October 2007 that ABSG Consulting Inc ("ABSG"), an entity related to ABS, carry out the "*design review, site survey and final testing, same as what [the Defendant] did for the Acergy 100 Ton reel drive system*" instead. The certificate from ABSG for the Acergy 100 Tons spooler tower was attached for the Plaintiff's consideration. Mr Terry Griffiths, the

Principal Pipeline Engineer of Trident ("Mr Terry Griffiths"), replied on behalf of the Plaintiff vide an email dated 17 October 2007 that *"the use of ABS Consulting* ***for design review of the reel towers*** *is appropriate given there is no Class applicable*, ***however can you please ensure the design/structural codes selected are appropriate "*** [emphasis added in bold]. The Defendant's project manager, Dr Yang Ting, clarified during cross-examination that the certification for the RDU, now agreed to be performed by ABSG instead of ABS, remained a "*full certification*".

13 It is not disputed that the Defendant was solely responsible for designing the entire RDU although this obligation was not expressly stated in the Sale and Purchase Agreement. In fact, the Defendant's Confidential Bundle of Documents ("Confidential Bundle"), containing *inter alia* the design drawings and calculations submitted to ABSG for the purpose of obtaining certification from ABSG for the RDU design, was never provided or shown to the Plaintiff at that time. Basically, the Defendant kept the design of the RDU confidential. The Plaintiff was unaware of the engineering details and the design calculations in relation to the design of the RDU.

14 After the Defendant had designed and fabricated the RDU, a factory acceptance test ("FAT") was carried out and signed off by Mr Terry Griffiths on behalf of the Plaintiff on 26 February 2008 and counter-signed by Mr Chong, the then ABSG surveyor with ABSG, and Mr Steven Gan, the Assistant Manager-After Sales of the Defendant. ABSG issued certificates for the design reviews it carried out, as well as the manufacturing processes and the FAT.

Inadequate design of the RDU

15 I will deal with the design inadequacies first and there are several alleged by the Plaintiff. I will address them seriatim.

Failure to take the vessel roll into account

16 The Plaintiff submits that the Defendant failed to take vessel roll into account in the RDU design. It is not disputed that the inertial forces arising out of the vessel's accelerations must form part of the design considerations for the RDU. The input design parameters for any computer program (*eg*, the STAAD.Pro analysis) using a finite element method (*ie*, Finite Element Analysis or "FEA") to perform the stress calculations for the various components of the RDU structure mounted on board a vessel must therefore take into account the expected motions of the vessel due to current, wind and wave action.

17 I accept the explanation of Dr Yang Ting that there are basically two acceptable methods of specifying such input design parameters for the *inertial* forces due to the vessel's motions for the purpose of stress calculations:

(a) Specifying the equivalent accelerations that the equipment mounted on a vessel would be subjected to in the X, Y and Z directions corresponding to the ship's transverse, vertical and longitudinal directions, which are to account for pitch, sway, yaw, heave, roll and surge motions of the ship out at sea; or

(b) Specifying the maximum pitch, yaw and roll angles of the vessel and their respective periods of oscillation, the maximum sway, heave and surge and also the specific location of the equipment installed on the vessel, so that calculations can be performed to translate and resolve them into their corresponding accelerations in the respective X, Y and Z directions.

18 In either case, those equivalent accelerations arising from the vessel's motions can be used for

computing the *inertial* forces in the X, Y and Z directions for the structural and stress analysis.

19 But if in either case the maximum roll angle of the vessel for example is expected to be large, then the weight of the equipment itself (which is a gravitational force and not an inertial force) must be resolved into the X and Y components because of the large roll angle and then *added* to the inertial forces in the respective X and Y directions (which arise separately from the equivalent accelerations due to the ship's roll) in order to perform the structural and stress analysis. If that is not done, there can be a serious underestimate of the expected *total* force (due both to inertia and gravity) in the X direction when the vessel is at an inclined position relative to the horizontal at its maximum roll angle.

20 It is also significant to note that the design parameters in the form of the accelerations in the X, Y and Z directions for use in the design calculations of the RDU were in fact proposed, not by the Plaintiff, but by the Defendant two months after the contract was made and at a point of time when the Defendant was already aware that the RDU was to be utilised in the Bass Straits of Australia. Mr Terry Griffiths requested Dr Yang Ting on 15 November 2007 to advise the Plaintiff of the design accelerations to be used for the RDU reel tower. Dr Yang Ting responded on behalf of the Defendant on the same day that:

> The accelerations we adopted for the operational conditions are:
>
> Longtitudinal: 0.4 m/s^2
>
> Transverse: 2.2 m/s^2
>
> Heave: 3.7 m/s^2
>
> The above data are also used for the Acergy reel tower design.
>
> We are calculating the reaction force and will let you know later on.

21 I find that these are acceleration values for computing ***purely*** inertial forces due to the vessel's motions. The vertical gravitational force or weight is ***not*** included in these acceleration values. If it had been, then the acceleration value for heave for example cannot be as low as 3.7 m/s^2 (or 0.377 g) when the equivalent acceleration value due to gravity is already 9.81 m/s^2 (or 1.0 g). Obviously, the weight of the equipment must be added to the inertial vertical force due to the heave acceleration of 3.7 m/s^2 to derive the ***total*** maximum vertical force acting on the structure for the purpose of stress analysis, thereby giving 3.7 m/s^2 + 9.81 m/s^2 = 13.51 m/s^2 (or 0.377 g + 1.0 g = 1.377 g). The same logic should apply in the case of roll. Any X component of gravitational force or weight acting in the X direction (due to the inclined position of the equipment on the deck of the vessel at the maximum roll angle), ***if significant*** , must be added to the inertial force in the X direction due to the maximum roll acceleration at that maximum roll angle to give the ***total*** maximum force in the X direction acting on the structure for the purpose of stress analysis.

22 The weight of the equipment must therefore be separately accounted for over and above the inertial forces arising from the stated acceleration values in the structural and stress analysis. In other words, in order to assess the maximum value of the ***total*** force in the X, Y and Z directions, the components of the weight (due to the vertical gravitational force) resolved in the X, Y and Z directions (relative to the vessel), ***if significant*** , when the vessel is positioned at its expected

maximum roll and pitch angles, must be added to the inertial forces due to the stated acceleration values in those corresponding directions as provided in [20] above for the design.

23 After a careful examination of the calculations, I find that the Defendant had additionally provided for the vertical force due to the RDU's weight in the vertical Y direction over and above the inertial force due to the heave design acceleration value of 3.7 m/s^2 (or 0.377 g) (as it should), but had simply ignored the possible transverse force due to weight in the X direction when the vessel is at an inclined position due to roll (which it should not do). Effectively, the Defendant simply assumed that the maximum possible roll angle of the vessel out at sea is either zero or always negligible in value, which in my view is an incorrect assumption considering the kind of sea conditions that may be encountered in the expected areas of operation of the RDU. I thus agree with the Plaintiff's submission that at least in this respect, the Defendant failed to take the vessel's roll into account in designing the RDU. Bearing in mind that the full reel alone is a massive 300 tons, a small roll angle of say 10**°** from the vertical is already enough to generate a component of force due to the reel's weight on the towers in the X direction of some 52.1MT [note: 1], which is not at all negligible. A roll angle larger than 10 degrees will obviously generate an even higher component force due to the reel's weight in the X direction on the towers, which must be added to the maximum inertial force in the X direction specified at 300 tons × 2.2 m/s^2 = 66 MT for the structural and stress analysis.

24 To overcome this criticism of its design, the Defendant argues rather unconvincingly that the 2.2 m/s^2 acceleration in the transverse direction would have included *both* effects of roll (*ie*, the roll acceleration to account for the inertial force and the equivalent acceleration due to the resolved X component of the combined weight of the Tower and the full reel of 300 tons at the extreme end of the roll motion of the vessel). If that is the argument, then the Defendant as the designer who had set these design parameters, has not shown to me that the quantitative figure of 2.2 m/s^2 is sufficient in itself to account for *both* effects during the typical roll motion of the vessel during the RDU umbilical laying operation out at sea and in particular in the Bass Straits, with the typical sea conditions expected in that area. The Defendant simply adopted the accelerations used in its design of a 300 tons RDU that it had constructed for another company, Acergy, without establishing the adequacy of the transverse acceleration design value of 2.2 m/s^2 (or 0.224 g) in the first place for this RDU (to be deployed for use in the Bass Straits) based on its purported premise that ***both*** effects (*ie*, gravitational and inertial effects) had been subsumed under ***one*** equivalent acceleration value.

25 Without the Defendant producing a set of calculations setting out its roll assumptions and the necessary vector calculations to demonstrate the equivalence with the acceleration figure adopted in the RDU design, it is not possible to establish to what extent and with what sufficiency each of the two effects of the roll has been incorporated within the purported composite acceleration figure of 2.2 m/s^2 Without knowing the roll assumptions used by the Defendant, it is also not possible to establish if the acceleration values used for the RDU's design are acceptable considering the kind of sea and weather conditions that the vessel with the RDU mounted on its deck is expected to be operating in.

26 In the absence of information from the Defendant of the design assumptions it had adopted in setting the design criteria for the X, Y and Z accelerations, the Plaintiff's expert, Mr Moore, had no alternative but to compare the acceleration values adopted by the Defendant for the RDU design with the typical acceleration values that classification societies have stipulated for use in the design and construction of ships and offshore structures to ensure compliance with their respective classification standards. At least, this would help to indicate whether there is any serious deficiency in the acceleration values adopted by the Defendant upon which the design of the RDU structure was

fundamentally premised.

27 In this regard, I have no reason to disagree with the following expert opinion and comments of Mr Moore in his affidavit of evidence-in- chief ("AEIC"):

9.2 Assessment of seaway ratings and comparison with class society values

Classification societies (Det Norske Veritas (DNV), American Bureau of Shipping (ABS), Lloyds Register (LR) and Noble Denton (ND)) establish and maintain technical standards for the construction of ships and offshore structures and provide validation services and surveys to ensure compliance with these standards. Typically, the offshore oil & gas industry would follow the guidance of one of these Class societies.

Table 4 provides a comparison of the accelerations used by [the Defendant] compared to those recommended by DNV in their code for portable offshore units [12]. This shows that the [Defendant's] values are significantly lower than those specified by DNV.

Table 4: Comparison of [Defendant] vs. DNV vessel accelerations

	PH Hydraulics	DNV
Longitudinal	0.04g	1.0g
Transverse	0.22g	1.0g
Vertical	0.37g	0.7g-1.3g

....

Normally, values quoted by class societies would be those used in the design of such equipment. The higher accelerations quoted by DNV would have been a more appropriate design condition in this instance. This would have been advantageous due to the intended operation of the unit within the Bass Straits especially when considering the notoriously extreme weather conditions applicable to this region, as described in the BPP-TECH Bass Straits weather assessment [14].

[The Defendant] has chosen to design the unit on the assumption that their figures chosen for the design of the unit (table 4) are relevant to the intended offshore operating location of the vessel. This is not appropriate as the equipment should be designed for use at any global location. For such equipment to be designed on the basis of use on a single, project specific, offshore location is not acceptable.

When purchasing or hiring such equipment, it is reasonable to expect that it can operate in any world-wide location and that the responsibility remains with the designer/manufacturer of the equipment to ensure that this requirement is fulfilled. The quotation provided by [the Defendant] [15] or the purchase order from [the Plaintiff] [16] does not specify any particular offshore operating location. In such cases, an appropriate Class design code (e.g. DNV) should have been used by [the Defendant] as the basis of design to ensure that all sea going conditions are met.

28 Where there is a significant difference between the acceleration values stipulated in the design codes of the well-known classification society cited by Mr Moore and those accelerations adopted by

the Defendant for its RDU design, it behoves the Defendant to justify in more detail how and why those accelerations values can be so much lower. When the Defendant does not explain how its low acceleration values were derived (which would affect the safety and structural integrity of the RDU as a piece of offshore equipment mounted on board a ship for operations out at sea and in particular in the Bass Straits), I will draw an adverse inference that the RDU, designed using the Defendant's acceleration values that were significantly lower than those recommended by DNV, is under-designed in terms of its structural strength. This is one of the many factors in my overall consideration as to whether the RDU is of merchantable quality and fit for its purpose.

29 Of course, if the Plaintiff can further demonstrate, as will be seen later, that the RDU is already under-designed when it is based on the Defendant's transverse acceleration design value of 2.2 m/s^2 (or 0.224 g), then the degree of under-design will necessarily be even greater when the acceleration design value adopted by the Defendant does not *in fact* include the effect of weight in the X direction due to the vessel's roll.

30 The Defendant next argues that it never agreed to design an RDU that could withstand the worst conditions in the Bass Straits, thereby suggesting that they had not *in fact* done any analysis to determine whether those accelerations adopted for its design were in fact appropriate for vessels operating in the Bass Straits. The Defendant relies on cl 9 of the Sale and Purchase Agreement which states that the purchase order including any attachments constitutes the entire agreement between the supplier and contractor. The Sale and Purchase Agreement makes no reference to the fact that the Defendant was to design the RDU for operations in the Bass Straits, although the Defendant does not dispute that it had been told that the RDU would be used for umbilical laying for the Longtom Project in the Bass Straits.

31 The Defendant wholly misses the point. When the Plaintiff as the client of the Defendant informs the Defendant of the specific area of operation of the RDU, and the Defendant, as the designer of the RDU, proposes values of accelerations as parameters for the design, which are subsequently stipulated in the Sale and Purchase Agreement, the responsibility for ensuring the appropriateness of those values of accelerations to be adopted as design parameters does *not* shift to the Plaintiff merely because the Plaintiff agrees or does not object to those design parameters. The Plaintiff is not the designer and does not know whether the design parameters are the correct and appropriate parameters to be used. Neither has the Plaintiff contractually accepted any responsibility to check on the correctness and appropriateness of such parameters. The Plaintiff may not even have the design expertise and domain knowledge. It is the Defendant as the designer under the contract who remains solely liable for the design parameters that it wishes to adopt even though the Plaintiff, as its client, may have accepted the Purchase Order which spells out the design parameters chosen by the Defendant. It must be borne in mind that the Plaintiff is not contractually a co-designer of the RDU. The Plaintiff engaged the Defendant to be the sole designer of the RDU for which the Defendant had been made fully aware that the RDU "*[p]rojects can be worldwide but first project is in Australia (Bass Strait)*."

32 I give an example. If after knowing that the RDU is to be used in the Bass Straits, the Defendant proposes extremely low acceleration values as design parameters, which are only appropriate for machinery installed on a vessel operating in relatively calm inland waters and which are therefore totally inappropriate or inadequate for machinery mounted on an ocean going vessel, can the Defendant, who has in the first place contractually undertaken the sole responsibility for the RDU design, turn around later, rely on its entire agreement clause and then raise a defence that the Plaintiff has already accepted those very low acceleration values proposed by the Defendant, and therefore the RDU, as designed, is deemed fit for its purpose simply because those design parameters have been contractually accepted by the Plaintiff? I do not think that the Defendant can do so. When

the Defendant has been informed of an important purpose and the area of operation of the RDU (*ie*, that it is to be used out at open sea, and its first area of operation is in the Bass Straits), it is incumbent on the Defendant, as the designer, to select and use appropriate acceleration values for the structural strength design that ensures that the RDU is at least operable out at open sea, if not in the Bass Straits, and which at the minimum is consistent with the typical acceleration values stipulated by classification societies (*eg*, DNV). I cannot accept that the values of accelerations selected by the Defendant for the RDU design would make the RDU fit for a purpose already made known to the Defendant, when those values are way below those recommended by classification societies for use as design parameters for machinery mounted on ocean going vessels. The comparison table of acceleration values set out at [27] above shows that inappropriate acceleration values were selected, proposed and used by the Defendant as the designer. I therefore reiterate that I have to take into consideration the likely under-design in the structural strength of the RDU, based on the relatively low acceleration values selected and used by the Defendant (*ie*, in comparison with those set out by the classification society DNV) having regard to the fact that the Defendant had been made fully aware of the known purpose and area of operation of the RDU, when I consider whether the RDU is of a merchantable quality and fit for its purpose.

33 I also accept the Plaintiff's submissions that the Defendant is under a statutory obligation to ensure that the RDU is fit for the particular purposes made known to it: see *National Foods Ltd v Pars Ram Brothers (Pte) Ltd* [2007] 2 SLR(R) 1048 at [54]. It has never been the law that in order for this obligation to bite, the particular purpose must *itself* be a term of the agreement: see *Benjamin's Sale of Goods* (M G Bridge, gen ed) (Sweet & Maxwell, 9th Ed, 2014) at para 11-055. In short, it must be part of the design obligations of the Defendant to stipulate appropriate design parameters in the contract specifications in the light of the known purpose and areas of operation of the RDU when the Defendant has undertaken full responsibility for its design.

34 Another troubling aspect is that the Defendant wrongly treated the reel to be fixed with a pin joint at both ends of the tower in its STAAD.Pro analysis. Clearly, it is erroneous to assume that the reel is not free to slide along its shaft in the transverse direction because the tower reel inserts are simply inserted into the reel hub and the reel is therefore free to slide. Accordingly, the transverse force calculations for the FEA analysis are affected.

35 After reviewing the Defendant's own design calculations, Dr Yang Ting eventually admitted that the Confidential Bundle shows that the Defendant had wrongly calculated the transverse force acting on the RDU tower due to the transverse acceleration by attributing only half of the transverse force to each of the RDU towers. Based on the Defendant's design parameters, the mass of the full reel of 300 tons subjected to a transverse acceleration of 2.2 m/s^2 in fact produces a total transverse force on the RDU of 66 MT (*ie*, 300 tons x 2.2 m/s^2). Since the reel can slide on the shaft (because it is not secured on both sides), it is wrong to assume that the 66 MT transverse force is shared equally by both RDU towers. If the frictional force is disregarded, the full 66 MT transverse force (and not 33 MT as was adopted in the calculations in the Confidential Bundle) must be taken to bear down entirely on the single tower that is restraining the full effect of the roll motion of the 300 tons reel.

36 I therefore find that the Defendant had wrongly computed the transverse force acting on the tower based on the transverse acceleration of 2.2 m/s^2 that it had itself proposed and adopted as one of the input criteria for the design of the 300 tons RDU. Accordingly, the Defendant significantly underestimated the transverse *inertial* force acting on each RDU tower during its structural design calculations premised on its own input criteria.

37 The ingenious response from the Defendant's expert, Mr Huang, is that the frictional force

restraining the slide in the transverse direction, which was previously omitted and not considered in the calculations, must now be taken into account for the structural analysis. The Defendant had to find a way to remedy the error in the structural design calculations.

38 Ms Renuka Devi, the Lead Structural Engineer from ABSG, very helpfully re-computed her structural analysis pursuant to directions from the court, using the same transverse acceleration of 2.2 m/s^2, but with the addition of the friction force. She summarised her results at Exhibit D3 (reproduced in the table below ("the Table")) showing the "Beam Unity Checks" under various scenarios and with different assumed friction coefficients.

Beam Unity Checks under various scenarios

Beam	1 As Submitted (From Confidential Bundle A115-A125)	2a (Case 2) Reel sliding, Friction coeff 0, Relevant LC	2a* (Case 1) Reel Sliding, Friction Coeff 0, All LC (From ABSG 13 Mar Report)	2b Reel sliding, Friction coeff 0.16, Relevant LC	2c Reel sliding, Friction coeff 0.3, Relevant LC	3a Struts bolted, Reel fixed both ends all LC	3b Struts bolted, reel sliding, friction coeff 0, relevant LC	3c Struts bolted, reel sliding, friction coeff 0.16, relevant LC	3d Struts bolted, reel sliding, friction coeff 0.3, relevant LC	4a Struts bolted, reel sliding, friction coeff 0, all LC	4b Struts bolted, reel sliding, friction coeff 0.16, all LC	4c Struts bolted, reel sliding, friction coeff 0.30, all LC
7	0.925	1.101	1.312				1.061			1.29	1.038	
8	0.976	1.151	1.267				1.112			1.245		
15	0.662	1.159	2.17				1.13			2.15	1.83	1.573
16	0.683	1.132	2.2				1.104			2.18	1.859	1.603
19	0.453		1.119							1.114		
20	0.439		1.105							1.098		
21	0.803		1.415							1.393	1.154	
22	0.821	1.02	1.391							1.362	1.122	
87	0.976	1.206	1.524				1.197			1.516	1.232	1.004
88	0.976	1.207	1.535				1.198			1.522	1.236	1.006
99	0.872	1.08	1.372				1.073			1.364	1.108	
100	0.869	1.077	1.371				1.07			1.361	1.105	
106	0.769		1.047			1.114	1.298			1.529	1.226	
107	0.752		1.066			1.141	1.327	1.022		1.52	1.215	
224	0.769					1.114						
225	0.752					1.141						

Unity checks for the baseline "as submitted" case are taken from the 2008 PH submission to ABSG. For the remaining scenarios, unity checks are only shown in the table if they are above 1.0

39 After considering Ms Renuka Devi's detailed explanation in court, I accept that the figures in scenario 4 *ie*, columns 4a, 4b and 4c in the Table above are inaccurate because they do not correctly simulate the actual forces on the structure in the as-built condition. Furthermore, all loading conditions (referred to as "LC" in the Table above) both relevant and irrelevant have been included in scenario 4. The right approach is to take into account only the relevant LC and model the actual forces acting on the actual as-built structure as follows under scenario 3:

(a) with "Struts bolted" (*ie*, correctly modelling fixed joints and not ball joints at the strut connections);

(b) with "reel sliding" (*ie*, correctly modelling the as-built condition where the 300 tons reel is free to slide along its shaft in the transverse direction),

where the coefficients of friction are respectively assumed to be "0" under column 3b (for a conservative design where friction forces are ignored), "0.16" under column 3c (for wet or lubricated steel to steel contact) and "0.3" under column 3d (for dry steel to steel contact).

40 In the opinion of Ms Renuka Devi, the coefficient of friction of "0.3" is inappropriate because of weather and wet conditions out at sea. Basically, if friction forces are to be taken into account for the structural analysis, then in Ms Renuka Devi's view, the appropriate value for the coefficient of friction is "0.16" and accordingly, column 3c of the Table is the correct column to look at and not the other columns.

41 In the Table produced by Ms Renuka Devi, a Beam Unity Check value that exceeds "1" indicates that the beam fails the unity check in terms of its structural strength. The beam designer therefore

has to redesign that beam to strengthen it. The higher the value is above "1", the greater is the inadequacy of that particular beam as a structural strength member. As can be seen, that critically depends on the correct choice of the appropriate coefficient of friction for the sliding surfaces.

42 Based on a friction coefficient value of "0.16", it appears that only one structural beam member marginally fails the Beam Unity Checks (*ie*, Beam member number "107" which has a unity check value of 1.022) as shown in column 3c of the Table. If one adopts a more conservative design where the effects of friction are ignored entirely as was done in the initial calculations, then many more structural beam members (*ie*, 10 beams) fail the Beam Unity Checks as can be seen under column 3b in the Table above.

43 I note that Ms Renuka Devi had opined that she would not normally take friction into account in her structural analysis. She said "*I mean, if the thing has been left to me, then I will not consider any friction, as a conservative design*". I agree with Ms Renuka Devi in this regard. It is a correct conservative engineering approach to take for the RDU design. I am of the view that having to rely on frictional forces as a restraining force to help to reduce the stresses on a ship deck mounted structure exposed not only to all the weather and sea elements, but also to the roll motion of a vessel out at open sea where rough sea state conditions can be expected is really scrapping the bottom of the barrel in terms of good engineering design for deck mounted equipment on board a ship such as the RDU. I would have expected instead to find adequate safety factors built into the RDU design (*ie*, with unity check values well below "1" for the structural beams) in order to provide for some over-design to cater for many potentially unknown and unexpected factors in the course of the actual RDU operations out at open sea. A practical engineering design for ship board equipment in my view should not normally be pushed to its engineering limits since weight is not as significant a consideration for design purposes on board a large ship unlike the design of a fighter jet where light weight is such a critical design consideration due to the need to achieve high speed, have a large fuel capacity for long range and yet be able carry a heavy weapons payload. Designing for stronger and heavier structural elements with sufficient safety factors should be the correct approach rather than one that results in an RDU design that teeters at its engineering limits, even when based on a 2.2 m/s^2 transverse acceleration, which by DNV standards is already far too low and unacceptable as a design criterion for portable offshore equipment.

44 For the reasons stated above, I find that the structure of the RDU is inadequately designed as the assistance of friction should be ignored and far too low acceleration values were used as design criteria by the Defendant.

