Report of Thorough Examination of Loader Crane

ATLAS
CRANES UK LTD

Service Division - Tel. 08444 996688

AVAILABLE & COST EFFECTIVE – WWW.ATLASGMBH.COM

LOADER CRANE SERVICE INSPECTION/THOROUGH EXAMINATION REPORT 105391

CUSTOMER	Inverlussa(crane sales)		64089	MK Job No.	731754
DEPOT	oban	CR.HRS		MILEAGE	
MAKE	atlas	MODEL	170vcs a12	SERIAL No.	0170p00535
VEHICLE	m v carol anne	MODEL		REG No.	
AUX EQUIP		MODEL		SERIAL No.	
ROTATOR-IF APL		MODEL		SERIAL No.	

Item Code	COMPONENT	1.2.3.4.NA
1	Subframe	1
2	Hold down bolts	1
3	Crane base	1
4	Pendulum Beam	NA
5	Hydraulic Tank	NA
6	Boom Stowage	1
7	PTO Switch	NA
8	Handbrake Interlock	NA
9	Hydraulic Pump	NA
10	Pressure Feed Hose	1
11	Suction Hose	1
12	Hand/Auto Throttle	NA
13	**OUTRIGGER LEGS**	NA
14	Beams	NA
15	Rollers	NA
16	Cam locks and R Clips	NA
17	Secondary leg locks	NA
18	Leg Stops	NA
19	Outrigger Hoses	NA
20	Stabilizer Legs	NA
21	Mechanical Leg	NA
22	Mechanical Leg Pin/R Clip	NA
23	Feet	NA
24	Stabilizer Hoses	NA
25	Stabilizer Rams	NA
26	Stabilizer Ram Pins	NA
27	Hydraulic Folding Legs	NA
28	Pressure Sensors	NA
29	Beam Position Sensors	NA
	VALVE BLOCK	
30	Dump Valve	1
31	Dump Valve Solenoid	1
32	Main Relief Valve	1
33	Main Relief Valve Seals	1
34	Outrigger Spool	NA
35	Slew Spool	1
36	Main Lift Spool	1
37	Jib Spool	1
38	Extension Spool	1
39	Auxiliary Spool	NA
40	Auxiliary Valve Seals	NA
41	Overload Protection	1
42	Overload Protection Seals	1
43	Valve Bank Pipes	NA
44	Swivel Connections	1
45	Valve block leaks	1
46	Control Levers	NA
47	Control Lever Decals	NA
48	Cross Rods/Cables	NA
49	Pressure Gauge/Fittings	NA

Item Code	COMPONENT	1.2.3.4.NA
	TOP SEAT	
50	Top Seat	NA
51	Fabrication	NA
52	Access	NA
53	Emergency Stop	NA
	STANDUP CONTROLS (T1)	
54	T1 Platform fabrication	NA
55	Controls	NA
56	Access	NA
	COLUMN	
57	Hoses	1
58	Bearings (Neck/Foot)	1
59	Slew Cylinders	1
60	Slew Position Sensors	NA
61	Fabrication	1
62	Top Pin/Bushes	1
	MAIN LIFT	
63	Cylinder	1
64	L.H.V / H.F.V	1
65	Pins/Bushes	1
66	Cylinder Power Link	1
	BOOM	
67	Fabrication	1
	JIB CYLINDER	
68	Cylinder	1
69	L.H.V /H.F.V	1
70	Pins/Bushes	1
71	Jib to Boom Pin/Bushes	1
	JIB KICKING LINK	
72	Jib Kicking Link	1
73	Pins/Bushes	1
	JIB	
74	Fabrication	1
75	Hoses/Pipes	1
76	Rotary Banjo/L.H.V	1
	JIB EXTENSIONS	
77	Fabrication	1
78	Cylinders	1
79	Cylinder Guides	1
80	Wear Pads	1
81	Pins/Bushes	1
	AUXILIARY EQUIPMENT	
82	Hose Guides and Carriers	NA
83	Quick Release Couplings	NA
84	(B30) Head Fabrication	NA
85	(B30) Head Pins/Clip	NA
86	Hook	1
87	Shackle	1
88	Hook Safety Catch	1
89	Winch	NA