Failure to design for fatigue

45 The Defendant admits that fatigue was not considered in the design of the RDU structure. The Defendant takes the view that it is not necessary to design for fatigue. As the RDU is subject to cyclical loading, the Plaintiff submits that fatigue ought to have been considered.

46 I agree with Mr Drago that it "*simply doesn't make engineering sense to design rotating machinery without considering fatigue*." The bearings and shafts in the RDU must be designed for fatigue as there will be several hundred thousands of hours of operation expected within the lifetime of the RDU. However, is the same applicable to the non-rotating structural parts of the RDU?

47 The Defendant says that the towers of the RDU were designed and manufactured in accordance with the AISC 360-05. As highlighted by Mr Moore, AISC 360-05 is a specification for structural steel buildings and Section 3.1 of Appendix 3 stipulates that an evaluation of fatigue must be made if the application of live loads is more than 20,000 cycles. I agree with Mr Moore that the

RDU will see more than 20,000 cycles of loading in its lifetime due to wave action on the vessel. The Defendant cannot pretend for the purpose of engineering analysis that the RDU is installed on land and not on board a ship. Accordingly, the RDU should have been assessed in accordance with the fatigue section of that code. Ms Renuka Devi also accepted that a check for fatigue should be carried out on the welding and connections of the RDU but the Defendant had not asked ABSG to do it. Dr Eccles and Mr Natarajan, testifying as experts on behalf of the Plaintiff, asserted that the connections are in many cases the most relevant areas to check for fatigue failure because they often fail first. I agree because welding connections typically have much lower fatigue strength than the underlying metal substrate of the structural members due to stresses from welding and heat distortion at the heat affected zone around the weld.

48 It appears that the Defendant took no account of the cyclical loading that the RDU structure would be subjected to. Dr Yang Ting had to concede that there was some "oversight" not to have considered fatigue in the structural design calculations due to cyclical loading. As such, structural failure due to fatigue may well occur within the operating lifespan of the RDU.

49 The fact that both the rotating and non-rotating parts of the RDU were *not* designed to withstand fatigue is yet another important factor that I have to take into consideration in my evaluation of the merchantable quality and fitness for purpose of the RDU.

50 Dr Yang Ting's explanation for the omission of fatigue analysis was that site-specific information on the load spectrum and the areas of operation of the RDU were needed. He said "*[b]ecause the customer cannot give you an input [on] how the load spectrum looks like. We cannot do the calculation.*" That to me is a lame excuse for a designer. The Plaintiff had informed the Defendant of one specific area of deployment of the RDU *ie*, Bass Straits. It is for the designer to research and find out for himself the load spectrum for the specified area of operation (or the equivalent load spectrum for similar areas of operation). It is not for the designer to complain that the customer did not provide the load spectrum information (which the designer in any event had not even asked his customer), or to simply ignore the engineering design requirement to perform the fatigue analysis merely because the customer had not volunteered any load spectrum information for that area of operation to enable the designer to perform fatigue calculations. If any design data is needed, I am of the view that the designer himself is responsible for finding out the required design data to enable him to undertake the fatigue calculations required for the RDU design. That is what the Defendant had been contractually engaged to do.

51 The Plaintiff urges me to infer that the Defendant deliberately omitted to ask ABSG to perform any fatigue calculations because the Defendant knew that such calculations if done would have shown up the structural design to be inadequate and ABSG would not have certified the RDU. At this stage, I am not prepared to make this inference although I am prepared to find a high level of technical or engineering incompetence on the part of the Defendant's designers not to have taken fatigue into account in the RDU design.

Failure to design for the effects of wind

52 Mr Loh, the General Manager (Engineering and Operations) of the Defendant, admitted that the Defendant should have checked for wind loading on the RDU structure but did not do so at the design stage. Ms Renuka Devi was of the same view that the effects of wind should be considered but she did not take that into account in her design review of the RDU as the Defendant told her that it was not necessary to do so.

53 The ABS Guide for Certification of Lifting Appliances July 2007 sets out at Chapter 2 the

requirements for the certification of, *inter alia*, heavy lift and gantry types of cranes installed on board vessels classed by ABS and requires data on the environmental loads on the crane structure including the effects of wind to be submitted together with other plans and supporting data for review and approval. Clearly, the dynamic wind loading is a design consideration.

54 As the RDU has to hoist a reel of some 300 tons upwards from the deck to the operating height prior to laying the umbilical, I consider the RDU to be akin to a heavy lift crane and the engineering considerations stipulated in the ABS Guide would be a useful benchmark for the engineering standards to which the RDU should have been designed. As the reel diameter is some 9.2 metres, the reel has a very big surface area subject to wind load. As such, I disagree that dynamic wind loading can be ignored in the design of the RDU.

55 I find that the failure of the Defendant to take account of wind load has contributed to the overall structural under-design of the RDU.

Bolting arrangements were under-designed

56 I accept the opinion of the Plaintiff's experts that the bolting arrangements for the following were inadequately designed:

(a) the bolts securing the struts to the main tower frame;

(b) the bolt connections between the pinion bearing housing and the sub-frame;

(c) the bolts securing the Sauer Danfoss gearbox to the motor mounting;

(d) the bolt connections between the pinion and gearbox flange; and

(e) the bolt connections for the end plates of the drive shaft.

57 The Plaintiff's Multimedia Bundle contains photographs and videos showing that loosening of the bolted joints had been observed. I also refer to an observation in a report on the tests conducted on 26 and 27 September 2012 titled "ATHK 300 T Reel Handler System – Load, Deflection, Alignment and Dimension" ("Nobletech Report"):

> Results – Deflections under full torque load, tower A, pay-out direction
>
> Observation
>
> Left side pinion bearing housing was seen to move when system was pulling on the steel rope when undergoing the torque load test. **The bolts holding the bearing housing were seen to be visibly loosening and at least one was sticking out by 2-3mm. A video recording was made to show the bearing housing moving**. This part of the test was aborted and one bolt was removed (by hand as it was very loose) and measured to be M16 diameter. The shank length was measured to be 30mm. Observation showed the bolt goes into an unthreaded ("free") hole in the bearing housing flange, which was about 15mm thick. After this, the bolt then threads into the steel behind the bearing cover, but as its shank is only 30mm long, and there are 6.5mm thick washers, only the final 5-8mm are able to thread into the hole and tighten the cover. [Emphasis added.]

58 The available video and photographic evidence and observations in the test report strongly support the theoretical analysis by the Plaintiff's experts that the bolting arrangements were in fact

under-designed. This is a serious defect because the structural integrity of the RDU depends on the bolts holding the various critical components of the RDU together.

Bolts securing the struts to the main tower frame

59 Mr Moore explained that bolted struts are not generally used for RDU construction as was done in the case of the Defendant's 300 tons RDU. According to Mr Moore, construction for this range of RDUs usually comprises an all welded or welded/bolted construction and uses much heavier section members. Although some manufacturers do employ the welded/bolted method of construction, however they are for lighter duty RDU applications, typically in the region of 80 tons to 150 tons. Mr Moore cites RDUs supplied by Aquatic Engineering & Construction (UK) and Sparrows Baricon (UK). Their RDUs in the 300 tons range differ insofar as they are of a more robust, all welded construction giving much greater structural rigidity than the bolted/welded construction of the Defendant's RDU.

60 I agree with the Plaintiff's submissions that although the bolting arrangements for the main struts are of critical significance to the structural integrity of the Defendant's RDU, yet the Defendant has not produced any evidence to show that they had carried out any calculations, *at the design stage*, to show the adequacy of these bolting arrangements. I draw the inference that no such calculations were in fact done *at the design stage*.

61 Dr Eccles carried out detailed calculations to check the adequacy of the bolted joints at the struts. He found that they were inadequate for both the tensile as well as shear forces. See Dr Eccles's AEIC from pp 109 to 125. Mr Huang on the other hand produced a set of calculations based on a pre-tension of 50% yield on four bolts of M30, each with a diameter of 30 mm, and a friction coefficient of 0.5. Mr Huang attempted to demonstrate that the maximum static friction generated was greater than the maximum slippage force of 353,009 N. However, Dr Eccles pointed out the erroneous approach in Mr Huang's calculations where the stress area of the cross section of the cylinder of the bolt was used when it should have been the actual thread stress area which is the relevant effective stress area. On re-calculating based on the same parameters assumed by Mr Huang, Dr Eccles explained at Exhibit P15 that slippage would occur:

> Based upon information provided in reference D13 (an email from Mr Huang Hai Tao to Mr Alan Hooper dated 8 July 2014) quotes the following:
>
> "The joint consists of 4 bolts of M30. Assuming a material yield strength of 550 MPa. Pretension to 50% of yield.
>
> Pretension = $0.25\times3.14\times30^2\times550/2$=194386N
>
> Friction coefficient = 0.5
>
> Max. static friction=$4\times1943860\times5$= 388772N > 353009 N => No slippage"
>
> The above calculation is based upon the bolt being a cylinder of diameter 30 mm in reality the thread of the bolt is weaker. The appropriate area to use for the thread is referred to as the thread stress area. For a M30 thread, the stress area is 561 mm^2. This value is used for M30 bolts and is quoted in Tab 9 of the confidential bundle, Clamping design and bolts calculation on page A132.
>
> Repeating the calculation using the stress area of the thread rather than the shank diameter of the bolt gives:

Pretension = 561 × 550/2 = 154275N

Friction coefficient = 0.5

Max. static friction=4×154275×0.5 = 308330N < 353009 N =>Slippage

62 Mr Hooper, the Defendant's expert witness, then pointed out that the yield strength of the bolts used was actually 640 MPa instead of 550 MPa. However, even with the higher yield strength figure, Mr Hooper basically accepted that this would still be "low" and in my view, inadequate from a design perspective if the necessary safety factors are to be included:

MR HOOPER:	The only comment I have is that Mr Huang has used a lower yield strength, the yield strength of the bolts is actually 640 Mpa instead of 550. Using these figures, even with the reduced bolt area, it comes out equal, not the same, it's slightly more than the slipping or the shear force.
COURT:	So it is slipping?
MR HOOPER:	It comes out at 359,000 Newtons, and the shear force is 353,000.
COURT:	Very close.
MR HOOPER:	Very close.
COURT:	But for the purposes of the design safety factors, it would not make it, basically?
MR HOOPER:	It would be -- yes, it's low.
COURT:	Inadequate?
MR HOOPER:	It's low.

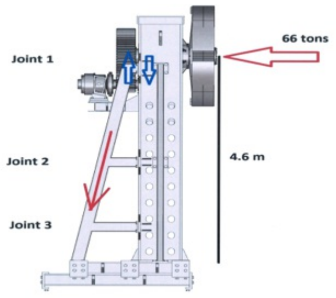
63 The second deficiency in Mr Huang's calculation is his adoption of 0.5 as the value for the coefficient of friction on the basis that the bolted joint is not painted. However, the bolt joint areas were in fact painted and the appropriate coefficient of friction for a painted joint is much lower. According to Dr Eccles, it is between 0.2 to 0.3 according to a research paper by Heistermann, C., *Behaviour of pre-tension bolts in friction connections, Lulea University of Technology, 2011*. Mr Huang finally had to concede during the hot-tubbing that with a painted joint, the coefficient of friction of 0.5 that he had assumed could not be achieved. In short, the bolted joints would slip and come loose.

64 Dr Eccles also pointed out a third error in Mr Huang's calculations. Mr Huang wrongly assumed that the bolt was fully threaded into the joint, but that does not appear to be the case in the design of the bolted joints. It can be seen from the engineering drawings that there would not have been full engagement of the entirety of the thread length (see Confidential Bundle at A127).

65 The effect of the failure of the bolts was explained by Dr Eccles and Mr Natarajan as follows:

DR ECCLES:	Your Honour, I have got some sketches to show you the implications of that.
COURT:	Okay.

DR ECCLES:	These struts are vital for the structure. If they slip, the bolts will come loose because of the slippage. Besides, the paint would be disintegrated because of the movement underneath the bolt and within the joint interface, leading to the structural integrity of the unit being jeopardised.



COURT:	The slippage you are talking about, is it joint 1, joint 3 or joint 2?
DR ECCLES:	It is potentially where Mr Huang has taken his numbers from.
COURT:	Which joint is it?
MR HUANG:	At the top.
COURT:	Joint 1?
MR HUANG:	Joint 1.
COURT:	What about joint 2 and joint 3, any slippage problem possibility?
MR NATARAJAN:	Your Honour, if joint 1 slips, from a structural point of view, there is no rigidity in the system, it loses the stiffness of the struts, so all of the load will go to strut joint 2, then that will go bust, then following on from that, it will be joint 3. So I think there is no transfers and longitudinal stiffeners in the system on the structure, for the record.
COURT:	Mr Huang?
MR HUANG:	What I want to say is this calculation just proves that our friction is not sufficient to take the shear force if a friction coefficient is taken as 0.3. If friction is not sufficient, then both will take the shear force. If both are strong enough to take the shear force, the joint will not slip.
DR ECCLES:	Can I come back on that, please, your Honour?
COURT:	Yes.

DR ECCLES:	Obviously, we have made various assumptions before, for example that the bolt is not fully threaded into the plate anyway, and so you could not develop anywhere near the loads you have assumed. Secondly, as you start to slip, the paint would disintegrate within the joint surfaces, the bolt would rotate, and it's likely the bolts would drop out of the structure. You are allowing movement of the structure inside the clearance holes, so the structure would be moving up to whatever the clearance holes held, and so it would be flapping away until -- well, my guess, until it fails by fatigue. So there's no structural rigidity there because the joint is moving, it's a friction grip joint.
MR NATARAJAN:	Your Honour, if there is going to be slippage, we shouldn't really be considering that as part of the stiffness into the system at all, in the structural analysis. That would be incorrect.
COURT:	Yes, Mr Hooper?
MR HOOPER:	I didn't understand what Mr Natarajan said, sorry, I was thinking of something else.
MR NATARAJAN:	Alan, what I'm saying is if this is going to be slipping, the stiffness in the system is not going to be from the strut, it loses its stiffness.
MR HOOPER:	I understand, sure. That's understandable because the structure is not there.

66 On the totality of the expert evidence, I find on a balance of probabilities that the bolting arrangements for the RDU struts were inadequately designed. I accept the explanation provided by Dr Eccles and Mr Natarajan on the severe implications arising from a potential failure of the bolting arrangements on the strength and integrity of the RDU structure as a whole, bearing in mind that the RDU has to carry a massive 300 tons reel and may be subjected to high wind load and considerable inertial forces due to wave action depending on the kind of sea condition encountered during the umbilical laying operation.

67 The Defendant’s only response to the Plaintiff’s submission that the bolting arrangements for the RDU struts were under-designed is simply that Dr Eccles’s calculations are unreliable because Dr Eccles used data from Mr Natarajan’s FEA analysis, which the Defendant submits is unsound. Since I have found to the contrary at [145] and [151] that Mr Huang’s FEA analysis is unsound and have accepted the results of Mr Natarajan’s FEA analysis instead, this puts to rest the Defendant’s argument that Dr Eccles’s calculations are unreliable. In any event, Dr Eccles had also analysed the adequacy of the bolts for the tensile stress independently of Mr Natarajan’s report. Dr Eccles re-calculated using the tensile forces that the Defendant’s own experts had assumed to be acting on the bolts and demonstrated that the bolting arrangements and the strut bolts were nevertheless still under-designed.

68 The Defendant in its closing submissions simply attached a further report from Mr Huang dated 1 October 2015 at Appendix E in a final desperate attempt to show that Dr Eccles’s analysis is erroneous. At the conclusion of the trial, I did not ask for any further report on bolt analysis by any expert from either side. The Plaintiff strongly objects to its introduction as further evidence. As the

trial is over, the Plaintiff no longer has any opportunity to bring further expert evidence to rebut the further report from Mr Huang or to cross-examine him on it. As the Plaintiff will be prejudiced, I am not admitting the further report of Mr Huang as evidence.

Bolting arrangement on the pinion bearing joint

69 The pinion gear is designed to drive the main gear. It is connected to the sub-frame housing by a bearing. At the other end, it is bolted to the Sauer Danfoss gearbox. The gearbox is itself mounted by a bolting arrangement. The pinion bearing is attached to the structure through an arrangement of six M-16 bolts, each 30 mm long. As the grade of the bolts was not specified in the drawings, the Defendant's practice according to Dr Yang Ting would be to use a bolt grade of 8.8.

70 Dr Eccles explained in detail in his AEIC why he considered the bolting arrangement on the pinion bearing joint to be inadequate. Mr Hooper confirmed that the bolts would slip on the design load:

DR ECCLES:	Yes, your Honour, we have discussed. Although we have differences in terms of loading, the bolt load, we are in agreement that the joint would slip on the design load, as designed.
COURT:	Fine.
MR HOOPER:	That's confirmed.

71 Mr Hooper further accepted that the under-design of the bolting arrangements at the pinion bearing housing would result in misalignment of the gears and an increase in the loads and stresses on the gears, which I am inclined to believe may also cause premature gear failure due to the increased gear loading:

DR ECCLES:	So once the joint slips, it would put the pinion out of align with the gear, which would – obviously what forces are generated then, it would be higher. But then that would lead to vibration, increased forces on everything else, and I believe personally led directly to the motor falling off.
COURT:	Do you agree with that, Mr Hooper?
MR HOOPER:	If the pinion bearing housing does slip, yes, there is misalignment on the pinion gear. No question about that.
COURT:	Then the rest of it follows, the gears will break after that?
MR HOOPER:	Depending on the load, your Honour. It may have run like that for quite some time.
COURT:	After it's misaligned, then you keep on running, even if it doesn't break the first time around, after a while, it would break.

MR HOOPER: It appears like tower B continued like that for several days, because they continued the rest of the lay operation with these pinions misaligned, because they had tightened up the bolts and I'm sure they didn't do any realignment. So the problem may have occurred in both areas, so the unit would still have continued under operation, depending on the load. Undoubtedly, there would have been more load on certain sections of the gears. There's no question about that.

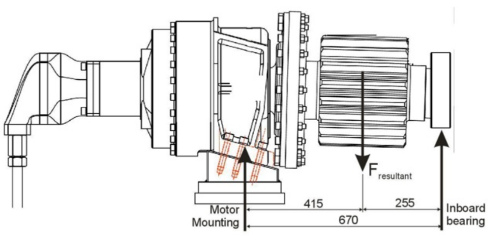
72 The Defendant's answer to this in its submission is that if there was indeed any misalignment of the gears due to slip, then uneven wear patterns across the gear teeth would result. But there was uniform burring found across the teeth of the gear wheel and pinion gear of Tower A of the RDU during the investigations following the failure thereby indicating that there had been no misalignment and hence no slippage at the pinion bearing housing. The Defendant further highlights that prior to the catastrophic failure of the RDU on 20 May 2009 (see [201]), a successful full torque test during the FAT was carried out on 28 February 2008 and the RDU had also finished laying umbilical No. 1 and completed a partial laying of umbilical No. 2 during the Longtom project. The Defendant is thus suggesting in its submission that the successful torque test and the completion of laying of more than one reel of umbilical demonstrate that the RDU is of merchantable quality and fit for its purpose.

73 The Defendant wrongly assumes that the RDU was operated to its design load during the course of the Longtom project for a sufficiently long period of time prior to the catastrophic failure, which therefore proves *that at the design loads*, the RDU is still of merchantable quality and fit for its purpose. However, there is no evidence that the RDU had *in fact* been operated to its full design load during the relatively short umbilical laying operation prior to the catastrophic failure. Dr Eccles's theoretical analysis of the slippage of the bolts that hold the pinion bearing housing in place was based on the full design load condition. If the RDU had *not* been operated to its full design load condition for a sufficient period of time out at sea, I would not at all be surprised if serious slippage and misalignment (as predicted by the theoretical analysis) had not occurred yet. The fact *per se* that there was no uneven wear found on the gears (even if true) is not sufficient for me to reject the conclusions of both the Plaintiff's and Defendant's experts, *ie*, Dr Eccles and Mr Hooper respectively, that they were "*in agreement that the joint would slip on the design load, as designed*."

74 As I have no good reason to disagree with the conclusions of the experts from both sides that the bolts would slip on the design load, I accordingly conclude that the bolting arrangement on the pinion bearing joint was under-designed.

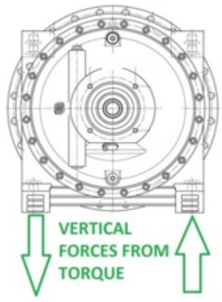
Bolting arrangement on the mounting of the gearbox and the pinion/gearbox flange interface

75 The pinion is connected to the output flange of the Sauer Danfoss gearbox by a series of bolts. The gearbox is itself located on a motor mounting as follows:

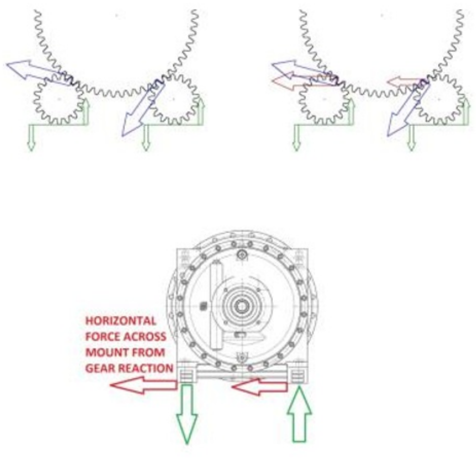


76 The Defendant submits that the Sauer Danfoss gearbox mounting was not under-designed because the Defendant had followed the manufacturer’s specification by using three M20 bolts of a higher grade 12.9 on each side of the mounting and by tightening them to the specified torque of 540 Nm. However, no records were produced to support the Defendant’s assertion that these bolts were in fact tightened to the torque of 540 Nm at the time of manufacture or testing prior to the delivery of the RDU to the Plaintiff.

77 In any event, the main question that has not been answered by the Defendant is what are the design parameters and load assumptions used by the manufacturer for the Sauer Danfoss drive unit and gearbox for its specification of those bolt sizes, grades and tightening torques, and whether the load assumptions of the manufacturer exceed those loads computed for this RDU design where the Sauer Danfoss drive units and gearboxes are being used for a totally different purpose (*ie*, to drive a 300 tons reel and not a much smaller and lighter cement mixer drum). Furthermore, the manner in which the Sauer Dandfoss drive unit and gearbox is mounted on a road-going cement mixer truck to turn the cement mixer drum is very different from that for the RDU. There is a very substantial eccentricity, non-existent for the cement mixer truck, but present in the RDU, which creates very significant shear loads for the mounting bolts as can be seen in the illustrations below. I accept the Plaintiff’s submissions that the Sauer Danfoss drive units and gearboxes were never designed to take these additional large loads present in the RDU at its design loads.



Vertical Forces from torque only in the case of the Cement Mixer



Additional Horizontal Force across the mount from the gear reaction in the case of the RDU

78 Without knowing the manufacturer's design parameters and load assumptions for comparison with the expected loads arising from the RDU operation, I do not accept the Defendant's bald assertion the Sauer Danfoss drive unit and gearbox mounting is correctly designed merely because the Defendant had adopted the manufacturer's specifications for the mounting bolts and tightening torque requirements.

79 Dr Eccles explained in his AEIC why the bolts at the Sauer Danfoss gearbox mountings were under-designed. Considering the maximum working design tension of 15 tonnes for the umbilical, the shear force generated is enormous and far exceeds the capacity of those bolts to restrain and the joint would slip. Slippage, even by fractions of a millimetre, would be sufficient to loosen the bolts, induce fatigue loading and reduce its operating life.

80 Mr Hooper did not challenge Dr Eccles on this point but took the position that in the case of the design of the foundation at the base of the Sauer Danfoss gearboxes, restraining keys or stoppers were in fact put there to take the shear load that Dr Eccles was referring to but "*unfortunately, there are, from what we can see, gaps between those stoppers and the gearbox*".

81 It appears to me that even if the design intent was to have the restraining keys or stoppers (instead of the bolts) absorb the very high shear loading, the fact that there were actual gaps present, as acknowledged by Mr Hooper, means that they were not effective as stoppers and the whole design intent is not achieved rendering the potential of slippage and fatigue of the bolts at the Sauer Danfoss gearbox mounting, a real potential problem as highlighted by Dr Eccles.

82 I turn next to consider the evidence of Mr Drago that the design of the interface between the Sauer Danfoss gearbox and the pinion was defective. As can be seen in the diagram below "Figure 4 – Drive Unit to Final Pinion Assembly and Pinion Mounting" reproduced from the AEIC of Mr Drago, the final drive pinion flange "Arrow 3" on the pinion gear "C" is mounted to the output flange "Arrow 2" of the Sauer Danfoss gearbox "B" by a series of bolts.

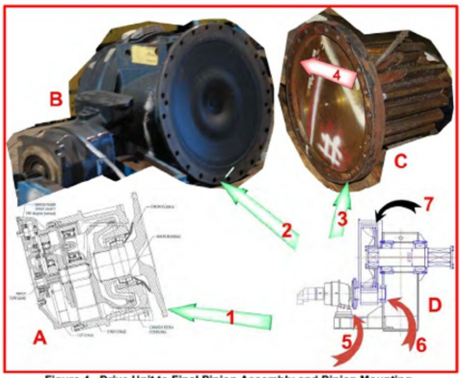


Figure 4 - Drive Unit to Final Pinion Assembly and Pinion Mounting

83 Mr Drago explained at p 65 of his AEIC that:

> While there appears to be a "pilot" bore in the pinion mounting flange, there is NO matching "pilot" journal on the Sauer Danfoss Drive unit. Without a mating pilot journal there is no mechanism to insure that the pinion axis and the Sauer Danfoss output shaft axis are coincident. This lack of concentricity control results in pinion runout in operation which in turn, results in varying stresses on teeth around the circumference of the pinion during operation and misalignment across the face width. The clearance in a typical bolt hole is much larger than the

allocable pitch diameter runout of the gear set thus the effective alignment of the gear set will be severely compromised. **This condition will be very detrimental to the load capacity and life of the final drive pinion – gear set. This is a very serious gear system design error.**

[Emphasis in original]

84 Mr Hooper attempted to demonstrate that the presence of a spigot on the flange of the pinion was sufficient to cure the problem. However, the following discourse at the hot-tubbing eventually led to Mr Hooper conceding that the spigot itself had a clearance of a maximum of 0.2 mm and with such a clearance, there was definitely allowance for some movement:

MR DRAGO:	Can you point to the spigot you are talking about?
…	
Mr HOOPER:	This spigot is right here. Is that clear?
MR DRAGO:	Where does that pilot?
MR HOOPER:	That pilots on to the outside diameter of the flange of the gearbox.
MR DRAGO:	Point to the dimension for that flange, for that pilot.
MR HOOPER:	It's 530 plus 0.2 plus 0.1.
MR DRAGO:	**That little flange there is taking that shear load?**
MR HOOPER:	Yes.
…	
COURT:	**Before you go to the page, I will check with Mr Hooper: The spigot has clearances?**
MR HOOPER:	**Yes.**
COURT:	**How wide is this clearance?**
MR HOOPER:	**Maximum 0.2mm.**
COURT:	Is 0.2mm <, from the point of view of the bolt there, such that it can enable the slipping to occur, 0.2mm?
DR ECCLES:	Yes, your Honour. Potentially, what would happen -- I'm saying "potentially" -- at 0.2 of a millimetre, the bolts would bend. Whether they slip or not depends upon the friction coefficient you assume. Though the bolts would bend backwards and forwards and fatigue.
COURT:	So ideally -- ideally -- the bolts themselves, on tensioning, should have enough capacity to ensure there's no slip. Correct or not, Mr Hooper?
MR HOOPER:	Correct.
COURT:	Because your spigot -- unless, of course, it's interference fit, and it's a very strong spigot, I can accept. But now you tell me there's a clearance of 0.2mm, which is not small. Definitely then you have to rely on the bolts to ensure there's no slippage. But then in which case, you are again relying on the situation where, after you slip, oh, the spigots come to play. Would I be right, Mr Hooper?

MR HOOPER:	**If you've got 0.2mm clearance, there is allowance for some movement. Definitely, yes.**
COURT:	**And if the bolts are not strong enough -- I mean, is it good enough for the design?**
MR HOOPER:	**Well ...**
COURT:	**Okay, I understand that. Mr Hooper smiled.**
MR DRAGO:	Not in my opinion. I just did a quick calculation. If you look at the concentricity of that journal and the tolerance on that journal, that allows about 8/1,000ths. That would imply that the mating part has about the same amount. That would be 16/1,000ths of an inch. That's again, on that pinion, even for that low quality, about ten times the allowable run-out.
COURT:	Mr Drago uses English measurements. 16/1,000ths of an inch. That is, what, 16 divided by 1,000 times one inch is 3-point -- I mean, I've forgotten my conversion.
MR DRAGO:	I think it is about 0.4mm.
DR ECCLES:	0.4.
COURT:	I'm more used to the millimetres. 0.4 does not seem big.
DR ECCLES:	**It's big for the bolts**, your Honour. I think it's big for the gears as well, but I'm just speaking for the bolts.
MR DRAGO:	**It's huge for the gears**; it's big for the bolts.

[Emphasis added in bold]

85 I note that Mr Hooper smiled and hence, tacitly acknowledged that the problem of alignment to achieve concentricity alluded to earlier by Mr Drago remained because of the clearance at the spigot. Accordingly, the design does not appear to be adequate.

86 I agree with the conclusion of Dr Eccles and Mr Drago that as a result of the potential movement arising from the 0.2 mm clearance at the spigot, the potential misalignment in concentricity at the interface between the Sauer Danfoss gearbox and the pinion flanges is unacceptably large for the bolts and the gears. After considering the totality of the evidence of the experts, I find that the bolting arrangements on the mounting of the Sauer Danfoss gearbox and the pinion/gearbox flange interface were both inadequately designed.

Bolting arrangement on the shaft end plates

87 The shaft end plates provide attachment of both the main gear and the hub drive insert to the main driveshaft between the gear and the reel hub. Ten M20 x 55 mm, grade 8.8 bolts attach the plate to the hub/gear, with a further ten bolts of the same specification attaching the same plate to the shaft.

Gear side of Tower B

Real insert side

88 Dr Eccles explained in his AEIC from pp 9 to 33 why the bolting arrangements on the shaft end plates were inadequate. At the hot-tubbing, Dr Eccles calculated that if the brakes were applied to slow down the 300 tons reel spinning at 0.29 rpm, corresponding to a lay speed of 500 m per hour, it would cause the bolts at the drive shaft end plates to be overloaded. With the clearances present, measuring about 1.3mm to a maximum of 2.5mm, between the hexagonal shaft and the gear on the one end and the reel insert on the other, the bolts would come loose. Mr Hooper disagreed with Dr Eccles and asserted that the hexagon in the hexagonal shaft took the shear load/torque transmission and not the bolts which merely held the gear hub in place. However, the hexagonal shaft would be able to take the shear load only if the clearances were absent and the hexagon fitted tightly within the hub. Mr Hooper then speculated that "*it may have been that on the end of the hexagon, there was a slight taper put on in order to get the hub to fit on..*.", which then took up all the clearance on the hexagonal section.

89 Mr Moore then pointed out that the design drawings made no mention whatsoever of any taper on the hexagonal portions of the drive shaft or any taper on the gear or the hub insert.

90 Mr Hooper then offered another explanation that even if there were clearances and the bolts loosened and slip occurred, the hexagon would still take over to bear the torque loading and "*the hexagon would wind up on the hexagon on the shaft and become very, very tight*."

91 However, I note that the RDU is designed not only to rotate the 300 tons reel to pay out the umbilical, but it can also rotate the reel in the opposite direction to wind back or pull in the umbilical. There can also be a sudden emergency stopping of the reel, whilst it is rotating in the clockwise or anti-clockwise direction. Given the manner in which the reel can be operated, it is therefore wrong to assume that the hexagon will always "*wind upon the hexagon on the shaft"* and therefore, the bolts will not loosen thereafter because they no longer bear the torque loading:

COURT:	Mr Hooper, would you say, since the machine has been designed also for the reverse, and if you are carrying, let's say, a best case scenario, an empty reel, you do the reverse to reel in, and suddenly there's a stop, same problem? It would jerk the other way?
MR HOOPER:	You would jerk the other way, yes.
COURT:	And so after that, you still think this is a design which is fit based on only the hexagonal part, never mind the rest of the bolts; you have others to look at.
MR HOOPER:	All I can comment on is that the hexagonal drive is there for a purpose of transmitting such a large torque, and the torque is considerable.
DR ECCLES:	Why was it designed in the manner it was?
MR HOOPER:	You have to ask the defendant that. It's an unusual arrangement, granted, but it has enormous capacity to transmit a torque.
DR ECCLES:	Not through the bolted joint.
MR HOOPER:	The bolted joint, as I said, was never intended to transmit the torque. The unfortunate thing is that the defendant had put bolts into the gear wheel as well as into the shaft. They could well have done without the bolts in the gear wheel and only had the bolts in the shaft.

92 I accept the explanation of Mr Hooper that the hexagonal drive would be able to transmit the very large torques generated especially during the braking of the 300 tons reel without the need for any assistance from the bolts. Since the Defendant had put bolts in both the gear wheel as well as the shaft, these bolts were made to bear these large torques unnecessarily and they would slip and come loose because they would not be able to take the shear loads. The shaft end plate might well come loose and fall off. As this is a rotating part, the bolting arrangement for the shaft end plates does not appear to me to be a safe design from this point of view.

93 The Defendant refers to the Plaintiff's Multimedia Bundle/Photo set 001.pdf and points out that after the repairs were completed on the RDU, the photograph of the new shaft end plate installed shows only a single circle of bolts *ie*, those engaged into the shaft end. The Defendant then surprisingly asserts in its submission that the design intent from the onset was for the torque in the main shaft to be transmitted through the hexagon fit and not through the bolts. I do not believe that this was the design intent from the beginning. It is only with hindsight that the Defendant realised the problem in its original design and then redesigned the bolting arrangement to allow the hexagon fit to take the full torque load instead.

Gears were under-designed

94 Mr Drago carried out a detailed analysis based both (a) on the full rated torque of the motor/gearbox of 61,200 Nm for the non-constant tension mode; and (b) on what he termed the "*system's rated constant-tension load…*" of "*…15 tons of pull at 9.2 meter diameter for the entire system*" in order to assess the "Service Factor" for each of the seven fatigue life conditions for the pinion and gear of the RDU. Mr Drago summarised his results in the following table for the RDU Pinion and Gear in both the Constant Tension Mode and the Non-Constant Tension Mode at Figure 8 at p 52 of his AEIC:

Constant Tension Mode		Operating Conditions Well Aligned Open Gear Set, K_m =1.652		Non-Constant Tension Mode	
Pinion	Gear	Service Factor	Fatigue Life	Pinion	Gear
0.709	0.310	Pitting Service Factor	50,000 Hours	0.565	0.247
0.540	0.751	Bending Service Factor		0.431	0.598
0.848	0.370	Pitting Service Factor	10,000 Hours	0.676	0.295
0.569	0.806	Bending Service Factor		0.453	0.642
1.099	0.481	Pitting Service Factor	1,000 Hours	0.875	0.383
0.640	0.915	Bending Service Factor		0.510	0.729
1.187	0.518	Pitting Service Factor	500 Hours	0.946	0.413
0.664	0.948	Bending Service Factor		0.529	0.755
1.422	0.620	Pitting Service Factor	100 Hours	1.133	0.494
0.724	1.034	Bending Service Factor		0.577	0.824
1.537	0.629	Pitting Service Factor	50 Hours	1.225	0.501
0.753	1.074	Bending Service Factor		0.600	0.856
1.659	0.629	Pitting Service Factor	5 Hours	1.322	0.501
0.850	1.178	Bending Service Factor		0.678	0.939

95 Mr Drago explained the significance of evaluating the Service Factor "*defined as the non-dimensional ratio of the inherent load capacity of the gear set divided by the actual applied load*" at p 51 of his AEIC:

> Where the Service Factor is equal to unity, the gear set will achieve the required life. Where the Service Factor is less than unity, however, the required life will NOT be achieved. Similarly, where the Service Factor is greater than unity, the required life will be exceeded. In general, for an application such as this we would recommend that the gears be designed such that the Service Factor in durability (Pitting) is at least 1.5 and that the Service Factor for Strength (Bending) is at least 2.0 at the design fatigue life. The greater Service Factor for strength relative to durability is deliberate so that any anomalous conditions that adversely affect the gear system will result in gradual surface degradation of the gear teeth (surface pitting), which will allow the machines to continue operating while corrective measures are taken, rather than resulting in tooth breakage which would cause almost immediate cessation of functioning of the machine.

96 Mr Drago concluded at p 60 of his AEIC that:

> In view of the Service Factor summary presented in Figure 8, it is clear that, even under "average industrial" alignment conditions, the subject final drive gear set is inadequately designed for this application. The pinion bending strength Service Factor does not reach a value of unity even at a fatigue life of just five (5) hours at either the constant tension or non-constant tension operating conditions. A fatigue life of less than five hours is simply unacceptable for this application.

97 In any event, Mr Hooper basically accepted that the design of the gears was flawed:

COURT: The next point we can move to after the bolts would be?

MR HOOPER:	It would be the gears. You asked me to get in touch with Mr Drago. I have contacted Mr Drago. We went through the analysis that we had presented and he discussed his analysis. What he has pointed out is that we are perhaps a little bit optimistic in our assumption of alignment. He has been more realistic than we have. There are also differences between the hardness that we used and the actual hardness of the material, which contributes other factors. We have also assumed a certain fillet radius. Mr Drago has carefully examined the actual tooth profile and the fillet radius is different to what we have assumed. All these factors affect the design and performance of the gears. There is also the material grade, which we can't clarify with him, but he is using the actual material grade that was used in the pinions. **If we were to readjust our calculations to take these factors into account, we would come up with a similar result to Mr Drago -- maybe not exactly the same numbers -- but the gears would not be quite sufficient for what they are intended to do.**

[emphasis added in bold]

98 It is clear from the totality of the expert evidence that the gears for the RDU were under-designed. I also note that the gears are essential operating components of the RDU. Both of these are important factors in my consideration whether the RDU as a whole is of merchantable quality and fit for its purpose.

Braking system was under-designed

99 The video presented at the opening of the Plaintiff's case illustrates the dire consequences of a reel spinning out of control. Any reasonable designer will consider this to be a grave risk requiring careful, cogent and complete mitigation. The adequacy of the design of the braking system for the RDU is thus critical.

100 However, the Defendant did not carry out any calculations for its design of the braking system:

COURT:	Let's confirm that; have there been calculations done on the braking system?
MR HOOPER:	I would have to check --
COURT:	By the designer?
MR HOOPER:	I haven't looked at it.
COURT:	You have been given all the papers; right? You have studied them. Can you confirm for me one way or the other whether such calculations have been done? If they have not been done, we are going to do it.
MR HOOPER:	**There is nothing in the design information submitted to ABS, no.**

[emphasis added in bold]

101 The design of the mechanical brakes appears to be flawed. The mechanical brakes do not allow for a gradual increase in braking force. They simply spring into place the moment the hydraulic pressure is dropped below a particular pressure.

102 It is not known specifically what that pressure is. Mr Hooper's evidence was that the brakes would be applied when the pressure dropped to 500 psi or below:

MR HOOPER:	…But this system, the brakes require approximately 500psi to release. That 500psi may be somewhere close to the tension that the plaintiff was trying to keep, so it's hovering right on the borderline of its operating window.
…	
COURT:	So when you release the brakes, then it goes into the constant tension mode. Do you mean to say you release the brake completely and then you go into the constant tension mode through the hydraulics?
MR HOOPER:	These brakes are either on or off.
COURT:	I see. There's no in between?
MR HOOPER:	There's no in between.
COURT:	I understand now. So these brakes are not the soft kind of brakes that you -- (unclear – simultaneous speakers)
MR HOOPER:	No.
COURT:	They are hard?
MR HOOPER:	Yes.
COURT:	They are either all on or all off?
MR HOOPER:	All on or all off.
COURT:	Nothing in between?
MR HOOPER:	Yes.

103 Mr Hooper then explained that the mechanical brake was simply designed as a parking brake and was not intended to stop a rotating 300 tons reel:

MR HOOPER:	Your Honour, what you have in your car, to use that as an example, is what you might call a dynamic brake. It's intended to stop the car from travelling at a high speed to zero speed.
COURT:	Without the brakes flying off.
MR HOOPER:	Without the brakes flying off. That's what we call a dynamic brake.
COURT:	What is this?

MR HOOPER:	**This is what I mentioned yesterday we term a parking brake. It is not intended to stop the reel when it's rotating**.

[emphasis added in bold]

104 I note however that there is no direct evidence to suggest that this was in fact the design intent of the Defendant. More importantly, Mr Hooper overlooked the fact that there is already an emergency stop function provided on the control panel of the RDU, which drops the hydraulic pressure to zero, and the mechanical brakes are immediately engaged to stop a rotating reel. Mr Moore said:

MR MOORE:	Your Honour, sorry to interrupt, what I think is also pertinent is this needs to come to a halt in an emergency situation where the emergency stop button is initiated. Obviously, that needs to be taken into account in the calculation as well because in that instance, the operators won't necessarily have the time to reduce the speed gradually.

105 The Plaintiff submits that absent a designed safe stopping process, pressing the emergency stop button would cause the RDU to self-destruct. This is a highly unsafe design. If the Defendant had tested the emergency stop function with an actual 300 tons rotating reel, as required by the ABS Guide for Certification of Lifting Appliances (see Chapter 2, Section 5 Testing of Cranes), this deficiency in the braking system would have become immediately apparent during testing. Such a test was however not performed.

106 No parameters whatsoever were provided in the operations manual as to the appropriate angular decelerations in order to stop the 300 tons rotating reel. All the manual says is this:

> Always start and stop the spooler tower by activating the joystick gradually. Do not attempt to accelerate or decelerate the spooler tower rapidly.

107 The Plaintiff submits that this is woefully insufficient. A reasonable designer would have provided specific operational parameters, and provided ways of ascertaining the angular velocity of the reel. Indeed, the design of the operations console makes it next to impossible for an operator to know the angular velocity of the reel.

108 From what Mr Hooper had described, I find that the design flaws in braking system create obvious and serious risks:

MR HOOPER:	In constant tension mode, if the operator accidentally knocked that lever back to neutral or inadvertently moved it back to neutral, the brakes will come on.
COURT:	That's pretty dangerous.
MR MOORE:	My question on that would be what warnings have been put on the unit, either on the control panel or the operator's manual, to inform the operator of that potential danger?

COURT:	Because you have to tell him, you cannot – normally in a car, you have gears, even going on to the next one, from neutral to drive, you have to consciously do it. You have to put a red button somewhere, something that "This is critical, you cannot do it, you must not do it, maybe only in extreme emergency," otherwise you only use the hydraulic brakes. You cannot have this thing coming suddenly throughout the operation. Especially when you say the control is only in the centre portion, that's all the worse.

109 Mr Hooper responded that the Defendant had "*trained the operators on how to use the equipment*". He quoted an extract from the manual (see [106] above) and said that the manual has clear instructions on how to operate the RDU.

110 I agree with the Plaintiff's submissions that:

(a) Mitigating a risk of this nature by training alone is insufficient. These deficiencies ought to have been designed out, and not left subject to human error.

(b) In any event, this does not answer the point that the mechanical brakes are also supposed to be able to stop the rotating 300 tons reel in an emergency. It makes no engineering sense to include as an emergency braking system a brake which cannot be safely operated when the reel mounted on the RDU is still rotating. Put another way, an emergency (stopping) brake which can only be operated when the reel is already stopped is quite pointless.

(c) Finally, the Defendant failed to provide any documents setting out the content of the training given to the Plaintiff's operators which provides the evidential basis for Mr Hooper's assertion that the operators were trained in respect of the proper operation of the RDU, for instance, training on the intricacies of how to safely apply the emergency brake, how to avoid excessive angular deceleration of the reel (with maximum deceleration values being stipulated) and how to control the lever in the constant tension mode to avoid any accidental knocking of that lever to the neutral or inadvertent movement of the lever back to neutral, which will abruptly apply the brakes on the rotating reel with disastrous consequences.

111 Another aspect of concern is the capacity of the brakes themselves. When Dr Eccles was informed by Mr Hooper during the hot-tubbing that the maximum torque capacity of each mechanical brake was approximately 1.5 times the motor torque, he opined that the bolts and gears would be overloaded by the braking torque. Dr Eccles pointed out that there would be a tremendous amount of energy in the system of some 2.9 MJ at the reel's maximum operating speed of 1.9 rpm and if the brakes were to absorb that amount of energy, they could well burn out and break:

COURT:	Then let me hear both sides and see what the calculations show.
DR ECCLES:	I've been looking at the condition you asked us to look at, when the reel suddenly stops, and the forces generated. Based on our calculations, what we believe would happen is that the brakes applied –
COURT:	Brake, you mean the mechanical brake?
DR ECCLES:	The mechanical brake is applied which --

COURT:	The highest --
DR ECCLES:	1.5 times the torque of the motor, which certainly my case -- I didn't realise that was the case. My calculations had assumed just the motor torque. Now I find the 1.5 potentially times that. But that's a separate point which I can address when I come to the work which I've completed. Based on the moment of the inertia, dependent upon obviously the rate of lay, but at a lay speed of 500m per hour, which is 0.29rpm, calculate that, it's approximately 14 seconds to stop by the -- essentially, what would happen, it would be the gear train would drive -- would provide the torque from the brake. But in the meantime, certainly, I didn't realise, that 1.5 times loading would overload the bolting system, would overload -- I believe I'm correct in saying the gears as well, because the torque would be so high. So in my particular -- obviously, I've just been considering bolting. It would overload a number of joints, which I can illustrate in a moment. But if it was being laid at 750m per hour, it would take approximately 21 seconds to stop. At the maximum speed of 1.9rpm, we calculate it would take 94 seconds to stop the drum rotating based upon the brake. However, we are concerned about that, because we've calculated at that RPM, the energy contained in the drum would be 2.9 megajoules which would -- the brake would have to absorb as a large amount of energy and it could well burn that brake out. So obviously our calculations are on the assumption that the brake can absorb that. But obviously, we've not had time to establish whether that's the case, but we are somewhat concerned that the brakes could actually -- well, the brakes could break.

112 I note that Mr Hooper did not dispute what Dr Eccles had stated above. Although during the hot-tubbing, the manufacturer's specifications of the mechanical braking capacity of the system became of much interest and Mr Hooper said that the Defendant was arranging to obtain it, the Defendant subsequently did not furnish the manufacturer's specifications for the brakes to prove that the brakes have the capacity to absorb at least 2.9 MJ of energy present in a 300 tons reel rotating at 1.9 rpm. The Plaintiff accordingly submits that the proper inference is that the brakes are themselves unfit for purpose.

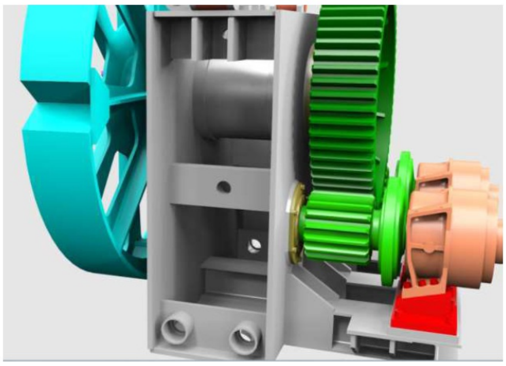
113 I therefore conclude on a balance of probabilities that the braking system as a whole was improperly designed. Indeed, as Mr Moore had pointed out, the Sauer Danfoss drive system and gearbox is designed to be utilised on a road-going cement mixer truck, an entirely different application from a 300 tons offshore RDU which the Defendant had adapted it for. In the circumstances, it is clear to me that the brakes designed for the RDU were deficient:

MR MOORE:	Your Honour, could I just add as well, these brakes that the defendants have used to create an effect to the reel, these brakes, as I have said before, the gearboxes are actually designed to use on a road-going cement truck, which the maximum weight of the drum would be about 20 tonnes and we are now asking these brakes to halt a reel of over 300 tonnes.
COURT:	It won't work.
MR MOORE:	It won't work.
COURT:	That's why it's definitely going to have burning marks on this (the brakes).
MR MOORE:	Yes.

114 The Plaintiff rightly highlights that the Defendant had not tested the mechanical brakes in its intended application *ie*, in safely stopping a *rotating* 300 tons reel. All that was effectively tested during the FATs was the brakes' capacity to act as a *parking* brake when a 15 tonnes line tension was applied at 9.2 m diameter. Thus the Defendant cannot rely on the successful FAT test of the brake as a mere *parking* brake to show that the braking system has been designed to safely stop a *rotating* 300 tons reel.