Item Code	COMPONENT	1.2.3.4.NA
90	Winch Rope	1
91	Rotator	NA
92	Rotator link Pin	NA
93	Rotator Bolts	NA
	ATTACHMENT/GRAB	
94	Hoses	NA
95	Rams	NA
96	Pins/Bushes	NA
97	Fabrication	NA
98	Cross Rods	NA
99	Rubbers	NA
100	Cutting Edge	NA
101	Clips	NA
102	General Wear	NA
	INFORMATION PLATES	
103	Load Plates	3
104	CE Plates	1
105	Serial Plate	1
	ELECTRICS	
106	Wiring	NA
107	Control Box	NA
108	ON/OFF Button	NA
109	Emergency Stop	NA
110	O.L.P Lights	NA
111	O.L.P System	NA
112	General Condition	NA
113	Leg Lights	NA
	SAFE STOWAGE SYSTEM	
114	Visual Alarm	NA
115	Audible Alarm	NA
116	Boom Above Horizontal	NA
117	Boom outside Body	NA
118	Legs Not Stowed	NA
119	Radio Remote Stowage	NA
	RADIO REMOTE	
120	Receiver	1
121	Transmitter	1
122	Battery Charger	1
123	Battery Condition	1
	OTHER SAFETY EQUIPMENT	
124		NA
	AUTOLUBE SYSTEM	
125	General Condition	NA
126	Operation	NA
127	Container Contents	NA
128	Pipework	NA
	PRESSURE SETTING BAR	
	VEHICLE RPM	

[redacted]

Item Code	DEFECT	STATUS D: Damage W: Wear & Tear	ACTION REQUIRED	DEADLINE
1	103 no load plates		phoned [redacted], he will send to customer	
2				
3				
4				
5				
6				
7				
8				
9				
10				
11				
12				
13				
14				
15				
16				
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				

Date	Start	Time on site	Time off site	Time Home	Mileage
	10 30	12 30	14 30	18 30	212

COMMENTS

Set olp as per sales booklet as [redacted] over phone lunch 12 00 1230

FAXED TO: FAX No.

TIME DATE

CONFIRMED

PARTS USED FOR SERVICE

ITEM	PART No.	QTY
1		
2		
3		
4		
5		
6		

☐ If this box is ticked,

it is in our opinion the crane is unsafe to use until the above repairs marked 'immediate' are effected

- GENERAL SERVICE ☐
- CRANE GREASED ☐
- HYDRAULIC FILTER CHECKED ☐
- HYDRAULIC FILTER RENEWED ☐
- HYDRAULIC OIL CHANGED/TOPPED UP ☐
- AUTOLUBE TOPPED UP ☐

CUSTOMER

NAME [redacted]
POSITION owner
DATE 23 3 15
CUSTOMER SIGNATURE

[redacted]

CUSTOMERS SIGN HERE

ENGINEER

NAME [redacted]
I hereby acknowledge I have/am proceeding work using the applicable Risk Assessment & Method Statement
DATE 23 3 15
ENGINEERS SIGNATURE

[redacted]

ATLAS SIGN HERE

How did you rate our service today? ☐ Excellent ☐ Good ☐ OK ☐ Poor ☐ Very poor

[redacted]

ATLAS CRANES UK LTD

Service Division - Tel. 08444 996688

AVAILABLE & COST EFFECTIVE – WWW.ATLASGMBH.COM

REPORT OF THOROUGH EXAMINATION OF LOADER CRANE 302288
Lifting operation and lifting equipment regulations 1998 (Regulation 9)

Date of Last Thorough Examination:	23 3 15	CALL No.	64089
Latest date by which the next thorough examination must be carried out:	23 3 16	Record Identification: No.	
Employer (and or equipment owner) Address:	Inverlussa Oban	Address at which examination was made.	Same
Reg No./Chassis No.	m v carol anne	Equipment Make/Model	atlas 170vcs a12
Loader Mounting details	marine crana	Serial No.	0170 p00535
		Year of Manufacture	2015

Safe working load - to cover range of SWL's & configurations & particulars of any test i.e. 10% for annual/ 25% in all other cases