Sub-frame Bearing Housing was under-designed

115 Mr Moore examined the fabricated sub-frame structure of the RDU that housed the bearings for the pinion gears, the main gear and the drive shaft and opined that they were inappropriately designed. Accordingly to Mr Moore, the relatively thin braces and the I–section beams within the structure provided little in the way of stiffening and torsional stability. The CAD model below shows the internal construction:



116 Mr Moore noted that the RDU's bearing housings were being supported and located by the bolts. In his view, heavy-duty fabricated structures are normally used to support such a drivetrain. The bearing housing is usually designed to be an integral part of the fabrication. This negates the need to rely on the bolts to attach the bearing housing to the main fabrication. In such cases, the item would be designed and manufactured as a casting or heavy-walled fabrication and would also incorporate strengthening members as part of its design. This ensures sufficient local rigidity and support of the loading experienced by the pinion bearing and provides a means of load transfer into the sub-frame fabrication which in turn transfers its load into the main support towers. Further, the bearing is normally located directly into the heavy walled structure via a very tight clearance fit. This

provides a rigid base to seat and support the bearings whilst under load and provides an accurate machined datum, consequently giving better alignment of the bearing to the outboard mounting of the pinion. A similar arrangement is also appropriate for the bearing housing for the main driveshaft. This is critical to ensure correct alignment of the pinion gear to the spur gear. A similar arrangement appropriate for the bearing housing for the main driveshaft is also critical to ensure correct alignment of the pinion gear to the spur gear. In his report, he showed the following examples of correctly designed bearing and gearbox housings.

117 In the light of Mr Moore's criticisms of the design from a conceptual perspective, Mr Hooper conceded that the whole bearing housing could be better designed and it was "*definitely not an optimum housing arrangement*." In fact, the Defendant had shown no technical justification from departing from what one would normally expect for such a design. There are no calculations or engineering analyses on record in the Defendant's possession that would support the sufficiency or structural robustness of the Defendant's design. I therefore find that the Defendant had not given due consideration to whether the bearing housing was appropriately designed *at the time it was designed*.

118 Thus the experts, Mr Huang and Mr Natarajan, had to separately perform a FEA Analysis to verify whether or not there was *in fact* an under-design of the sub-frame bearing housing. This will be discussed below under "Finite Element Analysis".

119 It suffices for me to state at this juncture that I reject Mr Huang's FEA analysis that the sub-frame bearing housing is structurally adequate and fit for its purpose because he had not used a sufficiently fine mesh for his model and had modelled incorrectly the pinion shaft as fully rigid when it was not the case. I accept Mr Natarajan's FEA as being more accurate as he had used a fine mesh analysis and modelled the pinion shaft as non-rigid. Mr Natarajan's FEA analysis shows that the stresses within certain parts of the sub-frame exceed the yield strength of the steel used and more importantly, the sub-frame deflections (in the range of millimetres instead of microns) are simply too large to maintain gear to pinion alignment within acceptable tolerances, which are in the order of

microns.

120 Apart from the theoretical analysis, the practical tests and measurements performed on the sub-frame under actual loads also show deflections in the order of several millimetres, thus broadly validating the results of Mr Natarajan's FEA analysis: see 6 AB 5067 to 5068; deflection videos in Plaintiff's Multimedia Bundle at SANY0016; SANY0020; SANY0021.

121 Accordingly, I find that the sub-frame bearing housing, which I regard as a key component in the RDU, was under-designed and did not have sufficient rigidity. I accept Mr Drago's evidence that the drive gear system is thus made to operate under severe misalignment conditions. This is another cumulative factor to be taken into consideration in my assessment whether the RDU as a whole is of merchantable quality and fit for its purpose.

Bearing arrangement for the main shaft was under-designed

122 Mr Moore took issue with the fact that Mr Hooper had adopted the wrong bearing separation distance for his calculations of the loads on the bearings for the main shaft. Mr Hooper accepted this was an error on his part. Mr Moore then produced a set of calculations at Exhibit P18 based on the correct bearing separation distances for various roll conditions and different safety factors having regard to the shock loads expected during the operation of the RDU. There was some debate as to whether the roll conditions were already incorporated within the three acceleration values that were adopted for the design.

123 Even assuming that the effects of roll were already taken into account within the three acceleration values, it is clear to me that shock loading must be considered during the RDU operation because as pointed out by Mr Moore, there is likely to be some variance in the different types and quality of the reels that would be supplied by the reel manufacturers and suppliers. As a snug fit between the RDU's reel insert and the reel itself cannot be assumed, some amount of shock load may well be imposed on the main shaft bearings as a result of the start and stop operation of the RDU. Specifically, ISO 76 requires the following safety factors to be applied:

Table 5 — Guideline values of static safety factor S_0 for roller bearings

Type of operation	S_0 min.
Quiet-running applications: smooth-running, vibration-free, high rotational accuracy	3
Normal-running applications: smooth-running, vibration-free, normal rotational accuracy	1,5
Applications subjected to shock loads: pronounced shock loads [a]	3
For thrust spherical roller bearings, a minimum S_0 of 4 is recommended for all types of operation. For case-hardened, drawn cup needle roller bearings a minimum S_0 of 3 is recommended for all types of operation.	
[a] Where the magnitude of the load is not known, values of S_0 which are at least 3 should be used. If the magnitude of the shock loads is known exactly, smaller values of S_0 can be applied.	

124 Mr Moore's calculations at Exhibit P18 show that even if roll is excluded, the SKF bearings specified in the design drawing would not be adequate if the requisite static safety factors are to be taken into account. Mr Hooper defended the adequacy of the SKF bearings by *assuming* that the load capacity of the SKF bearings would have incorporated the above safety factors as required by ISO 76 code on bearings. Without being shown some literature from SKF or the China manufacturer of the non-SKF bearings used in the RDU that their bearings are *inherently* designed for shock loads in accordance with ISO 76, I am inclined to take the view that the cheaper China manufactured

bearings in fact used in the RDU are likely to be of a lower quality than SKF bearings, and hence, even more likely than not to be under-designed particularly when shock loads to the bearings have to be considered apart from the static loads.

125 I accept the Plaintiff's submissions that the ISO 76 code stipulates the required safety factors to be applied to the manufacturer's static bearing load in order to achieve satisfactory performance, longevity and safety and it is wrong to assume that a manufacturer would have already taken the requisite safety factors into account when producing a static load rating for its bearings because a manufacturer cannot possibly know how its bearings are going to be used and in what type of application and environment. That is the reason why ISO 76 stipulates different safety factors for different application conditions.

Main Drive Shaft not established to be under-designed

126 The Plaintiff relies on Mr Loh's admission in cross-examination that the main drive shaft was under-designed:

Mr Tan:	No provision is made for heave, correct?
A.	Yes. But diameter of the shaft is 320.
Q.	Yes, I know. But if we add in heave, we will get a different calculation, correct?
A.	Okay, yes.
Q.	I'll just show you the different calculation. Your Honour, I have handed up a sheet which is printed on two pages. The front page should be headed A019, checking shaft diameter, and that first page simply uses the numbers used in A018 and A019. Over the page, we recalculate the force, taking into account heave, and the result of that is that we see the shaft needs to be more than 320. In fact, the shaft needs to be 337.3699mm.
A.	Yeah, I got it.
Q.	Yes. So simply by adding or providing for heave, we find that this shaft that was installed in the RDU of 320mm diameter was, in fact, inadequate. You agree or disagree, Mr Loh?
A.	Yes, from this calculation it is.
Q.	Thank you.

127 Mr Hooper basically did not dispute the method of calculation but highlighted that a grade of steel 40Cr used for the manufacture of the main drive shaft was tested as having a tensile strength of 825MPa (See ABSG Inspection Certificate No. 07-5772-SQ-01 at 1AB 190) and based on this tensile strength, there was no under-design:

MR HOOPER:	In the plaintiff's documentation, they have mentioned that the shaft is undersized, and we looked through the original calculations done by the defendant and the original material specified was AISI 5140, which if you look at the standard for that particular grade of steel, it has a yield stress of a range from 785 to 980Mpa.

COURT:	Mr Hooper, is under-size in terms of its diameter and therefore in its ability to take the torque, is it?
MR HOOPER:	Torque and bending and the load.

...

… So we approached it, we looked at the actual material that as used, which is **40CR, which is an equivalent to AISI 5140 and the actual mill certificate for that material shows that the yield stress is 825Mpa**. So if we apply the same safety factor, you might say, that the original defendant's engineer used, and he only used 43 per cent of the yield stress, **we work out that the minimum diameter is approximately 275mm, well within the range that the actual shaft is 320mm diameter. So it has a considerable reserve capacity. That calculation includes torsion as well as bending.**

[emphasis added in bold]

128 Given Mr Hooper's evidence above which I accept, I do not think the Plaintiff is able to establish that the Main Drive Shaft was under-designed.

Finite Element Analysis

129 FEA is a method of structural analysis used by the experts both from the Plaintiff (*ie*, Mr Natarajan) and the Defendant (*ie*, Mr Huang) to check the structural adequacy of the RDU. Mr Natarajan describes FEA as "*a computerised analysis technique in which a complex structure is analysed for its response to applied forces by decomposing it ("meshing" it) into a large number of small elements where the corresponding equations can be solved and then combined into a result for the overall structure*."

130 It must be stressed that the FEA is based on the primary assumption that the bolt connections are adequately designed and the bolts are strong enough under the respective design loads to ensure transmission of the loads through the bolted connections. If the bolts do slip and loosen, then the FEA analysis is no longer applicable because the structure already fails at the joints. The experts for both parties agree on this.

131 Nevertheless, the integrity of the structural design of the RDU was checked using the FEA on the *assumption* that the bolting arrangements are adequate (which, as shown earlier, is factually not the case.) If the FEA analysis shows failure, then it can be said that *even if* the bolting arrangements have been adequately designed, the structure of the RDU itself remains inadequately designed.

132 A major area of contention between Mr Huang and Mr Natarajan relates to the appropriate mesh size for use in the FEA model at critical areas of the structure. Mr Natarajan took the view that the 50 – 70 mm mesh size used by Mr Huang was too large to model the components such as the sub-frame plate having a thickness of only 25 mm. Mr Natarajan on the other hand, carried out his FEA modelling on the basis of much smaller mesh sizes of 25 mm and even smaller 10 mm mesh sizes for critical areas and he had selected his mesh sizes after performing a mesh sensitivity analysis. Mr Huang's reasoned that his large mesh is rendered permissible by a DNV engineering code applicable to the design of hulls of ships.

133 I do not accept Mr Huang's justification for using a large mesh when the RDU components being modelled, *eg*, the sub-frame components, are much smaller than structures like ships' hulls. I further note the Defendant's submissions at [141] that Mr Huang had himself conceded that incidence of localised high stresses can result from the geometry of the shape of the structure and to study such areas of high stress in greater detail, it is usual to use a smaller mesh size in the FEA model. That is precisely the point because the RDU components being modelled are of complex geometry and are likely to have high stresses of a non-uniform nature within the complex geometry, which can only be properly modelled when a fine mesh size is used in the FEA as was done by Mr Natarajan. Accordingly, I prefer the results of Mr Natarajan's FEA (based on a much finer mesh size) over that of Mr Huang's FEA (which was based on a coarse mesh size greater than even the plate thickness used in the RDU sub-frame component being modelled).

134 Mr Natarajan's approach to do a sensitivity analysis first, and then select the appropriate mesh size to use for the FEA, appears to me to be the more logical engineering approach to model correctly the stresses in the structure. Secondly, Mr Huang's reliance on *inappropriate codes*, applicable to building of ships or large floating production installations but clearly inapplicable to much smaller sub-frame components within the RDU, to justify both (a) his use of very large mesh sizes; and (b) an increase of allowable stress by 35% at the hot spot stress areas, is also contrary to the Defendant's contractual obligation to provide full certification according to the *appropriate standards and guidelines*. If Mr Huang were to rely on the ABS code on lifting appliances, which is the more appropriate code, Mr Huang will find that his position is not supported.

135 On the allowable stresses for use as the acceptance criteria in the FEA, Mr Natarajan disagreed with the use of 0.9 Fy (*ie*, Material Yield Strength) by Mr Huang. Mr Natarajan maintained the view that the lower figure of 0.64 Fy was more appropriate based on the American Institute of Steel Construction Specification for Structural Steel Buildings, Allowable Stress Design and Plastic Design dated 1st June 1989 (the "AISC Code") and the ABS Code on Lifting Appliances. Mr Huang tried to justify the use of 0.9 Fy by assuming that the dynamic loading arising from the ship's motion is similar to the effect of wind and seismic load applicable to onshore structures, where the allowable stress of 0.66 Fy can then be increased by a third to 0.88 Fy based on the "AISC – Allowable Stress Design Code".

136 I reject Mr Huang's justification for the use of a higher allowable stress of 0.9 Fy as the two situations are hardly analogous. The regular motions of a ship rolling and heaving out at open sea cannot be compared with the rather rare earthquakes for which the AISC – ASD code may allow a one third increase in the allowable stress when the additional seismic forces are factored into the design of static structures on land. It is rather absurd to have a ship mounted structure subject to frequent cyclical forces and fatigue loading being designed close to the Material Yield Strength (*ie*, at 0.9 Fy as advocated by Mr Huang) and in a manner similar to a static structure on land which may be subject only to the very rare seismic forces.

137 During the second tranche of the hot-tubbing which focussed on the FEA, Mr Huang asserted that the reel fitted snugly between the two towers and as a result of the snug fit, the reel itself functioned as a strong horizontal cross-beam preventing the two towers from bending or deflecting inwards as a result of the heavy 300 tons load from the reel. He took issue with the fact that Mr Natarajan had not taken that operating condition into account in his FEA model.

138 However, I note that Mr Huang himself had not modelled his FEA on that basis either when he prepared his expert opinion for his AEIC. Neither was the FEA by Dr Liu Li submitted by the Defendant to ABSG for verification and certification modelled on the basis that the reel was to be treated as a strong cross-beam for the FEA analysis. Further, nothing is shown to me from the design

documentation of the Defendant which suggests that it was a design consideration that the reel itself was to act as a horizontal cross-beam between the towers and that as an operating condition, it is imperative for the personnel operating the RDU to ensure that after inserting the reel between the two towers, the two towers are moved inwards each time to secure a tight fit with no clearances between the reel and the tower such that the reel can act effectively as a horizontal cross-beam for the two towers for the structural integrity of the RDU. I have my doubts whether achieving a snug fit *at all times operationally on the deck* is a viable design and operating condition, and having to depend on that for the structural integrity of such important major components of the RDU, *ie*, the two supporting towers, may in itself not be a safe method of design because the risk is too high, being much too reliant or dependant on the actions of the RDU operators.

139 Be that as it may, I requested the two FEA experts of both parties to redo their FEA to take account of the point raised belatedly by Mr Huang of the reel essentially acting as a cross-beam between the two towers of the RDU to see if in fact the results of the FEA do bear out Mr Huang's contention that the maximum stresses would be reduced safely below the yield strength of the steel used for the construction of the two towers.

140 As the trial was already coming to close, I gave them leave to address this issue by way of further written expert opinions, without a recall of the experts to testify. Mr Natarajan very helpfully carried out a further FEA using fine mesh sizes at critical areas to test the adequacy of the RDU structure with the reel acting as a strong cross-beam, able to act only under compression but having no capacity to absorb any tension. His results show stresses beyond yield at the sub-frame:

> To explore the question of what would happen when a reel is made snug to the towers BEFORE being lifted, so that the inward bowing of the towers due to the lifting process would be constrained by compression of the reel, we conducted the following analysis to arrive at an answer by the following two steps:
>
> 1) Determine the inward deflection of the towers under the effect of reel weight + heave assuming the reel is not snug and thus the towers can deflect inwards. The resulting inward deflection at the reel elevation was 31.8 mm.
> 2) Model the structure as in step 1 above but assume a reaction load of 33 te acting outwards from the reel to the towers (as a result of the reel compression) and thus resisting their inward deflection under the weight load. The resulting deflection at the reel elevation was 0.085 mm.
> 3) It can be concluded from step 1 and 2 above, and in particular from the fact that the deflection dropped from 32mm to 0.085mm, that each tower experiences reaction load of 33 te from the reel in the transverse outward direction when the reel is made snug to the towers before being lifted.
>
> Now that the value for the additional outward reaction load resulted from the reel being made snug is known from the above analysis, we repeated the analysis for the stresses on the tower under the effects of reel weight plus heave plus transverse acceleration (a transverse loading of 66 te (being 300 te x 0.22 m/sec² of transverse acceleration) PLUS the 33 te of compression loading determined in the step 3 above. The resulting stresses on the sub frame plates and transverse struts are shown in the Figure 2 and Figure 3 below.

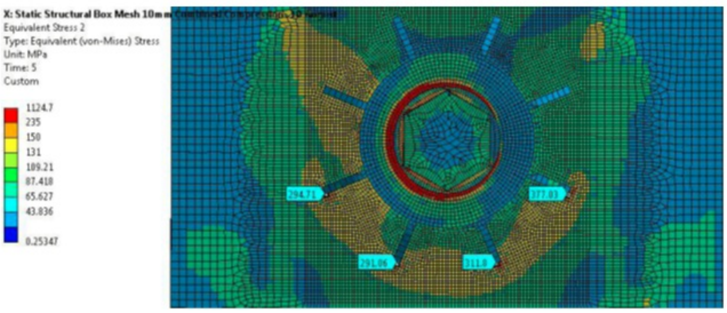


Figure 2 – Von Mises Stress with the Compression Load from the Reel Only

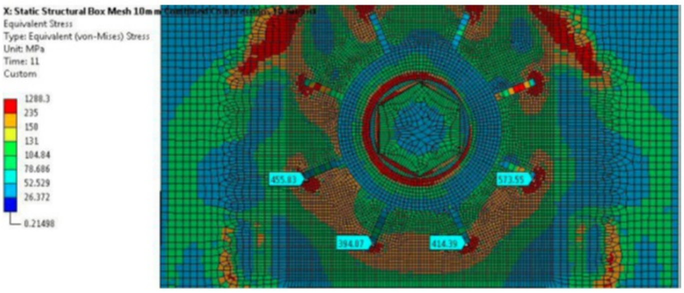


Figure 3 – Von Mises Stress with the Compression and Transverse Load from the Reel

Figure 2 and Figure 3 above describe the stresses when the towers are snug to the reel and when one of the tower is subject to the compression plus transverse loads. Due to the compression load from the reel on the tower, the effect of the transverse load is intensified when both compression and transverse loads acting in the same direction. During this phase, the von Mises stress becomes worse as tabulated in the Table 1 below.

	Probe 1 (Mpa)	Probe 2 (Mpa)	Probe 3 (Mpa)	Probe 4 (Mpa)
Transverse	377	294	312	291
Transverse plus Compression	574	456	414	394

Table 1 – Stress Comparison between Transverse Force Only and Transverse Plus Compression Forces

141 In short, treating the reel as a compression beam between the two towers leads instead to an increase in the stresses, which go well beyond that allowable. The attempt by Mr Huang to prove that the maximum stresses are reduced if the reel is treated as a horizontal cross-beam thus fails. The FEA shows that the opposite takes place.

142 I waited for Mr Huang's fresh FEA after I received the new FEA from Mr Natarajan. It was 7 weeks after the conclusion of the proceedings, long after Mr Natarajan had produced his fresh FEA report and well after the Plaintiff's closing submissions that Mr Huang decided to produce his fresh FEA report, presumably still using the coarse mesh modelling that I have rejected as his report was silent the mesh sizes he used. In his new analysis, Mr Huang assumed the coefficient of friction to be zero and he asserted that the results of his fresh FEA using the Nastran FEA programme show that the structural strength of the RDU is adequate when treating the 300 tons reel itself as a cross-beam preventing the deflection of the towers inwards due to the reel's weight. I have reason to suspect that the fine mesh modelling using the Nastran FEA programme does not show that the RDU structural strength has become adequate under the new conditions with the reel acting as a compressive beam between the two towers because Mr Huang would surely have highlighted that he had used a fine mesh analysis on this occasion, much in the same way that he had taken the trouble to state expressly that he had assumed a zero coefficient of friction. I infer that it is likely that Mr Huang's fresh FEA modelling remained a coarse mesh analysis, and his fine mesh analysis, if done and produced, would have shown stresses within the critical components in the RDU structure exceeding the yield stress of the steel (let alone the allowable stress), thereby corroborating the FEA done by Mr Natarajan using a different computer programme, the ANSIS FEA, which Mr Natarajan claims is newer and more up-to-date than Mr Huang's Nastran FEA programme.

143 The Plaintiff objects to the admissibility of Mr Huang's new FEA report, which was attached to the Defendant's closing submissions [note: 2]. As the Plaintiff would have the chance to respond in its reply submissions to the Defendant's closing submissions, I allow Mr Huang's belated report and Mr Natarajan's response to it, attached to the Plaintiff's reply submissions at Annex B to be included as part of the overall evidence so that there can be a better technical assessment of the fresh FEA modelling to take into account the point raised belatedly during the trial by Mr Huang that the reel itself could also act as a compressive beam between the two towers and should be modelled as such

in the FEA.

144 After a careful consideration of the explanation by Mr Natarajan in "Annex B – CRITICISMS OF HUANG'S NEW REPORT", I do agree with Mr Natarajan's criticisms on why Mr Huang's analysis of the case of the reel fitting snugly between the towers is flawed based on first principles. I agree with the Plaintiff's reply submission that Mr Huang's fresh analysis is an impossibility from an engineering perspective as it fails to produce a force balance consistent with the input acceleration conditions and it also fails to produce the correct contact between the towers and the reel on the left tower when the inertial forces generated during the roll acts on the right tower.

145 On the whole, I prefer the expert opinion of Mr Natarajan over that of Mr Huang. Mr Natarajan's overall analysis is also far more detailed, analytical and sound from an engineering perspective having regard to the Classification Society Rules and Codes. I reject the results of Mr Huang's FEA and the conclusions that Mr Huang had tried to derive from those results because he used an inappropriate mesh size to model the RDU structure and its components. The mesh size he used was simply too large to capture the geometry of the components and the critical areas within the RDU structure for stress analysis.

146 Mr Huang was given the opportunity to re-run his FEA based on a continuous mesh and with fine mesh sizes identical to that used by Mr Natarajan so that the court could compare the stress results thrown up by the different FEA computer programmes used by the two experts. Basically, I wanted to examine whether the two sets of results are broadly similar when the same fine mesh sizes are used which then affords me a greater level of confidence in the accuracy of those FEA results. If the results diverge significantly, then I have to examine in detail what has caused the results to be so different and what could have gone wrong. It is disappointing that Mr Huang refused to assist the court by re-running his FEA computer programme using fine sized meshes identical with Mr Natarajan. Instead, Mr Huang and the Defendant in its submissions chose to raise doubts on the credibility and reliability of Mr Natarajan's FEA analysis by asserting the following:

(a) Mr Natarajan's comments on the localised areas of stress concentration based on this FEA analysis were not borne out by the results of the Plaintiff's own physical load test carried out on the RDU which was witnessed by the ABS Surveyor, Mr Kim Dong Jin. Those results show no failure of the structure. Neither was there evidence of permanent deformation of any members which would establish generalised over-stressing.

(b) Mr Natarajan failed to produce a nodal reaction force summary for his FEA. Such a summary is critical in determining that the model is correctly constructed and there is a global force balance on each component.

(c) Mr Natarajan's initial FEA model had used a discontinuous mesh connection for the sub-frame stress analysis when a continuous mesh should have been used. When a continuous mesh was used in the rerun of Mr Natarajan's FEA model, the maximum stress on the sub-frame increased from 864 MPa to 1214 MPa, when logically the maximum stress should decrease. The Defendant contends that the FEA programme and model used by Mr Natarajan is unreliable and not credible.

147 In relation to criticism (a) above, I note that the physical load test carried out on the RDU was a static one which did not fully incorporate the design parameters that specify the equivalent loading from the large inertial forces due to the maximum transverse, heave and longitudinal accelerations acting *together* at the same time out at sea on the RDU structure together with the rated maximum umbilical tension load. Obviously it is wrong to conclude from the absence of permanent deformation

arising from a mere physical static load test under a more benign condition that the theoretical analysis by Mr Natarajan using the FEA computerised programme must necessarily be wrong if it demonstrates stresses exceeding yield at certain critical locations indicating permanent deformation at those locations when the RDU structure is subjected to both the tower and reel weight, the tension in the umbilical and the full impact of the inertial forces arising from the three accelerations, which together represent the theoretical maximum loading conditions for the design that are far more aggravating than what was simulated by the physical static test.