	Load Radius (Metres)	Test Load (Kgs)	Safe Loading (Kilogrammes)	Overload %
1	10.8	2000	1500	25
2	8.45		2000	
3				

NOTE: IF THE DATE OF LAST THOROUGH EXAMINATION IS UNKNOWN A 25% OVERLOAD MUST BE APPLIED

Nature of Examination	Please tick as appropriate				
			12 Monthly examination 9(3)(a)(ii)	☐	+ 10%
Second hand/ Non CE 9(1)	☐	+ 25%	4 Yearly	☐	+ 25%
Ist examination following installation 9(2)	☑	+ 25%	In accordance with examination scheme 9(3)(a)(iii)	☐	+ 10%
6 monthly examination 9(3)(a)(i)	☐	+ 25%	Following exceptional circumstances 9(3)(a)(iv)	☐	+ 25%

Identification of any part found to have a defect which is or could become a danger to persons.
The particulars of any repair, renewal or alteration required to remedy the defect either immediately or within a specified time.

Other defects and remedies

Observation and condition of attachment (state, make, model and serial no.)

CUSTOMER

NAME [redacted]

POSITION owner

DATE 23 3 15

CUSTOMER SIGNATURE

[redacted]

CUSTOMERS SIGN HERE

ENGINEER

NAME [redacted]

I hereby acknowledge I have/am proceeding work using the applicable Risk Assessment & Method Statement

DATE 23 3 15

ENGINEERS SIGNATURE

[redacted]

ATLAS SIGN HERE

How did you rate our service today? ☐ Excellent ☐ Good ☐ OK ☐ Poor ☐ Very poor

[redacted]

Annex B

Report from Caparo Testing Technologies on the Lock Nuts

UKAS TESTING
No. 0561

Unit 11, Planetary Road Industrial Estate
Willenhall, Wolverhampton
West Midlands
WV13 3XA

Tel: 01902 305200
Fax: 01902 303200
Email: willenhall@caparotesting.com

caparo
TESTING TECHNOLOGIES

CTT: Willenhall
A Division of Caparo Engineering Limited

LABORATORY INVESTIGATION REPORT

Date Received/Tested 21st July 2015 (Page 1 of 9)
Customer Marine Accident Investigation Enquiry from [redacted]
Address Spring Place, 105 Commercial Road, Southampton, SO15 1GH
Certificate No. PFJ168 Order No. N/A Customers Ref. No. N/A
Material Specification ISO 898-2 GRADE 8 Dimensions M24 Nyloc Nut

M24 x 2.0 pitch fine threaded nuts and a location plate were submitted by yourselves for a detailed investigation.

CHEMICAL ANALYSIS (Analytical Method Used: - OES) – Location Plate

C%	Si%	Mn%	P%	S%	Cr%	Mo%	Ni%	Al%	Co%
0.130	0.285	0.549	0.011	0.022	0.097	0.009	0.123	0.0104	0.012
Cu%	**Nb%**	**Ti%**	**V%**	**W%**	**Pb%**	**Sn%**	**Ca%**	**B%**	
0.316	0.002	<0.0010	0.001	0.003	<0.003	0.016	0.0009	<0.0002	

CHEMICAL ANALYSIS (Analytical Method Used: - OES) – M24 x 2.0 Nut

C%	Si%	Mn%	P%	S%	Cr%	Mo%	Ni%	Al%	Co%
0.109	0.088	0.426	0.013	0.031	0.003	0.026	0.0279	0.013	0.039
Cu%	**Nb%**	**Ti%**	**V%**	**W%**	**Pb%**	**Sn%**	**Ca%**	**B%**	
0.039	0.002	<0.0010	0.001	0.002	<0.003	0.004	0.0016	<0.0002	

PROOF LOAD TEST

Result (Kn)	
Requirements for grade 8 Nut style 1 - 395.5kn	Thread stripped on nut at a force of 301.21Kn – failed to meet grade 8 requirements

VICKERS HARDNESS TEST (ASTM E384: 2011E1) - Nut

HV – 10kg
225

BRINELL HARDNESS TEST (ASTM E 18: 2014A) - Plate

HRB
85

UKAS TESTING
No. 0561

Unit 11, Planetary Road Industrial Estate
Willenhall, Wolverhampton
West Midlands
WV13 3XA

Tel: 01902 305200
Fax: 01902 303200
Email: willenhall@caparotesting.com

caparo
TESTING TECHNOLOGIES

CTT: Willenhall
A Division of Caparo Engineering Limited

LABORATORY INVESTIGATION REPORT

Date Received/Tested 21st July 2015 (Page 2 of 9)
Customer Marine Accident Investigation Enquiry from [redacted]
Address Spring Place, 105 Commercial Road, Southampton, SO15 1GH
Certificate No. PFJ168 Order No. N/A Customers Ref. No. N/A
Material Specification ISO 898-2 GRADE 8 Dimensions M24 Nyloc Nut

VICKERS HARDNESS SURVEY (BS EN ISO 6507-1: 2005)

TEST	50gm
1	209.7
2	203.6
3	203
4	199.4
5	209.3

Note: Location of hardness test taken 0.5m, as close as possible to the nominal major diameter of the nut thread. Ref: ISO 898-2 9.2.33

Coating Thickness - (by micro examination)

A micro section was cut from the sample. The micro sections were encased in a resin base and then polished to a 1 micron surface finish.

Results (mm)	
EXTERNAL SURFACE	UPTO 0.0488
THREADED PROFILE	Nil

Photos 1 and 2

DECARBURISATION INSPECTION (ASTM F2328-14)

A longitudinal section was taken from the sample and polished, then etched in 2% Nital.

Examination of the section showed no evidence of decarburization.

No decarburization was detected, due to material had not been heat treated.

Photo 6

UKAS TESTING
No. 0561

Unit 11, Planetary Road Industrial Estate
Willenhall, Wolverhampton
West Midlands
WV13 3XA

Tel: 01902 305200
Fax: 01902 303200
Email: willenhall@caparotesting.com

TESTING TECHNOLOGIES

CTT: Willenhall
A Division of Caparo Engineering Limited

LABORATORY INVESTIGATION REPORT

Date Received/Tested 21st July 2015

Customer Marine Accident Investigation Enquiry from [redacted]

Address Spring Place, 105 Commercial Road, Southampton, SO15 1GH

Certificate No. PFJ168 Order No. N/A Customers Ref. No. N/A

Material Specification ISO 898-2 GRADE 8 Dimensions M24 Nyloc Nut

MICROSTRUCTURE

Micrographic Examination (ASTM: E 3: 2001 (R2007) E1)

A longitudinal section was taken from three samples and polished, then etched in 2% Nital.

Photo's 4, 5 and 6 – Non heat treated structure was evident that of a ferrite/pearlite structure.

MACROSTRUCTURE (ASTM: E340-01 (2012))

A longitudinal section was taken from the sample and subjected to a Macro etch in 50% HCL to boiling point for a time of approximately 8 minutes.

Photo 3

DETERMINATION OF INCLUSION CONTENT (ASTM: E 45: 2011 – Method A)

A longitudinal section was taken from the sample and polished.

Examination of the section revealed an inclusion content, as follows: -

TYPE A		TYPE B		TYPE C		TYPE D	
Thin	Heavy	Thin	Heavy	Thin	Heavy	Thin	Heavy
0.5	0	1	0	1.5	0	0	1.5

Unit 11, Planetary Road Industrial Estate
Willenhall, Wolverhampton
West Midlands
WV13 3XA

Tel: 01902 305200
Fax: 01902 303200
Email: willenhall@caparotesting.com

UKAS TESTING No. 0561

TESTING TECHNOLOGIES

CTT: Willenhall

A Division of Caparo Engineering Limited

LABORATORY INVESTIGATION REPORT

Date Received/Tested 21st July 2015

Customer Marine Accident Investigation Enquiry from [redacted]

Address Spring Place, 105 Commercial Road, Southampton, SO15 1GH

Certificate No. PFJ168 Order No. N/A Customers Ref. No. N/A

Material Specification ISO 898-2 GRADE 8 Dimensions M24 Nyloc Nut

COMMENTS

M24 Zinc plated nylon lock nuts were submitted for testing/investigation to establish possible cause of thread striping from nuts.
The nuts had failed as part of a fixing kit, for fixing a brick and block type crane to a back of a lorry.