148 The Plaintiff also points out in its Reply Submissions that there is evidence before the court that does in fact suggest that permanent deformation exists as could be seen in the photographs taken during the bearing inspection on December 2014. Straight lines superimposed on the photographs appear to indicate that the sub-frame plates were visibly bowed. I have no reason to disregard this photographic evidence produced by the Plaintiff which effectively rebuts criticism (a) above by the Defendant.

149 In relation to criticism (b), I accept Mr Natarajan's explanation that the ANSYS FEA programme used by him performs the nodal reaction force check automatically.

150 In relation to criticism (c), I had specifically directed the experts in their fresh FEA to focus on modelling with continuous meshing and fine mesh sizes in order to obtain more accurate results. I also wanted to avoid having to consider the contentious issue in relation to the use of the non-continuous mesh instead of the continuous mesh for the FEA. As such, there is no need for me to address the comments of the Defendant in relation to the use of the non-continuous mesh initially by Mr Natarajan, the results of which I have generally disregarded. I am relying more on the fresh FEA by Mr Natarajan using continuous fine meshes. It is apt at this point to reiterate that Mr Huang was invited but refused to participate in a discussion with Mr Natarajan, specifically so that Mr Huang could verify that the subsequent re-modelling using fine sized continuous meshes had been properly carried out by Mr Natarajan if Mr Huang were to believe that it was not. In fact for the FEA re-modelling, I asked both experts to sit together to re-examine each other's FEA analysis using fine mesh sizes and continuous meshing. Unlike Mr Natarajan, Mr Huang was not prepared to do so. He did not even produce a re-modelling of his FEA using fine mesh sizes and continuous meshing whereas Mr Natarajan did. In my mind, a jointly done FEA using two different types of FEA programmes is the best way forward in the light of such complex FEA computer programmes, where input criteria and programme settings are highly complex. For all the reasons I have stated, I do not view favourably the criticisms (a), (b) and (c) levelled by Mr Huang and the Defendant in its submission on the fresh FEA done by Mr Natarajan.

151 I accept the results of Mr Natarajan's FEA model based on a continuous fine meshed FEA which show that -- at the heave, longitudinal and transverse accelerations (without accounting for the effect of the gravitational weight force in the transverse X direction resulting from a roll) stipulated by the Defendant as the input parameters for the structural design of the 300 tons RDU -- the sub-frame and the struts supporting the towers already experience areas of high stress going beyond what is prescribed in either the AISC Code (which provides for an allowable stress of 0.66 yield) or the ABS Code on Lifting Appliances at the design loads (which provides for an allowable stress of 0.64 yield).

152 Therefore even assuming that the bolting arrangements for the struts are adequately designed, I find that the sub-frame and towers as designed are unsuitable for their design loads in that the stresses calculated using the FEA exceed the allowable stress. The deformations predicted by the FEA are also in excess of what is permissible for the gears. Gears require a high level of alignment with very low tolerances in the order of hundredths of a millimeter. I find that the RDU structure does not

have sufficient rigidity to provide the necessary support for the gears and keep them in good alignment when the RDU is loaded and operated with a 300 tons reel. I consider this to be another significant factor to be taken into account when considering if the RDU is of merchantable quality and fit for its purpose.

Poor manufacturing quality

153 The Plaintiff submits that the components (*ie*, the gears, bearings and bolts) of the RDU were not manufactured to a satisfactory quality. There was lack of a proper system for (a) inspection of components; (b) assembly/installation of components; and (c) maintenance of records documenting the manufacturing process/procedures. As a result of the Defendant's poor manufacturing practices, the Plaintiff submits that the RDU is not fit for purpose and/or not of merchantable or satisfactory quality.

Pinion gears were not properly manufactured

154 Based on the totality of the evidence before me, I find that the gears were not properly manufactured as:

> (a) concentricity was probably lacking between the pinion gear and main gear due to the method of manufacture;
>
> (b) pre-existing cracks were present in the pinion gears; and
>
> (c) the actual rim thickness to gear tooth ratio was not in compliance with the drawings.

Lack of Concentricity

155 It is not disputed that the pinion gear teeth were cut before the flange was welded on. According to the Defendant, this method of manufacture ensured concentricity between the flange and the pinion gear. However, Mr Drago disagreed.

156 I accept the expert opinion of Mr Drago that if the pinion teeth were to be cut before the flange is welded on, a significant detrimental run-out would likely be present. Mr Drago explained why that would be so at p 63 of his AEIC:

> The clearance between the end of the pinion teeth and the bolt flange is small thus the pinion toothed section was either hobbed before the shaft section was welded to the flange or the teeth were shaper cut after the pinion shaft section was welded to the bolt flange. **If the pinion teeth were hobbed before the flange was welded on it is very likely that that a significant, detrimental runout condition exists.** In order to investigate this further, we had the runout of the two journals next to the gear teeth measured2. The pinion was chucked on the bearing journal on the small end (Arrow 5, Figure 1A) and using the center in the shaft on the flanged end, the bearing journal runs zero. The small journal between the pinion and the flange (Arrow 4, Figure 1A) runs within 0.002 inch. It is important to note that this surface is coated. With the surfaces running as true as possible, the registering spigot on the flange TIR reading is 0.020 inch. The machined end face (Arrow 2, Figure 4) was found to have a TIR of 0.015 inch. These large runout values indicate that it is very likely that the pinion was welded to the flange after the teeth were cut. **The assembly of the finished pinion to the flange in this manner results in poor control over the concentricity of the pinion centerline with the centerline of the Sauer Danfoss drive unit output shaft. This large eccentricity error, which is in excess of**

three times the allowable pitch diameter runout tolerance for the pinion (per AGMA 2000-B88), further exacerbates the alignment error between the final drive pinion and gear. This is a very serious design error.

[Emphasis added in bold]

157 The mere assertion by Mr Philip Chua, the Senior Technical Manager of the Defendant, that there was concentricity does not carry much weight as he had not checked the concentricity himself unlike Mr Drago. All four pinion gear and flange assemblies with bolting holes were manufactured, machined and welded in China. After the items arrived in Singapore, Mr Chua said that the Defendant did not check their concentricity and "*then we just fit [them] up*". In short, the Defendant simply assumed that the pinion gear and flange assemblies were properly machined and good concentricity had been achieved. After receiving the items from China, the Defendant simply installed the items in the RDU without performing the necessary quality control and concentricity checks. In my view, it was also convenient for the Defendant to take such short cuts during the RDU fabrication. Accordingly, the Defendant failed to verify if these components delivered to the Defendant were properly manufactured in accordance with the dimensions and had kept within the concentricity tolerances (if any) stated in the design drawings for the pinion and flange assembly. If no concentricity tolerances were in fact stated in the design drawings, then it would suggest that the Defendant themselves were unaware of the importance of concentricity for the pinion gear and flange assembly in the design. How is the Defendant to expect the manufacturers in China to ensure that the concentricity is kept within certain tight tolerances to minimise alignment error if the design drawings themselves are silent on the allowable tolerances in the first place?

158 The Defendant failed to produce any contrary evidence of its own concentricity measurements to challenge the correctness of Mr Drago's run-out figures measured with calibrated instruments that show a lack of concentricity with reference to applicable gear standards. Accordingly, I accept Dr Drago's evidence that one of the four pinion gear and flange assemblies that he examined had a large eccentricity error in excess of three times the allowable pitch diameter run-out tolerance for the pinion (per AGMA 2000-B88). On a balance of probabilities, the inference is that the rest of the other three assemblies also suffered from the same range of eccentricity errors in the course of manufacture.

Pre-existing Cracks

159 Mr Drago performed a metallurgical evaluation of the failed pinion gear recovered from the incident. He found pre-existing quench cracks present in the tooth area. The quench cracks were observed to have extended towards the centre of the tooth. Further, non-destructive testing of the replacement pinion gear on Tower "A" (which was new and unused and had never been in service) also showed similar quench cracks. Based on the report entitled "K2- Reel Drive System Non Destructive Examination Report 8th July 2013" prepared by Mr Paul Goh from K2 Specialist Services Pte Ltd, the pre-existing quench cracks could only have been a result of poor manufacturing practices *ie*, the faulty heat treatment of the pinion gears.

160 After manufacture in China, the gears should have undergone appropriate quality checks and inspection before installation in the RDU in Singapore. Appropriate non-destructive testing should have been carried out to ensure that there were no defects in the first instance for such critical components of the RDU prior to their installation. However, there is no evidence of any such testing before the court. Based on the test results from K2 Specialist Services and the opinion of Mr Drago, I conclude that the gears were defectively manufactured and were not of satisfactory quality.

Lack of Proper Heat Treatment

Lack of Proper Heat Treatment

161 Mr Drago carried out a detailed metallurgical evaluation on the failed pinion gear and discovered the following deficiencies:

(a) An induction or flame hardened layer of non-uniform thickness was present on the gear teeth.

(b) Low hardness values were found in the core areas of the teeth as well as on the surface of the tooth roots. All of the values were lower than Rockwell C 20, indicating that the core areas were not heat treated in the first instance.

(c) The lack of heat treatment of the core areas was confirmed by the presence of pearlite and ferrite in the microstructure, thus indicating a non-heat treated structure typical of a soft core. This resulted in a low strength pinion gear, which would be associated with a much lower endurance fatigue life than what could be obtained through a standard heat treatment of the pinion component.

(d) Intergranular cracking was found in the heat affected zone of the weld between the pinion and the mounting plate securing it to the Sauer Danfoss gearbox unit. This was most likely a result of quench cracking (as seen in the pinion gear teeth) extending along the weld length on the pinion side.

162 The presence of intergranular cracking would, in the opinion of Mr Drago, severely weaken the joint between the pinion and the mounting plate, and further contribute to misalignment of the pinion and its premature failure in service. As these quench cracks and the other deficiencies mentioned at [161] were the probable result of poor selection and control of the heat treatment processes at the time of manufacture, they were therefore already present in the pinion gear prior to it being placed into service.

163 I have no reason to disagree with the above conclusions reached by Mr Drago from his metallurgical investigations. The poor heat treatment or the lack thereof on the core areas of the pinion gears during their manufacture in China meant that the pinion gears installed in the RDU were sub-standard.

Non-compliance with the Gear Rim Thickness Ratio specified

164 The gear rim thickness to gear tooth whole depth ratio shown in the gear drawing was 1.22. However, when Mr Drago measured the dimensions at each tooth location, he found the actual ratio to vary from 1.185 to 1.025. Thus, the gears were not manufactured in accordance with the Defendant's gear design.

Bearings used were not SKF bearings

165 The engineering drawings clearly specified that SKF bearings were to be used in the RDU. The specified SKF bearings perform a critical function as they support the main shaft bearing the load from the 300 tons reel. It is not disputed that cheaper and lower grade China manufactured bearings were used instead. The decision not to use SKF bearings was not due to mere inadvertence. It was sanctioned by Mr Harry Chua, the managing director of the Defendant.

166 Although the operations department knew that the bearings purchased were not SKF bearings, they nonetheless installed these non-SKF bearings in the RDU as long as they were dimensionally

similar to the SKF bearings. The actual load bearing capacity of the China made bearings was not even ascertained from the Chinese manufacturer at the time of the RDU design. The Defendant merely assumed that the load bearing capacity of the non-SKF bearings would be comparable to the SKF bearings because they were similar in dimensions. I consider this to be a risky thing to do from the point of view of a prudent engineer designing the RDU.

Inadequate bolt thread engagement lengths

167 Mr Philip Chua said at [12] of his AEIC that the assembly engineers and supervisors of the Defendant "*have been trained to ensure that all installed bolts must have sufficient length i.e. some threads protruding out of the nut, and for those with blind holes, the engagement length should be minimum 1.5 times of the bolt diameter*."

168 However, none of the threaded lengths of the bolt holes on the tower strut flanges met the requirement of at least 1.5 times the bolt diameter. M30 bolts were specified for the tower strut flanges. Hence, the minimum engagement length for these M30 bolts must be at least 45mm. The Nobletech Report dated 2 October 2012 ("Nobletech Report") stated that the thread length engagement varied from as low as 20.8 mm to 29.95 mm for these bolt holes on the tower strut flanges.

169 In any event, the minimum bolt thread engagement length of 45 mm could never be met as the plate thickness of the tower strut flanges was actually 32mm.

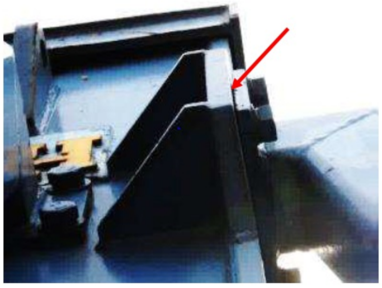
170 The bolted joints on the bearing housing were also too short. M16 bolts were specified thus requiring a thread engagement length of at least 24 mm. However, the plate thickness was specified to be 16 mm again making this required thread engagement length unachievable. The Nobletech Report revealed that the thread engagement depth was between 6 to 8 mm only.

171 I conclude therefore that the bolted joints on the tower strut flanges and bearing housing were both inadequately designed and manufactured in terms of the requisite engagement thread lengths for the bolts.

Flange gaps found at the strut connections to the tower frame

172 The Nobletech Report established the existence of gaps in the flange plates of the struts that were attached to different parts of the tower frame. Some of these gaps were significant (See 6 AB 5088). Mr Moore opined that these gaps were likely to be the result of poor joint design and the bolts loosening during the last operation and he added that:

> It is essential that any joints such as this do not have a gap between the components. Working faces must be in contact and have the correct pre-load applied; otherwise friction between the two components will be non-existent. This will lead to gaps, joint slip and transfer of loading to the bolts which will ultimately fail by shear or fatigue.



173 Given the results of Dr Eccles's analysis predicting loosening of the bolts on these joints due to cyclical loading, I am inclined to accept the Plaintiff's experts' conclusion that these gaps probably formed as a result of the bolts loosening during the RDU's operation.

Defective bolt clearance holes in the sub-frame end plate

174 From the photographic evidence, it is clear to me that the bolt clearance holes in the sub-frame end plate were badly drilled evidencing poor manufacturing practices. It would appear that there were machining errors with misalignment in the original hole pattern and re-drilling had to be done at the component fit up stage so that the bolts could be screwed into the threaded holes. Mr Philip Chua confirmed that the pre-drilling was done in China. During the assembly of the parts in Singapore, re-drilling was done if the bolts could not match the clearance holes.

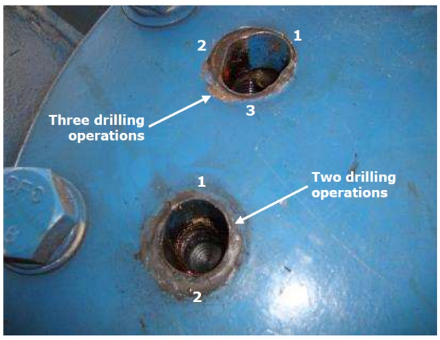


Figure 53: Elongation of holes due to initial incorrect positioning

175 The Plaintiff submits that the re-drilling modifications would exacerbate movement of the component, especially when combined with insufficient bolt pre-loading and would not be acceptable especially for a component that was critical to ensure the secure attachment of critical gear drive components. I agree.

176 As the RDU tower was constructed using high strength bolts, I accept the Plaintiff's submission

that normal practice would dictate that the defective components be rejected and replaced with correctly machined items. The AISC Code provides that for high-strength bolted construction or assembly, "*poor matching of holes shall be cause for rejection*."

Lower grade bolts used

177 The engineering drawings specified grade 10.9 bolts for the bolted connections in the main base and the sliding base. However as confirmed by Mr Philip Chua, a lower grade of bolts *ie*, grade 8.8 bolts was eventually used during the fabrication of the RDU. No explanation was forthcoming from the Defendant on how that could ever have happened.

178 In my view, usage of bolts of a grade lower than what was specified in the engineering design will compromise the strength of the bolted connections designed for those locations in the RDU.

No checks for compliance with drawings and inadequate inspection records

179 Given that the Defendant's manufacturing processes were supposed to be compliant with ISO 9001 and accordingly, there should be proper control of its technical/manufacturing records, it is surprising (and rather unbelievable) for Mr Philip Chua to claim that the Defendant had misplaced its records for the bolts installed in the RDU (*ie*, if the records existed in the first place). The AISC Code stipulates that the inspection of slip-critical, high-strength bolted connections shall be in accordance with the provisions of the RCSC Allowable Stress Design Specification for Structural Joints Using ASTM A325 or A490 Bolts. The Defendant failed to adduce any documentary evidence to demonstrate that it had in fact carried out such inspections of the critical bolts at the struts and tower bolted joints during the manufacturing process (as required by the AISC Code) to ensure compliance with design drawings.

180 I infer that it was more likely than not, that the Defendant did not perform any inspection or compliance checks to ensure that the grade of bolts actually installed in the RDU at the critical connections was in accordance with the engineering drawings. If the Defendant had done so, the inspection records for the high strength bolted connections would have been readily available and the wrong grade of bolts of 8.8 (see [177] above) would never have been used on the RDU that was delivered to the Plaintiff.

181 Although Mr Philip Chua also asserted that systematic checks were carried out during the production process such as dimension and fitting checks, however he did not furnish any contemporaneous inspection documents to evidence that these checks were in fact done. Mr Philip Chua later admitted that the Defendant had a duty to check but failed to check, prior to installation (as would have been required by ISO 9001), that all the machined and manufactured parts (such as the gears) from China complied with the dimensions stipulated in the drawings:

MR PRADHAN:	Mr Chua, what I understand is this -- it came from China, and then it was simply installed by PH.
A.	Yes.
Q.	Yes?
A.	Yes.

....

MR PRADHAN:	Mr Chua, then how do you know that there were checks done to measure the runout?
COURT:	Did he say there were checks done to measure runout? Did he say that?
MR PRADHAN:	Well, no, he hasn't, your Honour.
COURT:	You must ask him first. These are foundational questions.
MR PRADHAN:	Let me put it in these terms.
A.	**I presume you want to ask me whether we checked the runout. We did not check. This part already all machined in China, and then we just fit it up.**
Q.	So PH [the Defendant] cannot tell whether it was properly machined. Right? It cannot tell that the runout was properly --
COURT:	**Basically, PH didn't check all the machining parts and the dimensions. According to ISO standard, they must check to ensure that it's in accordance with the drawing dimensionally. Right, or not? You assume that it's correctly done?**
A.	**Yes.**
Q.	**There's no QC done on the dimensional check, the tolerance checks and all that. Nothing was done; right? You assume China will get it right for you. Correct or not?**
A.	**Right.**
......	
COURT:	What does ISO say? I don't know about ISO, I'm asking you. For this sort of thing, **what does ISO say?**
A.	**ISO says, I mean, you have to have duty to check on the product according to the drawing.**
COURT:	ISO says you take that plate, look at that drawing and check whether the dimensions are correct. Check whether the thickness is correct.
A.	Yes.
….	
COURT:	So ISO says dimensional checks on machine parts?
A.	Yes.

[Emphasis added in bold]

182 In relation to the assembly of the sub-frame, I saw no records of any checks made to ensure concentricity and alignment. From the totality of the evidence, it does not appear to me that any

such checks were in fact done by the Defendant. As the Defendant's solicitors' letter dated 30 July 2013 had stated that:

> Paragraph 1 – Pinion
>
> a. The concentricity was measured by mounting the shaft on a lathe and rotating it 360o with a dial gauge to measure the concentricity from datum A. The whole assembly was machined as one piece, after which the concentricity was verified. A final verification took place when the pinion assembly was matched with the other assemblies. There are no records of the actual measured results.
>
> b. The concentricity tolerance was determined to meet the gearbox manufacturer's requirements and follows the Defendants' standard for such mating flanges with components.

Mr Philip Chua was questioned at length on the basis of those facts stated therein. Eventually, he admitted that he was not sure of the source of the factual information.

183 Mr Philip Chua stated at [17] to [19] of his AEIC that:

> 17. We fabricated the subframe which houses the main shaft and pinion shaft. We installed the main shaft and gear into the subframe. We aligned the pinion with the gear, checking the backlash, contact and parallelism*. The position of the mounting flange of the pinion shaft bearing housing is fixed after the pinion was aligned, and then welded to the subframe. *(Note: backlash, contact and parallelism are the 3 checks for gear alignment).
>
> 18. We then aligned the pinion shaft and gearbox output flange concentrically.
>
> 19. My sketch illustrating this is attached marked "**PC-3**".

184 It turns out that Mr Philip Chua did not have any personal knowledge of these alleged processes of alignment of the pinion with the gear and checking of the backlash, contact and parallelism. I therefore give little weight and utility to those unsupported assertions of Mr Philip Chua in his AEIC. Not only does the operations department under his charge not appear to have any internal standards or procedures which were applicable to the installation of the gears as can be seen from his evidence below, they also do not have the capability to check the gears:

> Q. But in terms of when you come to the installation of whatever comes from China, you don't bother checking against any standard or anything like that? You don't?
>
> A. No, we don't check the standard against our engineering department.
>
> Q. I suppose that also means that your operations department didn't have any internal standards which it applied to the installation of gears? Internal engineering standards, like a document saying how the gears should be installed -- how something should be installed, what checks should be done at the end of the installation process; there's nothing like that. Your operations department wouldn't have anything like that?
>
> A. As I said, the gears are not our speciality. So we are unable to check the gears.

Poorly trained workforce

185 Mr Philip Chua claimed at [12] of his AEIC that the "*assembly engineers and supervisors have been trained to ensure that all installed bolts must have sufficient length i.e. some threads protruding out of the nut, and for those with blind holes, the engagement length should be minimum 1.5 times of the bolt diameter*".

186 When it was established that in reality none of the bolts at the tower strut flanges or the pinion bearing housing fulfilled that requirement, Mr Philip Chua had to concede that the reason for non-compliance with the drawings was the poor training of the Defendant's employees, thereby contradicting the assertions in his own AEIC that they were properly trained.

Non-Compliance with ISO 9001

187 The Defendant holds itself out as having a quality management system for its design and manufacturing processes that is ISO 9001 compliant. It says it is ISO 9001 certified.

188 Chapter 7.3 of the ISO 9001:2008(E) ("Chapter 7.3 ISO 9001") document spells out the stringent requirements for planning, controlling, review, verification and validation including the maintenance and control of proper records throughout all the stages of the design and development of a product.

189 Understandably the Plaintiff expects the Defendant to have gone through a detailed and rigorous exercise in accordance with Chapter 7.3 ISO 9001 when it designed and manufactured the RDU since the Defendant represents to all its customers and the world at large through its website and advertising literature that it is certified to be compliant with ISO 9001.

190 I agree with Mr Moore that it is standard engineering practice to maintain a technical manual which sets out all the design calculations for the RDU so that they can be inspected at a later date. These design calculations are an important part of the design and development records that the Defendant should have properly kept and maintained.

191 It is clear from the review by the expert witnesses of the documentary records produced by the Defendant at the trial that not all the design calculations for all the bolts, gears, bearings and brakes and the safety factors used were available and produced to the court. For instance, bolted connections were used extensively in the RDU but the Confidential Bundle only shows "Main skid bottom bolts calculation". What about the calculations for the bolts at other bolted connections present in many other parts of the RDU *eg*, the bolted joints on the transverse struts? None appears in the Confidential Bundle. With no explanation forthcoming from the Defendant, the inference I make is that complete design calculations for all the bolts, gears, bearings and brakes ("complete design calculations") were never done by the Defendant or if they had been done, then they were deliberately not disclosed. As the Defendant is supposed to be ISO 9001 compliant and from the fact they could still tender other fresh documentary evidence in relation to this RDU project eight years later during the trial, I am discounting the remaining possibility that the complete design calculations have been lost and can no longer be found. It is not likely in my view.

192 I further note that the person primarily responsible for the design of the RDU is purportedly one Hu Xuya as stated on the drawings. Yet the Defendant decided not to call him as a witness. It later transpired that Hu Xuya did not design the Plaintiff's RDU but had apparently designed the Punj Lloyd 300 tons RDU. When the Defendant secured the order for the Plaintiff's RDU, the Defendant assigned Ms Tan Sin Liu, a junior design engineer, to be in charge of this project for the Plaintiff as it was "*quite similar; almost the same*" as the Punj Lloyd 300 tons RDU according to Mr Lei. Mr Lei testified that "*she [Ms Tan Sin Liu] just modified the job number, changed the job number, the individual*

number, the name and also the file name from the existing joints [sic]". However, Ms Tan Sin Liu clarified that her design involvement was as follows:

> Q. Can you explain to the court, what was your involvement in relation to the design aspect of this RDU?
>
> A. This job was assigned to Hu Xuya and me and then because Hu Xuya was busy with other job, and this job we have a similar one, we did similar RDU before, so mainly was done by me under guidance of other senior engineer and manager, Lei Chengyi. By the time, Xu Bin is no longer around.
>
> Q. Mr Lei Chengyi was the design manager?
>
> A. Yes.

193 I accept the Plaintiff's submission that no evidence was adduced by the Defendant on the existence of any initial design calculations necessary to handle the design modifications to convert the design of the larger, heavier and presumably more solidly built Punj Lloyd RDU to that of the Plaintiff's RDU, which was built with a lighter tonnage of structural steel and yet was specified to have a higher tension capacity and a higher rotational speed for the reel. At that time, the Defendant did not even have its own in-house structural engineer to perform structural calculations to size the structural elements for a new RDU or modify an existing RDU design to meet different technical specifications. It therefore engaged an external consultant to perform a FEA modelling *subsequently* to verify the structural integrity of its own initial modified design, which was apparently produced without any initial structural engineer's design input for the sizing of the structural elements.

194 I pause here to state I am not prepared to draw an inference that this must necessarily be an indication of a fraudulent practice because design is an iterative process and a subsequent re-design could take place after receipt of comments from the external structural engineer even if there is no initial design input from a structural engineer. If so, then the choice of drawing out the design first and then seeking the engineer's validation later, as opposed to getting the structural engineer's input first before embarking on a detailed design may well be a matter of preference and working or design efficiency, and not so much an indication of a fraudulent design practice.

195 I now return to the main discussion. According to Mr Hooper, the Defendant's own expert witness, the only design calculations he had seen from the Defendant were those contained in the Confidential Bundle that were submitted to ABSG. However, that Confidential Bundle was more concerned with the structural calculations for the wire spooler towers of the RDU. Except for the wire spooler towers, I note that the Confidential Bundle made no reference to any design codes applied by the Defendant when it designed the specific parts or components for use with the RDU. This may suggest that the Defendant did not even apply proper codes in the course of designing the RDU although it is clearly the Defendant's contractual responsibility to apply the appropriate codes in the design of the various parts of the RDU, its equipment and its specific parts.

196 Further, the factors of safety have not been clearly stated for the design of certain critical components of the RDU. Dr Eccles rightly pointed out that in high volume production (*eg*, hundreds of thousands of cars of one type being manufactured), the design analysis is very detailed based on accurate loads where the factor of safety can then be reduced to a minimum to reduce production costs. Whereas in low volume production (*eg*, the RDU with only one being made to a particular set of specifications) where the loads are not as well-defined and where there is not as much time available to do as detailed an analysis, what needs to be done then is to increase the factors of safety to take

account of the designer's likely ignorance of all the loading details, the manner of application, the exact forces, moments and stresses generated within the structure and so on, in order to avoid a potential failure. Basically Dr Eccles's experience is that if low factors of safety are used, then very detailed design analysis, very comprehensive load analysis and also extensive testing is necessary to prove the design. Otherwise, the safety factors must be increased to cater for the "unknown" so to speak. I cannot agree more.

197 What is most troubling from the design perspective is that Dr Yang Ting admitted that no calculations were done for the stresses on the bolted joints, the gears and the main bearing housing although ABSG was supposed to have reviewed in particular "Section 6 – Structural Calculations in the Confidential Bundle to verify compliance with the applicable requirements in the AISC Code". The calculations in Section 6 in the Confidential Bundle were mainly to establish that the towers and struts of the RDU were strong enough and would not break under load. It appears that the Defendant failed to perform the necessary design calculations for many other critical components of the RDU.

198 Mr Raymond Drago pointed out numerous deficiencies in the drawings of the gears to indicate that the gears and pinion shaft were not properly designed. I list a sample of some of his comments for the gear in drawing no: PH07-0074-22001-REV C01:

3. Drawing does not define part markings, location and type.

4. Drawing does not define Standard Tolerances for un-toleranced dimensions.

5. Drawing does not define the Specification which governs Geometric Dimensioning and Tolerancing (GD&T) tolerance system (i.e.: ASME Y14.5M-1994 or similar).

6. Drawing does not identify direction of rotation (D.O.R).

7. Drawing does not define Specification which governs welding operations.

8. Note #4 does not adequately define welding of the Flange to Pinion Shaft:

   a. Length of bead stated as "MIN. 15MM", what about "MAX" if any? If welded all around, then it must be stated so.

   b. ALL welding details must be specifically defined on the drawing by use of the appropriate welding symbol callouts.

.....

19. Heat treatment Note #5 is completely inadequate. Who is to decide the time, the temperature and the type of quench??? Must be specifically defined on drawing2 !

20. Note #3 states that the forging be 100% UT inspected:

   a. To what specification?

   b. To what rejection criteria?

   c. What indications are recorded?

   d. What indications are reported?

e. Who is qualified to perform the UT inspection?

…

25. The root diameter of the gear teeth is not defined. This is a critical parameter in determining the bending strength of the teeth and must be specified explicitly.

26. The configuration of the tooth root fillet is not defined. This is a critical parameter in determining the bending strength of the teeth and must be specified explicitly.

27. The gear tooth geometry data table is incomplete. Missing important parameters include tooth thickness and the method to be used to measure tooth thickness, backlash with the mating member, center distance on which stated backlash is to be achieved, and the part number of the mating gear.

199 In the light of the paucity of records showing detailed calculations of all significant aspects of the RDU design, I have serious doubts that comprehensive calculations on all necessary aspects of the design had in fact been done to ensure that the Plaintiff's RDU, albeit a modified design of the Punj Lloyd RDU, would be able to meet its design specifications and perform up to its expectations in terms of its merchantable quality and fitness for purpose. The following closing submission of the Defendant at [44] suggests that, if at all, only minimal re-design/engineering work was done by Ms Tan Sin Liu, who was a junior engineer at that time, before she modified the earlier Punj Lloyd design by Hu Xuya into a design for the Plaintiff's RDU:

> [Ms Tan Sin Liu] said she had used Mr Hu's earlier design from the Punj Lloyd RDU and her Design Manager Mr Lei had countered-signed all her plans…

Since Mr Lei had merely countered-signed Ms Tan Sin Liu's plans and had considered the specifications of the two units to be identical (when they clearly were not), it could not have been the case that Mr Lei was involved substantially in any re-design work in the course of supervising Ms Tan Sin Liu. Accordingly, I accept the Plaintiff's submission that the ineluctable inference is that the Defendant was professing to carry out design work specific to the unit to be manufactured for the Plaintiff, when in reality, the principal designer, Hu Xuya, was working on something else, and the junior designer, Ms Tan Sin Liu, was being supervised by Mr Lei, who did not consider there to be any difference between the two RDUs (which clearly was not the case from a design perspective).

200 Having regard to the totality of the evidence before me, I find that the Defendant failed to perform the required design calculations for many other critical components of the RDU.

Cause of the catastrophic failure on 20 May 2009

201 Much time was spent during the trial by the parties' experts to try to establish the cause of the catastrophic failure of the RDU on 20 May 2009 when the hydraulic drive motor and gear assembly on Tower A of the RDU broke loose from its mounting and fell down mid-way during the umbilical laying operation involving the second reel. Apparently, there was noise heard prior to the failure, including a "high departure angle" observed on the umbilical where it left the chute at the aft of the vessel, though whether it was high departure angle relative to the horizontal or to the vertical was seriously disputed. A high departure angle observed on the umbilical as measured below the horizontal indicates "low tension" in the umbilical whereas a high departure angle measured upwards from the vertical indicates "high tension" in the umbilical. It is unclear to me from the ship's log what the high

departure angle meant exactly when it was recorded at 1800 hrs that "*[v]essel all stop as departure angle on the umbilical is high even though the tension dial on the control panel is at 0*." That the tension dial on the control panel was at "0" however suggests that it is more likely than not that the tension in the umbilical was low rather than high, ***unless*** the tension control system was also malfunctioning at that time. At 1712 hrs, the vessel commenced at a very slow speed of lay at 100m/hour. At 1752 hrs, the vessel was to increase the lay speed progressively at 100m per hour increment at that time. So 8 minutes later at 1800 hrs, I do not think they would have increased the vessel's speed very much yet and the vessel was probably not much above the speed of 100m per hour. At such a slow speed of lay, I doubt that the umbilical tension could have been very high. More significantly, Mr Halvorsen who was on board the vessel at the material time and who should be familiar with the terminology used by the crew in the ship's log, testified that:

(a) The departure angle was measured from the horizontal *ie*, the angle of the umbilical as it entered the water.

(b) The high departure angle observed at that material time indicated that the umbilical was slack, and thus the line tension was low.

(c) Three operators on board the vessel would make sure daily maintenance and checks were done on the towers of the RDU, as well as the hydraulic power pack and the other components. They would monitor the chute as well**.**

(d) As the umbilical laying operation was continuously monitored, someone on the vessel would have noticed if the umbilical should become very taut.

(e) The dynamic positioning system on the vessel would have recorded if there was any sudden exceptional load on the system. Furthermore, the operators on board the vessel had not communicated that the umbilical was "taut".

Taking all everything into account, I am more inclined to accept the position of the Plaintiff that the tension in the umbilical was actually low rather than high and the large departure angle described in the log at 1800 hrs was actually referenced from the horizontal rather than the vertical.

202 As the persons who were present on the vessel during the accident and who might have witnessed the accident were not available to give evidence (a) on what exactly happened; (b) on the actual umbilical departure angles at the material time; (c) on why they did what they did; and (d) to explain more clearly the records of the vessel's daily logs of events which were brief and not sufficiently comprehensive, it was rather difficult to have a clear picture of the events preceding and during the accident on 20 May 2009. There were so many possible hypotheses of what could have gone wrong: loosening of the bolts holding down the hydraulic drive motors; initial misalignment of the two towers prior to hoisting the 300 tons reel; inherent manufacturing defects in the form of heat treatment cracks in the gears leading to the gear teeth breaking off first due to fatigue; a sudden snagging of the umbilical cord on the seabed or a sudden jamming of the rotating reel during umbilical laying when the vessel was still moving forward resulting in a significant increase in the umbilical tension; a possible malfunction in the hydraulics leading to a sudden application of the mechanical brakes or a sudden manual application of the mechanical brakes, whilst the reel was still rotating resulting in very high shear forces acting on the bolts holding down the hydraulic drive motors; possible negligence or mistakes of any of the operators when operating the RDU such that the RDU's loading and operating parameters were exceeded; and the list could go on.

203 The possibility of a sudden snagging of the umbilical because of some large rock, obstruction or

shipwreck on the seabed was eliminated because according to Mr Halvorsen, a prior seabed survey along the umbilical laying route had been carried out and the route was free of such obstructions. This essentially puts to rest the Defendant's submission that the failure resulted from some "*sudden unexpected increase in force in the umbilical being laid coupled with the use of the RDU without a tensioner*" possibly due to some obstruction lying on the seabed. The Defendant relies, *inter alia*, on Mr Hooper's hypothesis that the tension in the umbilical had risen to such a level that the RDU tower structure had in fact slipped aft along the skid rails during the course of the umbilical laying on 20 May 2009 and the crew had to weld down the towers to the rail apparently to prevent the RDU from further skidding towards the stern of the vessel. The Plaintiff's calculations showed however that the clamps as designed by the Defendant would have been strong enough to prevent any slipping of the tower structure towards the aft even if the umbilical tension reached the breaking force for the umbilical. In other words, the umbilical would have snapped before the RDU could slip. These calculations would rule out Mr Hooper's hypothesis that the RDU tower structure had skidded because some high tension had suddenly developed in the umbilical. I further note that the log entry at 1500 hours on 20 May 2009 in the ship's main log of events that was relied upon by Mr Hooper to justify his hypothesis of the tower's slippage along the rail did not in fact say that there was any tower slippage or that the welding of the back plates onto the reel track was to prevent slippage. Instead the log of events stated that the welding down of the Towers to the reel track was to assist with *alignment*:

> Hold back plates welded onto reel track to assist with alignment.

204 Even after eliminating the above possibilities offered by the Defendant, so many other possibilities still remain and it is a very difficult task evidentially to eliminate them one by one. The problem I have in coming to grips with the Defendant's position is that it remains unclear to me what had caused the sudden unexpected increase in the force in the umbilical. The sudden increase in force is just an intermediate link in the chain of causation and does not go back far enough along the causation chain to tell me the possible origins for that sudden unexpected increase in force that the Defendant has hypothesised.

205 Then Mr Hooper asserted that the sudden application of the brakes had caused sudden high stresses to develop:

Court:	…Do I get you right, that what your position is, is the sudden application of a braking force. That means from the operator's control, he had suddenly stopped the running – stopped the turning of the reel through the motor, the hydraulic motor, and because of the inertia, the [angular] inertia of that reel, it is so huge and the sudden stopping of that reel, plus the existing tensions that may be in the cable, raised the stresses on the gears so high that the gears broke at the same time the bolts came loose. Would that be right, as a summary of your position?
Mr Hooper:	That's a fair statement, sir.

However, nothing in the Defendant's Closing Submissions explains why the brakes were applied suddenly by the RDU operators at that time. If that is Mr Hooper's reason for the catastrophic failure, then I have this to say. The RDU is designed with a braking system, naturally to be available for use in an emergency. I wonder what good is a braking system in the RDU as designed when the mountings of the hydraulic motor may loosen and the hydraulic motor may fall off should the brakes be applied suddenly to a reel that is still rotating. An emergency brake for the RDU must be designed to withstand precisely the huge forces that will result from an emergency stopping of a 300 tons rotating reel. A car driver will be astonished if he is told by the car manufacturer that the car brakes cannot be used for emergency braking as the car brake mountings will then come loose and the car brakes will fall off. If indeed the hydraulic braking system of the RDU cannot be used for emergency braking as Mr Hooper appears to suggest, I would hardly think that the RDU is of merchantable quality and can ever be regarded as being fit for its purpose.

206 In any event, the Plaintiff wisely decided that it is not relying on this incident in its closing submissions to prove its case because it is similarly an uphill task for the Plaintiff to determine the exact cause of the failure with so many possible variables and then to establish a link from the true origins for that failure to prove that the RDU is not of merchantable quality and not fit for its purpose.

207 Even if the Defendant establishes as a fact that the failure is not due to any fault on its part, it is not strictly relevant for my consideration in this case when the Plaintiff does not even assert that the catastrophic failure is due to any design failure on the part of the Defendant or arises from the fact that the RDU is not of merchantable quality and not fit for its purpose. When there is no such assertion by the Plaintiff, I am already assuming, in favour of the Defendant, that the catastrophic failure cannot be attributed to the Defendant in any way.

208 As such, it is not necessary for me to deal with the submissions of the parties that try to establish the true source and cause of the failure since the Plaintiff is not relying on that failure to prove its case here. But if I am wrong in not making a finding on the cause of this catastrophic failure in deciding this case, then I would briefly state that on balance I prefer the following expert opinion of Mr Hooper for the reasons he had stated that the failure was ***not*** due to initial gear failure stemming from the pre-existing cracks on the gear teeth arising from poor heat treatment during the manufacturing process (which helps to eliminate only one possibility out of many other possibilities):

> I cannot accept the fact that if these teeth are being pushed in one direction on the pinion, that you get the opposite breakage on the teeth. If you go back to the first picture I brought up, on page 113, which is the first figure I brought up, it's almost that you can see the imprint of the gear wheel crossing it and pushing the bottom section of the teeth to the right and pushing the tips or the ends of the teeth at the top to the left, just on a few teeth where the pinion actually happened to mesh with the gear wheel as it was being thrown out of the machine. So to me this clearly shows the damage to the teeth, the fracture, is caused by the accident, not by the failure of the teeth in situ.

209 However, to the extent that Mr Hooper had relied on the even burring that he noticed across the entire face of the gears to opine that the pinion and gears were therefore all meshing in alignment during the RDU's operation before the sudden catastrophic event, I will disagree with him.

210 Mr Hooper's observations of uniformity in burring are not buttressed by any detailed magnified photographs or actual precise measurements of the burr using suitable calibrated instruments at various points across the width of the face of each gear tooth. Mere visual observation is not a sufficient basis to conclude that there is no misalignment when the tolerances for alignment are limited to, at most, 30 to 40 microns according to Mr Drago, whose opinion on this point I accept. Mr Hooper had not undertaken any measurements of the burring profile to this degree of preciseness and accuracy to be able to persuade me that the gears must have been properly aligned within acceptable gear tolerances just because of the apparent appearance of uniformity of the burr based on his mere visual observation.

211 In fact, the Plaintiff in its reply submissions have referred me to photographic evidence in the Plaintiff's Multimedia Bundle showing very uneven grease patterns observable on the working faces of

the main and pinion gears, which I accept, that do point positively to actual gear misalignment. A bad case of gear misalignment may be readily revealed by the very uneven grease patterns observable with the naked eye, but an apparent appearance of uniformity of grease patterns observed with the naked eye may not necessarily establish that the gear alignment is in fact within the acceptable gear tolerances (which are usually in the order of microns), unless proper instrumentation is used to measure the actual alignment or misalignment of the gears.

212 The Defendant has not disputed the static misalignment results presented in the Nobletech Report. These measurements were performed using calibrated instruments and attested to by ABS. This contrasts with Mr Hooper's assertion of no misalignment based just on a visual appearance of no burring, which I find hard to accept.

213 The Nobletech Report also presented deflection measurements of the main gear with respect to the sub-frame that carries the shaft and bearings. These measurements reveal deflections in the order of several millimetres which are unacceptably large when gear tolerances are in the order of microns. Again the Defendant is not able to dispute these objective measurements of the large deflections which would likely give rise to severe gear misalignment.

214 I also agree with the following explanation provided under "SECTION 3 – GEAR TRAIN VISUAL INSPECTION" of the Nobletech Report:

> The gear and pinion are visually inspected for any obvious defects, cracks, chips, or signs of stress. Any such visible defects are noted and photographed. Also, the grease contact pattern on the flanks of the gear and pinion teeth is inspected after the tests. A normally running aligned gear system will show an even wear patterns and an even pattern across the sides of the gear teeth where the grease on the pinion and gear gets squeezed out between the gear and pinion teeth when they come into contact during operation. Any unevenness is noted and photographed.

215 The grease patterns on the gears were observed and photographed during the testing by Nobletech Marine Offshore Pte Ltd to be "non-uniform". I accept these observations to be supporting circumstantial evidence that there was in fact misalignment of the gears prior to failure, though I believe that neither the misalignment nor the existence of heat treatment cracks on the gears was the cause or the primary initial cause of the catastrophic failure.

216 I am more inclined to take the view that the primary initial cause of the failure was the loosening of the securing bolts for the Sauer Danfoss hydraulic motor that caused subsequent massive misalignment and consequential gear tooth failure, before the motor came off its mounting and fell down from Tower A towards the deck. The gear tooth fractures probably came after the bolts had loosened, and not before, for the reasons stated by Mr Hooper. I tend to agree with the conclusions reached by Vince and Pratt from Pirtek as detailed by Mr Halvorsen at [91] and [93] of his AEIC (though it is strictly hearsay evidence from Mr Halvorsen):

> 91. Following a detailed inspection of the failed RDU system on the deck of the vessel, both Pratt and Vincent reported back to me that several bolts were loose on the system and that the likely cause of the incident was that the bolts had come out and caused the drive motor to misalign and shear off.
>
> …
>
> 93. Pratt from Pirtek issued his inspection report on 21 May 2009:

First inspection of the drive motors on the starboard side found to have the main bolts holding the gearbox in place were loose also the bolts holding in the bearing housing were also loose and 1 bolt missing.

When Vince and I checked the bolts on the drive motors on the Port side the left hand drive motor had 3 bolts on the left side were loose and the 3 on the right side were tight all bolts and that bearing cover were also loose.

The main drive gear on the port side also had the centre bolts loose.

All bolts on the port side have been tightened and broken bolts replaced.

We have isolated the starboard side removing the right hand motor and isolating the hydraulic lines.

A test has been conducted to see if the port side is in working order and there are no hydraulic leaks, the test has been successful.

My conclusion on the drive motor failure is that the cause was from the bolts being loose therefore causing the drive motor to twist and get damaged by the force of the gears not aligning properly and it breaking the mounts on the gearbox, damaging the gears, gearbox and hydraulic hoses and fittings.

[emphasis in original]

217 The conclusion by Vince and Pratt above (underlined in bold) makes engineering sense to me. But there again if a further question is asked on what force or combination of forces acting over time had caused the securing bolts to loosen in the first place, there are going to be numerous possibilities and the answer is not at all clear cut because it will likely be the totality of the forces arising from the possible combination of motor torque, gear misalignment, vibration during the hydraulic motor's operation, shear forces acting on the bolts due to the occasional sudden starting and stopping, acceleration or deceleration of the massive 300 tons reel in the course of umbilical laying, inherent inertial and gravitational forces due to wave action that generate deflections within the structure and drivetrain, tension in the umbilical cord and other forces acting over a period of time that were eventually transmitted to the bolts to loosen them. These are likely to be the forces that the RDU would be subjected to during its operation. It may not even be possible to identify which is the predominant force/factor that caused the bolts to loosen in the first place. Another real possibility is that the bolts themselves were not properly and securely fastened or tightened at the time of manufacture. Furthermore, should the bolting arrangement at the mounting of the hydraulic motor be under-designed, then these forces acting in combination would more readily cause the bolts to loosen over a relatively shorter period of time. I do not wish to hazard any guess here as to what combination of possible factors or forces caused the securing bolts eventually to loosen for one of the Sauer Danfoss motors at Tower A. Whatever it is, this failure may also indirectly indicate that there may be some inadequacy with the bolting arrangements for all four Sauer Danfoss motors on the RDU (two on Tower A and two on Tower B) because the mounting bolts were similarly found to be loose after an inspection of Tower B as well, as can be seen at [98] of Mr Halvorsen's AEIC:

98. I did not receive a response. At 2.23pm (on 21 May 2009), I emailed Yang Ting again:

Yang Ting,

The technician and mechanical engineer have commenced inspection of Tower B. Preliminary findings are that all three mounting bolts on one side of the forward motor was loose. Also one bolt head was missing. The hydraulic hose connections are also being checked and they have reported that some of these are also not fully tightened.

218 However, the Defendant claims that the RDU must necessarily be of merchantable quality and fit for its purpose simply because:

(a) the cause of the catastrophic failure at tower A cannot be attributed to the Defendant;

(b) the RDU had run for approximately 32 hours and had laid 8,644 m of umbilical before the catastrophic failure on 20 May 2009; and

(c) the RDU was able to complete the laying of the umbilical for the rest of the Longtom project using only one tower B.

219 On point (a) above, I have already assumed that the catastrophic failure at tower A is not to be attributable to the Defendant solely on the basis of the Plaintiff's position taken in its closing submissions that it will not place reliance on the failure to prove its case.

220 However, if I have to make a specific finding on what part of the RDU gave way *physically* first, then I am inclined to believe that the mounting bolts loosened first, which led to misalignment, and the crushing of the gears and it finally ended up with the motor falling off its mounting. I agree with Mr Hooper that it was not the gear teeth physically breaking first, which led to the mounting bolts loosening and the motor falling off its mounting. The fact that the mounting bolts loosened first may also possibly indicate that the bolting arrangements for all the four Sauer Danfoss motors on the RDU were inadequately designed and/or the bolts were not securely tightened at the time of fabrication prior to delivery to withstand the ***normal*** forces arising from the umbilical laying out operations out in the Bass Straits at that time, in which event the Defendant would have to be made liable for the catastrophic failure. I do not believe that the RDU had met with any unknown and unexplained sudden massive force outside of the ***normal*** forces present during the rather unexceptional umbilical laying operations at that time. Had such an unknown and unexplained sudden massive force existed, which I think not, then blame for the catastrophic failure rested, not with the Defendant, but elsewhere. If it were otherwise, then it is likely for the blame to rest with the Defendant because of an under-design of the mounting bolts and/or a failure by the Defendant to properly tighten all the mounting bolts prior to delivery to enable the mounting bolts to withstand the ***normal*** forces that the bolts would be subjected to in the course of the RDU's normal umbilical laying operations at that time.