Dimensions of the nut established that the nut was in fact manufactured in accordance with DIN 985. This established a thin nut type (M24 fine thread pitch 2.0) with reduced loadability (Style O).

Loadability of the design of nut would be in accordance with Din 267 Part 4 based on a proof load of 800 N/mm^2 proof load value 307000N.

The marking/identification of the nut did not indicate that the nut was that of reduced loadability, has it was identified as a Grade 8 nut (Style 1) nominal size nut, and not 08 (Style 0) this would indicate that the nut was in fact a thin nut with reduced loadability.

Loadability of a regular style nut Grade 8 would have a proof load strength based on 1030 M/mm_2 for M24 fine thread nut proof load value 395500N.

Quote from BS EN ISO 898-2:2012 – Thin nuts (Style O) have a reduced loadability compared to regular nuts (Style 1) or high nuts (Style 2) and are not designed to provide resistance to thread striping.

This would indicate that the incorrect type of nut was selected/used for the application of being part of the fixing kit supplied with the brick and block type crane.

Several tests – see attached report was carried out to establish what condition / material grade and proof load strength resistance the nut displayed.

Testing revealed that in fact the M24 Nut actually displayed material grade type, that of a 06 type nut.

Load ability of 06 thin type nut in accordance with DIN 267 Part 4 based on a proof stress of $800N/mm_2$ would give a proof load value of 23000N

No. 0561

Unit 11, Planetary Road Industrial Estate
Willenhall, Wolverhampton
West Midlands
WV13 3XA

Tel: 01902 305200
Fax: 01902 303200
Email: willenhall@caparotesting.com

TESTING TECHNOLOGIES

CTT: Willenhall
A Division of Caparo Engineering Limited

LABORATORY INVESTIGATION REPORT

Date Received/Tested 21st July 2015 (Page 5 of 9)
Customer Marine Accident Investigation Enquiry from [redacted]
Address Spring Place, 105 Commercial Road, Southampton, SO15 1GH
Certificate No. PFJ168 Order No. N/A Customers Ref. No. N/A
Material Specification ISO 898-2 GRADE 8 Dimensions M24 Nyloc Nut

06 Thin type nut (style O reduced loadability) would be manufactured from a plain carbon steel, containing small amounts of Mn only, and no further alloy element to assist in the ability to be heat treated for a nut of size such as M24.

Compare that to a grade 8 nut of regular size – would have normally been made from a low alloyed steel, followed by heat treatment operation. This would result in a much stronger nut, with much high proof load strength to be obtained.

Micro evaluation of the nut established that the nut had been supplied in the NON heat treated condition i.e. Displayed a ferrite and pearlite structure and not that of the tempered martensitic structure that would have been evident for a grade 8 nut.

Quote from BS EN ISO 898-2:2012

Section 6 Materials – Nuts with fine pitch thread and property classes 05,6 (with D>M16), 8 {regular nuts (Style 1)}, 10 and 12 shall be quenched and tempered (Heat treated condition).

BS EN ISO 898-2:2012 Table 2

Also identifies the maximum property class of mating bolt, screw or stud for a given nut grade.

This for a grade 10.9 stud (as supplied in the fixing kit) then the correct nut to be used would be a grade 10 Nut of regular size (Style 1)

From BS EN ISO 898-2:2012

<u>Section 5 Design of bolt and nut assemblies</u>

Regular nuts (Style 1) and high nuts (Style 2) shall be material with externally threaded fasteners according to table 2. However, nuts of a higher property class may replace nuts of lower property class.

Unfortunately this was not the case with the fixing kit supplier in that a much lower grade of nut was used.

Grade 10 nut or high grade 12 nut should have been selected for a 10.9 grade stud assembly.