221 On point (b) above, I do not think that one can infer *per se* from a mere 32 hours of operation before the catastrophic failure that the RDU must necessarily be of merchantable quality and fit for its purpose. If I have to infer anything at all, the more likely inference is to the contrary that it is not. 32 hours is such a short period of operation that one would not likely expect any RDU that is of merchantable quality and fit for its purpose to suffer from such catastrophic failure in such a short time. As I have said, I am not going to hold it against the Defendant as the Plaintiff is not relying on the catastrophic failure at tower A to prove its case.

222 On point (c) above, the fact that the RDU completed the laying of the umbilical for the rest of the Longtom project using only one tower B does not mean anything at all to demonstrate the ability of the RDU to meet its design specifications or to show that it is of merchantable quality and fit for its purpose. It is clear that the personnel operating the RDU with only one tower B had to be and were extra careful and gentle in operating the RDU during the remainder of the umbilical laying operations.

They had to lay the umbilical at well below the maximum rated umbilical tension of 15 tonnes, and they had to keep checking and tightening the bolts particularly at the mountings of the Sauer Danfoss motors on the RDU on Tower B, which kept loosening. How could this manner of operation in order to complete the Longtom project ever demonstrate that the RDU remains of merchantable quality and fit for its purpose when it is operated close to but not beyond the maximum limits allowed by *its design specifications*? It is akin to saying that passing a load test of one tonne will be sufficient to demonstrate the equipment's fitness of purpose and merchantable quality when the specified maximum operating load for the equipment is actually ten times higher at 10 tonnes.

Fraudulent STAAD.Pro Modelling to obtain ABSG certification

223 I will now consider whether the Defendant had been reckless, dishonest or fraudulent in the manner it secured its ABSG certification for the RDU.

Misrepresentation that no wind load was to be considered

224 The Plaintiff submits that the Defendant falsely represented to Dr Liu Li (who produced the STAAD.Pro analysis for certification by ABSG) that the RDU was supposedly intended to be operated "*in the cabine [cabin]*" and as such wind effects need not be considered and the same false information was in turn conveyed to ABSG who then certified the RDU design without taking into account wind loads.

225 Ms Renuka Devi from ABSG emailed the Defendant on 18 February 2008 with the following query:

> Please clarify the following comments to complete our design review
>
> 3. Both transit and operating conditions, No wind load has been considered – Please clarify

226 Dr Liu Li sent the following email on the same day in response to Ms Renuka Devi's query, which was copied to the Defendant:

> As per the requirement by [the Defendant], wind load was not considered in the report. [The Defendant] suggested that spooler located in the cabine, hence there is no wind load applied.

227 The Defendant admits that it did not design for wind load for the RDU, but submits that it was due some miscommunication between Mr Lei and Dr Liu Li. There was no grand plan to defraud on the design by stinting on the design for wind. It was simply an omission on the part of the Defendant.

228 I am not persuaded that there was miscommunication. In the first place, both Mr Lei and Dr Liu Li read and speak English. They are able to testify fairly fluently in English. No evidence was produced to show me how the alleged miscommunication between the two of them could possibly have arisen. In my view, there is nothing so complicated about a simple written query to clarify whether wind load had been considered for both transit and operating conditions. Furthermore, there was time to read and to re-read the email if necessary. Dr Liu Li's written reply to Ms Renuka Devi was drafted in simple clear English to explain why wind load was not considered: the "spooler" was located in a cabin and hence, there was no need to apply any wind load. It is absolutely false to have described the 300 tons reel to be located within an enclosed cabin on board the ship. During umbilical laying operations, the 300 tons reel would be mounted on the RDU situated on the open deck towards the aft of the vessel, the "Maersk Responder", where the entire RDU structure and the 300 tons reel would be fully exposed to the wind.

229 I accept the Plaintiff's submission that the Defendant and Dr Liu Li were aware that the initial results of the STAAD.Pro analysis showed that the unity checks were not met and that fixed joints had to be revised to non-existent ball joints to enable the structural model of the RDU to barely pass the unity checks. The Defendant and Dr Liu Li must have been aware that transverse wind loads acting on the 9.2 m diameter reel mounted on the RDU had to be omitted to avoid a failure of the unity checks. Hence, it was not a mere miscommunication. I conclude that Dr Liu Li and the Defendant did not want the wind load to be considered at that time because they knew that the additional wind load would result in a failure of the unity checks. So it was deliberately and dishonestly misrepresented to Ms Renuka Devi that there was no need to consider wind load because the reel (or spooler) was to be housed within an enclosed cabin in the ship, when they clearly knew it was not the case.

Inaccurate modelling of fixed joints as ball joints

230 The Defendant submits that Dr Liu Li was simply exercising his independent "professional judgment" in treating certain bolted fixed joints as ball joints in his FEA modelling. These were the joints located where the eight transverse struts were bolted to the main base of the RDU. Dr Liu Li denied colluding with the Defendant to give the Defendant a favourable FEA report just to satisfy ABSG while knowing that the design was inadequate.

231 What is not disputed is that Dr Liu Li's preliminary report sent to the Defendant on 26 October 2007 showed that the RDU design failed the STAAD.Pro analysis in that various structural members had exceeded the unity check value of 1. Dr Liu Li thereafter discussed with the Defendant on how to resolve this. No records of any emails or notes of the discussions were produced to show what exactly transpired between them. As the Defendant had already started procuring the constituent parts of the RDU by 23 October 2007, there was limited scope by early November 2007 to substantially re-design the RDU, which might result in a delay in delivery.

232 As a qualified structural engineer, Dr Liu Li must have known that the type or nature of the boundary conditions specified for the STAAD.Pro analysis would influence the final results. He understood the difference between a joint fixed in all directions (*ie*, a fixed joint) and a ball joint rotatable in all 3 directions (*ie*, on the X, Y and Z axis). Based on his experience and professional judgment, Dr Liu Li opined that it was reasonable to change the fixed bolted joints to ball joints for his revised STAAD.Pro analysis of the RDU that he sent to the Defendant a week later on 2 November 2007. He said that he took "*a long time to study, to check whether is it okay or not …*" to change the model from fixed joints to ball joints. With the revision to ball joints, the RDU passed the unity checks.

233 I believe that Dr Liu Li and the Defendant must have discussed at some length on how to get the RDU structure to pass the STAAD.Pro analysis and the unity checks with minimal change to the actual structural design, as any major changes would probably involve additional cost and delay for the Defendant. The easiest way out of the quandary is to tinker with the input criteria or the input boundary conditions for the joints in the RDU model to be used for the STAAD.Pro analysis, which then changes the way the forces and the stresses are redistributed throughout the finite elements modelling the RDU structure under the same design loads.

234 I gave Dr Liu Li every opportunity to explain how he exercised his professional judgment to support his remodelling from fixed joints to ball joints and what basis he had in mind at that time so that I can better decide whether the change he made was in fact based on a careful and proper engineering analysis rather than a motivation simply to pass the unity checks regardless whether it was reasonable to do so from an engineering perspective, which would in my opinion be a rather

reckless thing to do.

235 It is significant to note that the STAAD.Pro programme itself allows for "Partial Moment Release" corresponding to the rotational degrees of freedom in the MX, MY and MZ axis where there is a reduction in the stiffness of a fixed joint by a factor of "f1". I do not think that Dr Liu Li would be unaware of this facility within the STAAD.Pro programme. Further, the notes to the STAAD.Pro programme provide that "*it may be necessary for the user to perform a few trials in order to arrive at the right value of f1, which results in the desired reduction in moment*." The following verbatim evidence of Dr Liu Li shows that he could not give a satisfactory explanation on how he managed, without doing any calculations or test trials, to simply assume that the correct factor "f1" must be "zero" in value and then change his modelling criterion from a totally rigid fixed joint in his initial preliminary FEA model to a totally rotatable ball joint incapable of resisting any moments in each of the 3 axes *ie*, MX, MY and MZ axis, when it was already clear from the Defendant's structural drawings presented to him that the bolted joints comprise rather solid steel plated structures bolted together, which even if treated as slightly flexible structurally could not possibly be regarded as having absolutely no capacity whatsoever of resisting or transmitting any moments that would justify a total release to zero moments at these bolted joints for his revised FEA using the STAAD. Pro programme:

Q: So, Dr Liu, you are telling us that 26 October you sent the preliminary results to PH showing that it didn't pass STAAD.Pro. We know that on 2 November, you then sent a file where the releases are changed and fully released in MX, MY, MZ. You are now telling us that in that period, you didn't do any trials to arrive at the right value of the release. In fact, all you did was go from fully fixed to fully moveable in all three directions, MX, MY, MZ. Is that correct?

A. In fact I told you before, the first time I submitted only for the primary analysis, just to show the progress, what I have done, right? Then later I -- based on my study, I think this one is more suitable for the pin joint. That's why I modified. I think it's reasonable.

Q. But you did that modification without any -- running any trials. Correct?

COURT: The question is, how do you come from 100 per cent to 0 per cent without doing any trials? It may well be 50 per cent. You shouldn't have gone all the way to zero until you do a [trial] and know whether it's 50 per cent release, 75 per cent release, or only 25 per cent release, depending on the stiffness of the foundation and how it's attached. Without the size of the foundation plate, the connecting foundation plate, and so on and so forth, how do you zoom from maximum [moment] to no [moment]? How do you do that without doing any tests? Experience cannot substitute for an actual factor you use. Because experience can only tell you "I can do so", but it may not be able to tell you how much.

A. Okay, sir, may I consider, if first time I use the pin joint and everything is fine, there's no question, right? Can I say that?

MR TAN: Yes.

A. I don't know why --

COURT: Do you mean to say certain things --

A. First time, I use the pin, you think it's fine. Then first time I do the primary analysis, and then second I think this is reasonable, I change, you say "You are wrong." So I --

COURT: No, no. The question is not where you start from. It doesn't matter where you start from, but eventually whatever you do, you must model what exactly is being built. That's the main point. So the question is, if you started with a pin joint, then when you go and look at the structure and you say "No, this is not really a pin joint; it is more like a fixed joint", then you may then have to modify your first model to be more exact. In fact the best way to start off by fixing it exact -- more close -- as close as possible to what exactly it is, rather than start with something which it is not, and then modify. Of course sometimes when you have between zero and one, you look at the joint, it's somewhere in between; you may have to do something to check what should be the value, and then put it in the STAAD.Pro to start off with. Model it right in the first place. That's the point. I don't care whether it's a pin joint or fixed joint, eventually you must model it exactly as what it is.

A. Yes, sir, I told you -- I said before, this only preliminary design, and normally, you see, if I modelled as the fixed, I didn't do anything; I just draw line, connect. Right. Opening moment, we have considered. Then, if I have to consider this as the pin joint, I have to take a long time to study, to check whether is it okay or not. In fact maybe this affect me is the one I'm doing it at that time. This hose reel, before I do something, I do some drawings and some reference, so I saw it also designed in such a way. So I think maybe it's reasonable. Yes, I should do some more trials, and even if -- based on my logic at this time, I do not try. I just do two times. One pin, one fixed. All pass. That's all. But at that time I think a pin is much more suitable. That's what I think.

COURT: Of course, usually when you model an actual structure which is somewhere between pin and somewhere between fixed, the best way, of course, a shortcut would be to model the worst-case scenario first. If it passes the worst-case scenario, it passes, and you don't care about the rest. And between pin and fixed, the worst-case scenario is always the fixed, in terms of stresses.

A. Yes, I learned from this case.

[My comment: This suggests that he did not perform the necessary calculations or tests at the material time.]

COURT: That's why most people start off with fixed.

A. No, sir. In fact for most of the case, pin is more conservative. Really. If you are interested, maybe later I find some material for you.

COURT: You think the same pin is -- you get higher stresses than --

A. Yes, right. For some cases. I can show you. Yes. For some cases, pin is more conservative, so I changed it. At that time I do the design --

COURT: In this case, which is more conservative: pin or fixed?

A. So this one, I checked online before, for these two cases. For some cases, pin is conservative; for some cases, fixed is conservative. It depends on what is your structure design.

COURT: In this case?

A. In this case, I have no idea.

[My comment: This shows that Dr Liu Li had no idea at that time and remained so even at the trial as to what would be correct position to adopt for the Plaintiff's RDU. Yet he simply proceeded to model on the assumption that it would be correct to use "f1" as zero for the bolted joints in the RDU, which essentially converts them into ball joints for the STAAD.Pro FEA analysis. It shows the recklessness involved in seeking to pass the unity checks.]

COURT: You have no idea which one is the worst-case scenario, whether it's all pin or all fixed?

A. This --

COURT: Because I'm sure the structure is somewhere in between the two, not at both ends. Whether it's closer to fixed or closer to pin is a matter of debate.

A. For this one, I think for -- you see, for this brace, compared to the whole thing, it's not so stronger. So at that time I do the design, and the vendor also only provide us the XYZ. They didn't --

COURT: Did you know that XYZ are the forces? It's just that given the drawing of the structure, is it closer to a model for a -- I mean, when you model as totally fixed or model as totally pin, which one is more accurate in this case? Which one is the more conservative design, where you can do it once and then forget about it?

A. But if now I do this project, I just use two. I don't care whether -- which one is conservative. I just do one for the bolt, one for the fixed, and which one is concerned didn't pass –

COURT: If you pass both, obviously you pass -- you will pass both.

A. Yes. Because --

COURT: Because this is probably somewhere in between, you see.

A. Anyway, we have to pass. Right? What we do is to make this structure sufficient and make structure to meet the criteria. Right? At this time, I think this one is reasonable, so in fact I'm not so concerned about the -- whether they pass or not. If it cannot pass, I just ask them to modify.

COURT: Yes, I know that.

A.	It's easy. For example, if I have to change to the pin, I have to find the -- I have to determine this one as a pin. I have to make a lot of modification. If I change the size, it's very simple; I just define the size and apply. That's all. Just give me one minute, two minutes, I can finish it. But if I have to define the, pin I have to see whether this is suitable -- you see, I didn't consider as the big structure as the pin; only small one. Right? So I have to consider this one. **I didn't do the exactly calculation at the time. I see it just based on my experience and based on what I have done before. So maybe it's not so reasonable, but I think it's practical.**

[My comment: This evidence confirms that Dr Liu Li did not perform any calculations at the material time to determine whether it would be appropriate to convert the bolted joints into purely ball joints.]

COURT:	Anyway, he has given his answer. He did not do the calculations, but based on his experience, he thinks that the pin is reasonable at the end, considering the structure at that point.
MR TAN:	Yes.

[Emphasis added in bold]

236 Dr Liu Li tried to rely on certain literature that he produced (*ie*, Exhibits D29 and D30) to argue that his modelling of the fixed joints as ball joints was permissible. However the literature he produced says nothing about modelling a rigid fixed joint as a ball joint in particular for the FEA. Instead, D29 suggests to the contrary that the end-plates at the bolted connections should be relatively thin and retain sufficient flexibility and ductility for it to be classified as a simple pin connection. Hence, the connection itself must be designed for rotational capacity before it could be regarded nominally as a pin joint. In the present case where the bolted connections were meant to be solid heavy duty rigid bolted connections (very different from those shown in D29) and had not been designed to have rotational capacity, there is in my view clearly no technical justification to treat them as completely ball joints capable of rotating freely in all directions for the purpose of the FEA analysis.

237 D30 itself states that the following two requirements must be satisfied before classifying a connection as nominally pinned, based on its strength:

(a) The design moment resistance of the connection does not exceed 25% of the design moment resistance required for a full-strength joint.

(b) The joint should be capable of accepting the rotations resulting from the design loads.

238 Dr Liu Li did no such calculations for the rigid bolted connections for the RDU before he proceeded to treat them as ball joints for the FEA based on his “judgment”. If Dr Liu Li had simply assumed a “f1” factor of say 0.5 or even a figure lower than 0.5 based on his judgment without doing any calculations, I would not have considered him dishonest as it could arguably still be regarded as a considered “judgment” call. I would not immediately fault him as a bolted joint has characteristics somewhere between a ball joint and a totally rigid welded fixed joint. In my view, the bolted joints for the RDU are probably much closer to a totally fixed welded joint than to a fully rotatable ball joint. Given the multiple bolts at each joint, the fact that they were meant to be tightened very tightly and

given the thick steel flanges and the manner of construction, it is clearly absurd to regard them as ball joints, fully rotatable in all 3 directions. When Dr Liu Li simply uses zero for the "f1" factor, he is in fact making an absurd engineering assumption that the fixed joint is the same time as a ball joint or has become one and behaves structurally in the same manner as a ball joint. I can no longer believe that Dr Liu Li, a structural engineer of his experience, is making an engineering judgment call at all, when it reaches a high level of absurdity from an engineering perspective. He is not. I am driven to conclude that Dr Liu Li's treatment of those bolted joints as completely ball joints for the STAAD.Pro analysis is due more to a deliberate and dishonest effort to try to get a pass for the unity checks, without any regard whatsoever of the true nature of the joints at those locations in the RDU. Dr Liu Li is an experienced STAAD.Pro user and yet when I asked if a partial release could be modelled in the STAAD.Pro, he avoided giving me a straight answer and then suggested that this sort of thing was "very difficult to do in the industry", when the truth is that the STAAD.Pro manual shows exactly how it is to be done. I agree with the Plaintiff's submission that Dr Liu Li had sought to portray the partial release modelling as being difficult to perform in an attempt to justify his unjustifiable choice of full moment release, which magically changed the solidly bolted joints in the RDU into fully rotatable ball joints for his FEA modelling of the RDU.

239 Accordingly, I find that the Defendant and Dr Liu Li had a clear motive to achieve the unity check of 1 for every beam in the RDU structure and given the constraints facing them, an easy way out was found to pass the unity checks by simply changing the rigid fixed joints to ball joints although they clearly knew that the actual RDU structure was to be constructed with rigid fixed bolted joints that did not have a rotational capacity anywhere near freely rotating ball joints. From the very beginning of the design stage to the end of the construction stage, the Defendant never at any time intended to construct any ball joints for the RDU. Dr Liu Li cannot now claim that he had no knowledge of that. Reluctantly, I have to infer that they were less than honest in their approach in knowingly setting the wrong boundary conditions for the revised FEA modelling using the STAAD.Pro programme just for the RDU to pass the unity checks.

240 The revised STAAD.Pro file of Dr Liu Li premised on an inaccurate FEA modelling with ball joints was eventually submitted by the Defendant to ABSG for their review on 18 February 2008, and the Defendant managed to secure ABSG's certification on that basis. ABSG was thus misled by the Defendant into issuing the ABSG certificate. The ABSG certificate issued in respect of an RDU design containing ball joints would not be valid for the Plaintiff's RDU as the Defendant eventually built the Plaintiff's RDU without any ball joints.

Inaccurate modelling of RDU reel as fixed in the X direction

241 The Defendant concedes that it had ***wrongly*** modelled the reel in the RDU as having been fixed in the X direction with notional pin joints where the reel meets the towers at each end. Thus if the ship rolls to the portside, the tower on the starboard side is ***wrongly*** assumed to be able to prevent the reel from sliding towards the portside by virtue of the notional pin joint that connects the reel to the starboard tower, thereby resulting in both towers acting together to share the total inertial force and weight of the reel resolved in the X direction. Therefore, the Defendant's STAAD.Pro analysis submitted to ABSG had ***erroneously*** computed the unity checks on the basis that there was a reduction in the transverse force acting on each tower due to a sharing of the transverse load by both towers and a corresponding reduction in the stresses within the structural elements and joints in each tower, which is not the case.

242 In reality, the RDU had been constructed such that the 300 tons reel would ***in fact*** be capable of sliding transversely in the X direction when subjected to the roll motion of the ship. There is in fact no pin joint attaching each side of the reel to the respective side of each tower that allows, for

instance, tower A to assist in pulling the reel back if the reel slides away from tower A in the X direction. After sliding towards and leaning against tower B fully, all the gravitational and inertial forces arising from the reel resolved in the X direction will bear down entirely on tower B. These loads will not be shared by tower A on the opposite side. With the ship rolling to the other side, the converse happens.

243 The net result I find is that the inaccurate modelling of the RDU severely underestimates the forces in the X direction on the towers for the purpose of the FEA. Ms Renuka Devi accepted that the whole load analysis for the STAAD.Pro programme would be changed. ABSG was requested to redo the STAAD.Pro analysis with the correct as-built connections for the reel to the towers. Ms Renuka Devi confirmed that ABSG would have rejected the design and informed the Defendant that the RDU would need to be redesigned. Ms Renuka Devi further stated that she would not have taken the effects of friction into account because the RDU was operating in a marine environment. As can be seen on Exhibit D3 (reproduced in the Table at [38] above), with the reel sliding and with a coefficient of friction at zero, the RDU would have failed the unity checks. It is not surprising to me that Ms Renuka Devi took the position similar to that of the Plaintiff's experts, that friction resistance would not be considered by a prudent designer in mitigation of a transverse inertial force generated in a marine environment, where the ship would heave, roll and pitch out at sea, and the mitigating friction force itself might not even be a constant due to changes in the reel's 300 tons deadweight reaction force at the friction contact surface as the ship heaves and rolls at different angles and with different accelerations. I share the views of Ms Renuka Devi and the Plaintiff's experts that reliance on a variable friction force to prevent stress overload in a ship mounted structure due to forces generated by wave action is simply not prudent engineering practice.

244 Clearly, it is the result of the inaccurate model presented by the Defendant to ABSG that enabled the Defendant to secure the ABSG certification for the RDU structure based on the STAAD.Pro analysis. The more difficult question is whether the depiction of the reel as non-sliding in the X direction was mere gross negligence or sheer incompetence on the part of the Defendant and Dr Liu Li, or was it an error already known to them before or at the time of submission to ABSG of the Confidential Bundle (containing the design parameters and Dr Liu Li's STAAD.Pro analysis) for ABSG's review and certification. In short, was it innocent or deliberate?

245 What is clear to me is that the Defendant and Dr Liu Li must have known from the way the RDU was to be constructed that the connection between the reel and the tower could not possibly be notionally treated as a pin joint. They must have known that the reel could slide in the X direction. It was so obvious that the reel was not going to be bolted down or secured by any pin or ball joint to the tower. How then could they have possibly made such a fundamental mistake? Much time and conscious effort is needed to perform the STAAD.Pro analysis and to input each of the connection criteria at each node in the FEA. This is not a case where the FEA subcontracted by the Defendant to Dr Liu Li to do was so rushed that an innocent mistake was inadvertently made by Dr Liu Li due to a lack of attention. Dr Liu Li as a structural engineer ought to and would have known that the STAAD.Pro analysis would give very different results if wrong or inaccurate input parameters for the joints were to be set for the STAAD.Pro programme. Had this been the only "mistake" (*ie*, without the intentional non-inclusion of wind load and the intentional erroneous modelling of a fixed bolted joint as a ball joint), I would have given the benefit of doubt to the Defendant and Dr Liu Li that it might have been purely gross negligence on their part. However, after considering the totality of the circumstances, and in particular how the wind load came about to be excluded and how the actual fixed bolted joints turned into ball joints for the revised FEA, I am not prepared to give them this benefit of doubt of gross negligence on account of their erroneous modelling of the reel as fixed between the two towers when it was actually capable of sliding. I find it difficult to accept that it could be an innocent mistake or that Dr Liu Li could be so ignorant or so incompetent as to

consciously stipulate for the STAAD.Pro programme a pin joint connection at each end of the reel to each of the RDU towers where none in fact exists at each of those locations.

246 In conclusion, I am satisfied that the Plaintiff has proved to the requisite high degree of probability that the Defendant had fraudulently modelled some of the critical structural connections of the RDU (*ie*, (a) the reel to tower sliding connection; and (b) the struts to tower base frame bolted connections) and further fraudulently stipulated the absence of wind load on the RDU in the submission of its STAAD.Pro programme to ABSG for its design review, and thus had deliberately and dishonestly misled ABSG into giving ABSG certification for the Defendant's structural design of the RDU, which certification the Defendant would not have otherwise obtained had it modelled the RDU correctly. The Plaintiff was accordingly dishonestly misled into believing that the ABSG certification that it had sought and received for the RDU had been properly obtained by the Defendant from ABSG.

Full ABSG certification was not obtained

247 Mr Derek Chong of ABSG testified on the scope of full certification by ABSG as follows:

> Q. I see. Have you ever come across a situation where ABSG has been asked to fully certify a piece of equipment?
>
> A. What you mean is a full certification?
>
> Q. Yes, what does that mean?
>
> A. Okay, a full certification involves -- in short, I call a EPCIC process, engineering review appraisal, procurement inspection, construction, installation and commissioning. Commissioning includes two types of commissioning, factory acceptance test and site acceptance test.
>
> Q. I see. So what you have been asked to do doesn't come anywhere close to --
>
> A. Not close, yes.
>
> Q. Even if we add on the fact that some of your colleagues were asked to review the design of certain parts, that would also not --
>
> A. Because for review they need to cover all the aspects of engineering, including structure, mechanical and electrical. Structure they need to specify the various codes and standards, included also for mechanical involved piping process and so on. For electrical also have instrumentation control, safety and so on, so there's a wide spectrum of engineering to look into.

248 Thus the design review for full certification by ABSG should have covered, *inter alia*, a review of the structural, mechanical and electrical design of the RDU. Mr Ravi Chandran, the Lead Electrical Engineer of ABSG, testified however that ABSG did not carry out any mechanical design review. ABSG only carried out a structural and electrical design review of the RDU. Accordingly, the Defendant did not obtain full certification from ABSG. The totality of the following evidence leads me to conclude that the Defendant had dishonestly misrepresented to the Plaintiff that the RDU had full certification from ABSG, when it knew that only partial certification was obtained:

(a) The Defendant did not even instruct ABSG to carry out a mechanical design review.

(b) The Defendant did not request ABSG to carry out a hydraulic design review, although Mr

Loh knew that one was necessary for a full certification of this type of equipment.

(c) ABSG's quotation provided by ABSG to Mr Steven Gan stated that ABSG's scope of work only related to the design review of one unit of spooler tower, one unit of power pack skid and the "submitted electrical items". Clearly, the design review of mechanical items and equipment was not included in the scope of work that ABGS quoted for.

(d) Having dealt with ABSG for some ten years on other projects prior to the Plaintiff's RDU, the Defendant would have realised that Mr Ravi Chandran who signed on the certificate provided by ABSG could not have carried out a mechanical design review, as he was an electrical and not a mechanical engineer.

(e) If a mechanical design review was indeed carried out, a separate letter or certificate would be issued by ABSG signed by a mechanical engineer, as well as a separate invoice for the mechanical design review. No such documents were produced by the Defendant.

(f) ABSG had only raised an invoice for the structural design review and an invoice for the electrical design review. The Defendant was not able to produce any ABSG invoice to show that it had been billed for any mechanical design review of the RDU.

249 The Defendant submits that it had sought a full certification from ABSG because the Confidential Bundle that it submitted to ABSG for the design review contained, *inter alia*, drawings, calculations, data and adequate information for the hydraulic system, gear sizing, clamping design and bolts. ABSG was at liberty to request for more information if the information provided by the Defendant was insufficient.

250 I find it unbelievable that the Defendant is now claiming that the job scope for the design review as delineated by quotations, emails and invoices flowing between ABSG and the Defendant was meant to (or could simply be enlarged to) include a design review of the mechanical system by virtue of some information being furnished to ABSG on the mechanical and hydraulic system in the Confidential Bundle without the Defendant having to pay for the additional work of reviewing and certifying the mechanical and hydraulic system, which according to Mr Ravi Chandran is very costly to do. Obviously, ABSG would only do what it was asked and paid to do as evidenced by the quotations and invoices and no more. I do not think the Defendant could have any genuine misunderstanding on this. Having used the services of ABSG for some ten years previously, I believe that the Defendant would have known that the mechanical and hydraulic system review would be a separate item and would be very costly to do. I do not think that the Defendant could have been mistaken that ABSG did not in fact perform any mechanical and hydraulic system review for the RDU.

251 The Plaintiff disputes the factual assertion by the Defendant that the material in the Confidential Bundle sent to ABSG was, to begin with, adequate for full certification. I accept the Plaintiff's position that the Confidential Bundle omitted entire sets of relevant detail about the RDU and its principal parts. The Plaintiff highlights in its submission the following non-exhaustive list of serious omissions:

(a) No calculations were done with respect to the bearings. The bearings on the main shaft are particularly relevant as they are at the start of the load path into the machine structure, and are thus particularly heavily loaded and also heavily exposed to environmental shock loads.

(b) No calculations were done with respect of the strut bolts, shaft end plate bolts, bearing housing bolts, Sauer Danfoss gearbox mounting bolts, or indeed any bolts at all except for those bolts connecting the base of the tower to the machine base. That is a total of 20 tower base bolts only out of several hundred other bolts in the RDU holding together heavily loaded structural and mechanical parts, where bolting performance is absolutely critical (as opposed to minor fittings like the control panel face and suchlike where the loads are insignificant).

(c) No gear calculations were done corresponding to any known modern standard. As Mr Drago pointed out, the "Hertzian Contact" and "Barth Equation" used by the Defendant had been out of date for about one whole century, and were developed before widespread use of the various forms of surface hardening in gears. Mr Drago used a modern standard (the American Gear Manufacturer's Association codes), and even the Defendant's expert Mr Hooper used the broadly equivalent and modern ISO 6336 standard in his calculations.

(d) No calculations were performed in respect of the sub-frame assembly at the start of the load path, and which also carries the motors and gearboxes which drive the unit. Omitting this crucial assembly from analysis is hardly "adequate" to fully certify the unit.

(e) No calculations were performed with respect to the brakes and their capacity, for example, to take the extreme loads generated in the course of stopping a rotating 9.2 m diameter reel of 300 tons, particularly in an emergency.

252 If the Defendant had, as it appears to be suggesting, originally intended and requested for the hydraulic system to be mechanically certified because adequate information on the design of the hydraulic system were initially provided to ABSG, then it is surprising that the Defendant did not include the following in the Confidential Bundle sent to ABSG:

(a) No calculations and specifications were provided for any hydraulic components except for a very simplistic calculation of the horsepower and flow required to achieve the required torque and speed. None of the required ratings of valves, bypasses, electrohydraulic components, etc. were present, and there was also no calculation at all in respect of the critical electro-hydraulic servo system which was supposed to maintain constant torque as per the machine's contractually required constant torque/constant tension capability (and whose inability to do so with any degree of accuracy became clear during trial).

(b) No calculations were provided about the adequacy (or otherwise) of the hydraulic pump which failed during the second FAT testing of the unit.

253 The paucity of the design information on the hydraulic system sent to ABSG shows that the Defendant never intended to ask and never asked ABSG to perform a full mechanical review of the RDU's hydraulic system as part of its so called "full certification" for the RDU. The Defendant had therefore misled the Plaintiff into believing that it had obtained full certification from ABSG, when it knew as a fact that ABSG had not provided full certification for the RDU. The Plaintiff obviously would not have taken delivery of the RDU had it known that there was in reality no full but only partial certification by ABSG.

Applicability of clause 25 in the Sale and Purchase Agreement

254 As I have found (a) that the ABSG certification was fraudulently or dishonestly obtained by the Defendant in the manner set out in [223] to [246] in order to meet the contractual requirements; and (b) that the Defendant had, for the reasons stated at [247] to [253], also dishonestly misrepresented to the Plaintiff that full ABSG certification had been obtained, cl 25 of the Sale and Purchase Agreement is not applicable as a limitation clause favouring the Defendant by limiting the scope of

damages payable by the Defendant for its breach of contract.

255 I do not need to discuss why cl 25 is not available to the Defendant as the parties have agreed that in the event of fraud being found (which I have), cl 25 purportedly excluding damages payable by the Defendant for "*any consequential or indirect losses (whether or not foreseeable by either Party ..) including but not limited to loss of profits, products, and business interruption or economic losses arising out of or as a result of the performance of the work and regardless of the cause, or reason for the said loss or damage and regardless of whether the same may arise as a result of the negligence of the other*" would not be applicable. It is quite clear that the parties could not have intended cl 25 to cover fraudulent acts or omissions of the Defendant or its employees, servants or agents. In my view, cl 25 cannot be construed to limit the extent of the damages payable by the Defendant where fraud or dishonesty is involved.

Lifting of confidentiality for the Confidential Bundle

256 The Plaintiff has applied to the court to lift the confidentiality restriction in relation to the design drawings and calculations that were submitted to ABSG which were referred to in court as the Confidential Bundle.

257 The Plaintiff says that the Defendant is not seeking to protect confidentiality but rather its reputation. I do not think that lifting the protection of confidentiality over the proprietary design documents will achieve more than what the Plaintiff intends to have that this judgment does not already do in relation to the Defendant's reputation.

258 In my opinion, even if the design is flawed the Defendant's design drawings and accompanying calculations are nevertheless proprietary in nature. The Defendant can readily redo the calculations found in the Confidential Bundle, modify their design and production drawings and develop an improved RDU design for its future use.

259 As such, I will allow the court order covering the confidentiality of the Confidential Bundle to stay.

Punitive Damages

260 The Plaintiff seeks punitive damages and relies on *Whiten v Pilot Insurance Company* [2002] 1 RCS 595 ("*Whiten*") where the Supreme Court of Canada affirmed (at p 617) the general principles adopted in Canadian case law, that punitive damages would generally be awarded against a defendant "*in exceptional cases for 'malicious, oppressive and high-handed' misconduct that 'offends the court's sense of decency'*". It would involve "*misconduct that represents a marked departure from ordinary standards of decent behaviour*". The Supreme Court of Canada considered how other common law jurisdictions (*ie*, England, Australia, New Zealand, Ireland, United States) had addressed the various issues concerning punitive damages. I find the following passage of the judgment at p 645 to contain a very helpful summary:

> (1) Punitive damages are very much the exception than the rule, (2) imposed only if there has been high-handed, malicious, arbitrary or highly reprehensible misconduct that departs to a marked degree from ordinary standards of decent behaviour. (3) Where they are awarded, punitive damages should be assessed in an amount reasonably proportionate to such factors as the harm caused, the degree of the misconduct, the relative vulnerability of the plaintiff and any advantage or profit gained by the defendant, (4) having regard to any other fines or penalties suffered by the defendant for the misconduct in question. (5) Punitive damages are generally

given only where the misconduct would otherwise be unpunished or where other penalties are or are likely to be inadequate to achieve the objectives of retribution, deterrence and denunciation. (6) Their purpose is not to compensate the plaintiff, but (7) to give a defendant his or her just desert (retribution), to deter the defendant and others from similar misconduct in the future (deterrence), and to mark the community's collective condemnation (denunciation) of what has happened. (8) Punitive damages are awarded *only* where compensatory damages, which to some extent are punitive, are insufficient to accomplish these objectives, and (9) they are given in an amount that is no greater than necessary to rationally accomplish their purpose. (10) While normally the state would be the recipient of any fine or penalty for misconduct, the plaintiff will keep punitive damages as a "windfall" in addition to compensatory damages. (11) Judges and juries in our system have usually found that moderate awards of punitive damages, which inevitably carry a stigma in the broader community, are generally sufficient.

261 The fundamental purpose of punitive damages as appears at point (7) above is therefore to "*give a defendant his or her just desert (retribution), to deter the defendant and others from similar conduct (deterrence), and to mark the community's collective condemnation (denunciation) of what has happened*".

262 On whether punitive damages should be available to punish defendants who breach their contracts in a high-handed and outrageous fashion, the Ontario High Court in *Brown v Waterloo Regional Board of Commissioners of Police* (1981) 136 DLR (3d) 49 said:

> 52… **Punitive damage awards should be part of the judicial arsenal in contract cases in the same way as they are in tort cases. I see no sound reason to differentiate between them**. Canadian courts, unlike English courts, have retained their broad power to award punitive damages in tort cases. Thus, if a high-handed breach of contract also happens to amount to tortious conduct, punitive damages would be awardable pursuant to tort theory. It is said that if this conduct is purely a breach of contract and not tortious then no punitive damages can be awarded, despite the callousness of the conduct. That makes no sense. **It is wrong to treat one contract breach different from another merely because one violates tort principles while the other does not. … By allowing punitive damages for contract breach … those who plan to breach contracts in a callous fashion will think twice.**
>
> 53 Consequently, I conclude that **it is not beyond the power of this court to award punitive damages in those rare situations where a contract has been breached in a high-handed, shocking and arrogant fashion so as to demand condemnation by the court as a deterrent**.
>
> [Emphasis added in bold]

263 I agree with the Canadian authorities cited by the Plaintiff and the position canvassed by Phang and Lee in *Restitutionary and Exemplary Damages Revisited* (2003) 19 CLJ 1 that the courts should ordinarily have a residual power to award exemplary damages for breach of contract in "*truly exceptional situations when the defendant's conduct has been outrageous*" (at p 26). Ralph Cunnington also supports this position where he wrote in *Should punitive damages be part of the judicial arsenal in contract cases?* (2006) 26 Legal Studies 369 (at p 380) that:

> Contracts are frequently broken in circumstances that evoke outrage and require deterrence. All too often compensatory damages are inadequate for this purpose. Surely there is a strong argument that, in such cases, punitive damages should be awarded to effect deterrence.

264 The Singapore Court of Appeal recently recognised the possibility of recovering exemplary damages in the context of breaches of contract. In *MFM Restaurants Pte Ltd v Fish & Co Restaurants Pte Ltd* [2011] 1 SLR 150 ("*MFM*"), the Court of Appeal noted that "*there is an arguable case, in principle, for the award of punitive damages in contract law*" though the question is "*still an open one*" as the case law is itself inconclusive (see *MFM* at [53]). Unless there is a Court of Appeal decision ruling out the availability of punitive damages for breach of contract, I am inclined to hold that the court has the power in an exceptional case to award punitive damages in the context of a breach of contract, when the defendant's conduct in breaching the contract has been so highly reprehensible, shocking or outrageous that the court finds it necessary to condemn and deter such conduct by imposing punitive damages.

265 Has the threshold been crossed on the facts of this case such that punitive damages should be awarded? The Defendant accepts that there were some deficiencies in the RDU. However, it submits that at the most, they constitute minor or partial non-performance of its contractual obligations with no recklessness or fraud on its part. I do not agree.

266 The Defendant had asked a junior engineer to copy an RDU design which at that time was not yet tested, built or certified by any third party agency. The junior engineer then modified it to fulfil the design requirements of the Plaintiff's RDU which were different. No comprehensive design file or calculations were produced save for what was in the Confidential Bundle which was inadequate. I infer that comprehensive engineering calculations were not performed to ensure that the design of the RDU would meet the specifications and the user requirements as made known to the Defendant.

267 The Defendant fraudulently misled the third party certification agency ABSG by submitting a model of the two towers of the RDU which misrepresented what the Defendant intended to build. Critical joints in the RDU structure were modelled in the FEA analysis as having full freedom of movement in all three axes as if they were fully rotatable ball joints when in fact they were strong inflexible and fully fixed joints securely fastened with four bolts at each joint. Worse, this was after the Defendant knew that when properly modelled, the joints and the structure would not pass the unity checks for ABSG certification. The Defendant understood that fatigue and wind load had to be included in the design from an engineering perspective because of the environment that it was operating in. But as the false model of the RDU submitted for certification was already close to the border line for passing the unity checks, the Defendant chose to omit fatigue and wind load for the design and certification so that its false model would not fail the unity checks. In other words, the Defendant was aware of the RDU's structural failings and falsified structural submissions to the relevant safety homologation body to make the design pass. In doing this, the Defendant had acted dishonestly and fraudulently.

268 The Defendant knew it had only obtained partial certification from ABSG but it misled the Plaintiff into believing at the time of delivery of the RDU that the required full certification had been obtained when it was not true in fact.

269 The Defendant purchased important components – gears, bearings and bolts – and installed them without having them properly checked and tested at its work site. It installed defective and substandard parts without checking. Construction drawings specified SKF bearings but cheaper bearings made in China were purchased and installed in the RDU instead. It was the Defendant's practice always to purchase bearings from China apparently in disregard of what might have been specified in the design, engineering and production drawings. This practice was sanctioned by the Defendant's management. The engineering/production drawings provided that bolts of grade 10.9 were to be used in the main base and the sliding base but eventually a lower grade of bolts *ie*, grade 8.8 was used in the RDU.

270 The fraudulent inputs to the STAAD.Pro modelling just to obtain certification by ABSG, the deliberate deviations during construction of the RDU for components specified in the design drawings and the installation of components without first checking on their quality show a blatant disregard by the Defendant of its contractual obligation to design and construct an RDU for the Plaintiff that was satisfactory in quality and fit for its purpose.

271 The workmanship and manufacturing quality was so poor that the Defendant is unable to respond to the Plaintiff's criticism of them in its submission.

272 Given the above, I agree with the Plaintiff's submission that the Defendant's overall conduct is sufficiently outrageous and reprehensible to call for an imposition of punitive damages. I find that the threshold to award punitive damages has been crossed. This is an exceptional case. What was done was blatantly irresponsible designing, engineering and manufacturing.

Conclusion

273 As I have ordered this trial to be bifurcated with assessment of damages to be conducted later, this judgment deals only with the question of liability.

274 The Plaintiff is entitled to damages on the basis that the Defendant had breached its contract by not delivering an RDU that was of merchantable quality and fit for its purpose to the Plaintiff.

275 For the reasons I have stated, I further award the Plaintiff punitive damages to be assessed.

276 Finally, I would like to place on record my appreciation to counsel for preparing detailed and comprehensive e-submissions that are fully hyperlinked to all the references. The hyperlinking made it so much more convenient for me. I would also like to specially thank Mr Jonathan Seow from the Information System Department of Rajah & Tann for assisting the court with setting up the "document codes and paths" to open a page in PDF Reader through a short cut key that executes on a hyperlink in a Word document for the references in the closing submissions of both parties, which I have requested to be hyperlinked to the relevant page in the affidavits, notes of evidence, bundles of documents, exhibits and bundles of authorities for ease of perusal.

277 I am presently arranging for a standard Word document template to be developed so that e-submissions with full hyperlinking can be prepared by the parties before me in future using the court's template, and there will then be no difficulty with the functioning of the hyperlinking facility when the submissions are received from the parties and opened by the court subsequently.

Glossary of Witnesses - Appendix 1

7 Expert Witnesses

Plaintiff's expert witnesses

1. Mr Darren Moore ("Mr Moore"), the Principal Engineer of BPP-Technical Services Limited, possesses engineering expertise and working experience in the international offshore oil and gas industry. He provided the court with a comprehensive technical appraisal of the design and construction of the RDU.

2. Dr William Eccles ("Dr Eccles"), the Engineering Specialist of Bolt Science Limited, specialises in the design and analysis of bolted joints, including the investigations of joint failures and the determination of their causes. He has acquired working experience in, among others, the marine, plant and machine tool industries. Dr Eccles explained in detail the findings of his investigations into the structural integrity of the RDU's various bolted joints.

3. Mr Shreenaath Natarajan ("Mr Natarajan"), the Technical Director of 2H Offshore Engineering Ltd, has acquired over 12 years' worth of specialist experience in the design, analysis and integrity monitoring of riser and conductor systems. He is a Naval Architect by training. He assisted the court by providing a design review of the entire reel handler unit and its structural components.

4. Mr Paul Goh Aik Boon ("Mr Paul Goh"), the Inspection Supervisor in K2 Specialist Services, has undertaken numerous supervisory roles in a wide array of projects on behalf of K2 Specialist Services. He provided his findings from the visual and magnetic particle inspections of the four spur pinions in the RDU.

5. Mr Raymond J Drago ("Mr Drago"), the Chief Engineer, Drive Systems Technology Inc, is a renowned authority in the field of gear technology and specialises in gear design, analysis and testing of all forms of gear systems. He has acquired significant working experience and specialist knowledge and has been lauded for his educational efforts and contributions to the field. Mr Drago explained to the court the resultant findings of his very detailed technical appraisal of the gear systems of the RDU.

Defendant's expert witnesses

6. Mr Alan Gregory Hooper ("Mr Hooper"), the Director and President of Promor Pte Ltd, possesses significant experience in the drilling, construction and production fields of the offshore oil and gas industry. He provided a technical appraisal of the RDU, as well as an assessment of the probable causes of its failure.

7. Mr Huang Hai Tao ("Mr Huang"), the Managing Director of Hyin Engineering Pte Ltd, has amassed over 24 years of experience in undertaking structural analyses in the context of oil and gas fields of the marine and offshore industry. He is a Naval Architect by training. He provided to the court his findings based on a Finite Element Analysis of the RDU.

17 Factual Witnesses

Plaintiff's factual witnesses

1. Mr James Rodriguez de Castro ("Mr James Castro"), a Director of the Plaintiff, set out in his evidence the contractual relationship entered into between the Plaintiff and the Defendant in respect of the purchase of the RDU. He explained the Plaintiff's basis for concluding that the RDU was not safe to operate and the Plaintiff's consequent losses suffered.

2. Mr Eilert Halvorsen ("Mr Halvorsen"), a Director of Trident Australasia, provided a contextual understanding of the commercial negotiations between Trident and the parties leading up to the rental of the RDU, and the subsequent failure of the RDU during the offshore umbilical installation operation in the Bass Straits.

3. Mr Tamas King ("Mr Tamas King"), the ROV Project Engineer of Oceaneering International, provided to the court details of the number of projects per year which required lay equipment in the Asian and Middle Eastern regions, and the corresponding number of RDUs ordinarily required for these projects within the region.

Defendant's factual witnesses

4. Dr Yang Ting ("Dr Yang Ting"), the Project Manager of the Defendant, set out in detail the parties' discussions on the construction, certification and testing of the RDU. He further set out his findings on the damage sustained to the RDU as well as his analysis of the conclusions of the Plaintiff's expert witnesses.

5. Ms Yang Mee Lan ("Ms Mary Yang"), the Senior Sales Manager of the Defendant, explained the Defendant's rates, earnings and business opportunities in relation to third party rental of its RDUs.

6. Mr Loh Lan Chiung ("Mr Loh"), the General Manager (Engineering and Operations) of the Defendant, sought to explain that the RDU did not suffer from any design and/or manufacturing defects. In particular, he asserted that the RDU was constructed in accordance with its drawings. He provided his basic technical analysis of the RDU and its respective components.

7. Mr Jaster Suet Zheng Mun ("Mr Jaster Suet"), a former Associate Service Engineer of the Defendant, provided a narrative of his experience on board the *Maersk Responder* in performing a service call for the RDU on behalf of the Defendant. He also briefly provided his observations of the RDU when on board the vessel.

8. Mr Steven Gan Kwok Chang ("Mr Steven Gan"), the Assistant Manager-After Sales of the Defendant, set out his role in the RDU's certification and testing processes as well as the preparation of the project schedule for the construction of the RDU.

9. Mr Philip Chua See Kwang ("Mr Philip Chua"), the Senior Technical Manager of the Defendant, explained the working processes of the Defendant's operations department in the construction of the RDU.

10. Dr Liu Li ("Dr Liu Li") was a freelance structural engineer at the material time. The Defendant subcontracted the STAAD.Pro analysis to Dr Liu. Dr Liu Li explained to the court how he performed the STAAD.Pro analysis for the RDU structure.

11. Mr Lei Chengyi ("Mr Lei"), a former Design Engineer of the Defendant, provided details of his communications with Dr Liu Li in relation to the STAAD.Pro analysis.

12. Ms Renuka Devi ("Ms Renuka Devi"), the Lead Structural Engineer of ABSG Consulting Inc, gave evidence in relation to her structural design review of the wire spooler towers. She certified the structural design of the RDU based on the instructions and STAAD.Pro analysis provided by the Defendant.

13. Mr Ravi Chandran ("Mr Ravi Chandran"), the Lead Electrical Engineer of ABSG Consulting Inc, gave evidence in relation to the electrical design review of the RDU.

14. Mr Derek Chong ("Mr Chong") was the then ABSG surveyor with ABSG Consulting Inc who signed off on the ABSG Inspection Certificates certifying that the materials used for the fabrication of the mechanical parts such as the gears and the pinion shafts were satisfactory.

15. Mr Steven Seow ("Mr Seow"), a former Sales Director of the Defendant, gave evidence regarding the market for the sale and rental of RDUs. Mr Seow also gave evidence about expanding his business to own a fleet of RDUs.

16. Ms Tan Sin Liu ("Ms Tan Sin Liu"), a Mechanical Engineer for the Defendant, was then the engineer who was in charge of the design of the Plaintiff's RDU.

17. Mr Derek Chua ("Mr Derek Chua"), the Executive Director of the Defendant, gave evidence on what he perceived to be the likely cause of the failure of the RDU on 20 May 2009.

[note: 1] 300 tons × Sin 10° = 52.1MT.

[note: 2] The submissions were agreed to be filed in the following sequence: Plaintiff's Closing Submissions to be followed by the Defendant's Closing Submissions and thereafter, the Plaintiff is to file its Reply Submissions.