UKAS TESTING
No. 0561

Unit 11, Planetary Road Industrial Estate
Willenhall, Wolverhampton
West Midlands
WV13 3XA

Tel: 01902 305200
Fax: 01902 303200
Email: willenhall@caparotesting.com

TESTING TECHNOLOGIES

CTT: Willenhall
A Division of Caparo Engineering Limited

LABORATORY INVESTIGATION REPORT

Date Received/Tested 21st July 2015

Customer Marine Accident Investigation Enquiry from [redacted]

Address Spring Place, 105 Commercial Road, Southampton, SO15 1GH

Certificate No. PFJ168 Order No. N/A Customers Ref. No. N/A

Material Specification ISO 898-2 GRADE 8 Dimensions M24 Nyloc Nut

Incorrect identification of nut and incorrect material grade of nut was the cause of threads of the nut to fail.

See Photos below

Signed for and on behalf of CTT: Willenhall:

[redacted]

- [redacted] - Chief Chemist
- General Manager
- Operations Manager

Date of Signing: 24 July 2015

DECARBURISATION, MICROSTRUCTURE AND MACROSTRUCTURE ARE CURRENTLY OUTSIDE THE SCOPE OF OUR UKAS ACCREDITATION

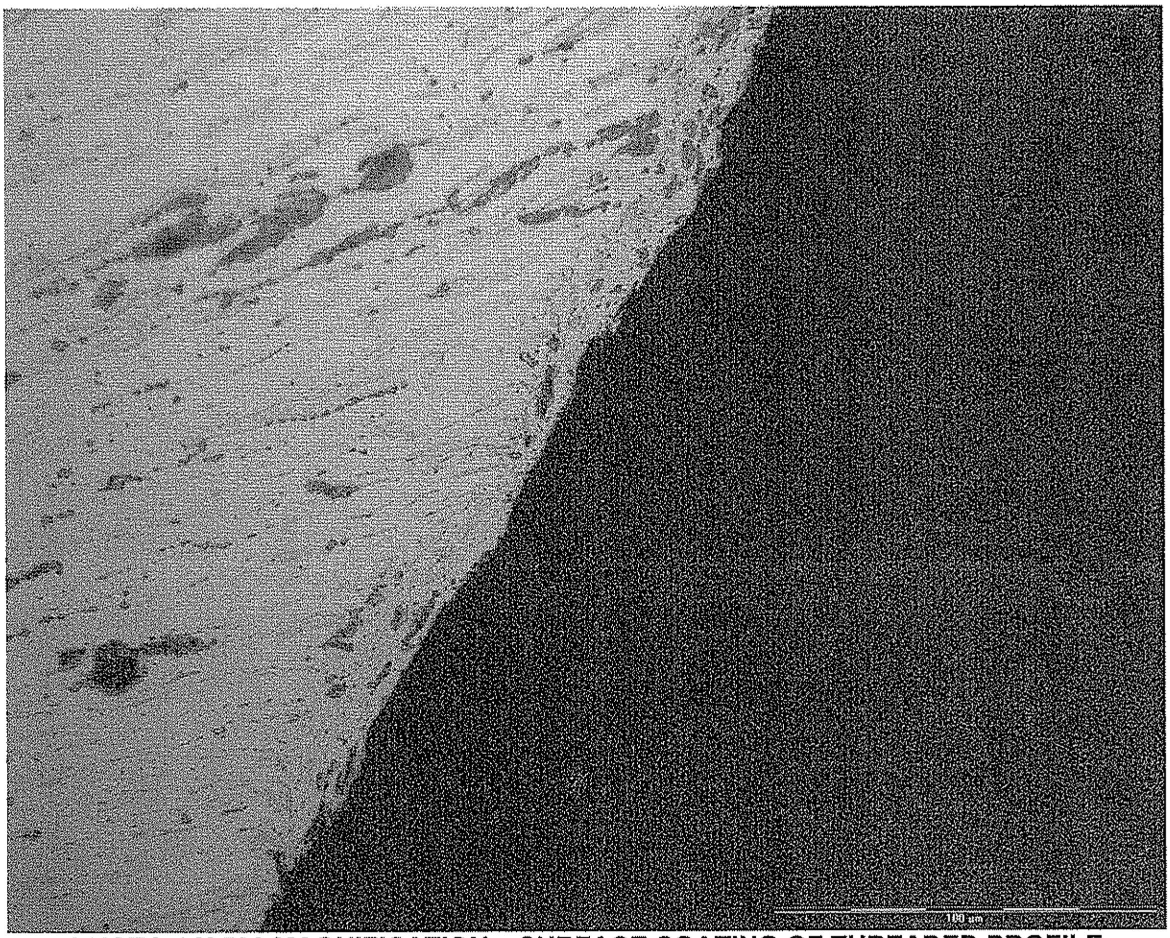

PHOTO 1 X 500 MAGNIFICATION – SURFACE COATING OF THREADED PROFILE

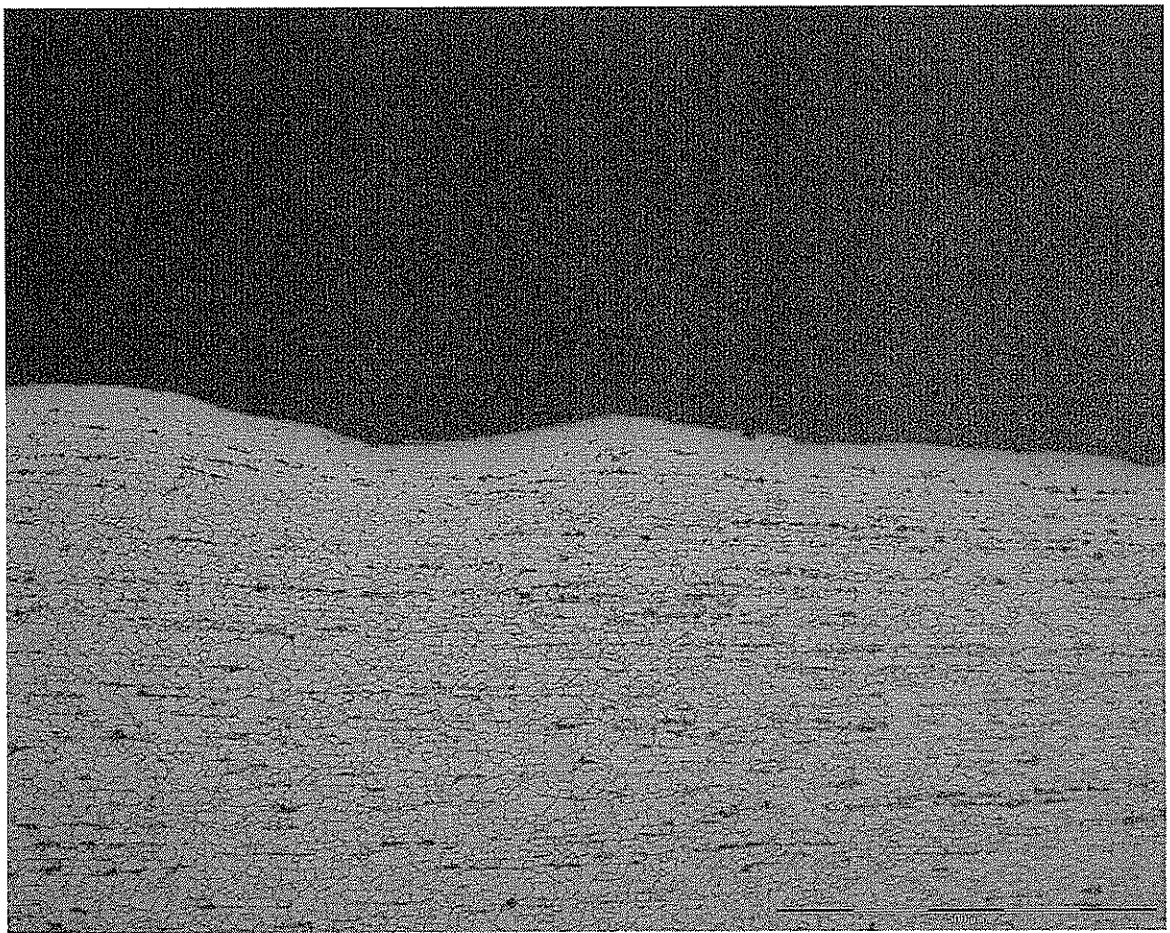

PHOTO 2 X 100 MAGNIFICATION – COATING THICKNESS EXTERNAL SURFACE

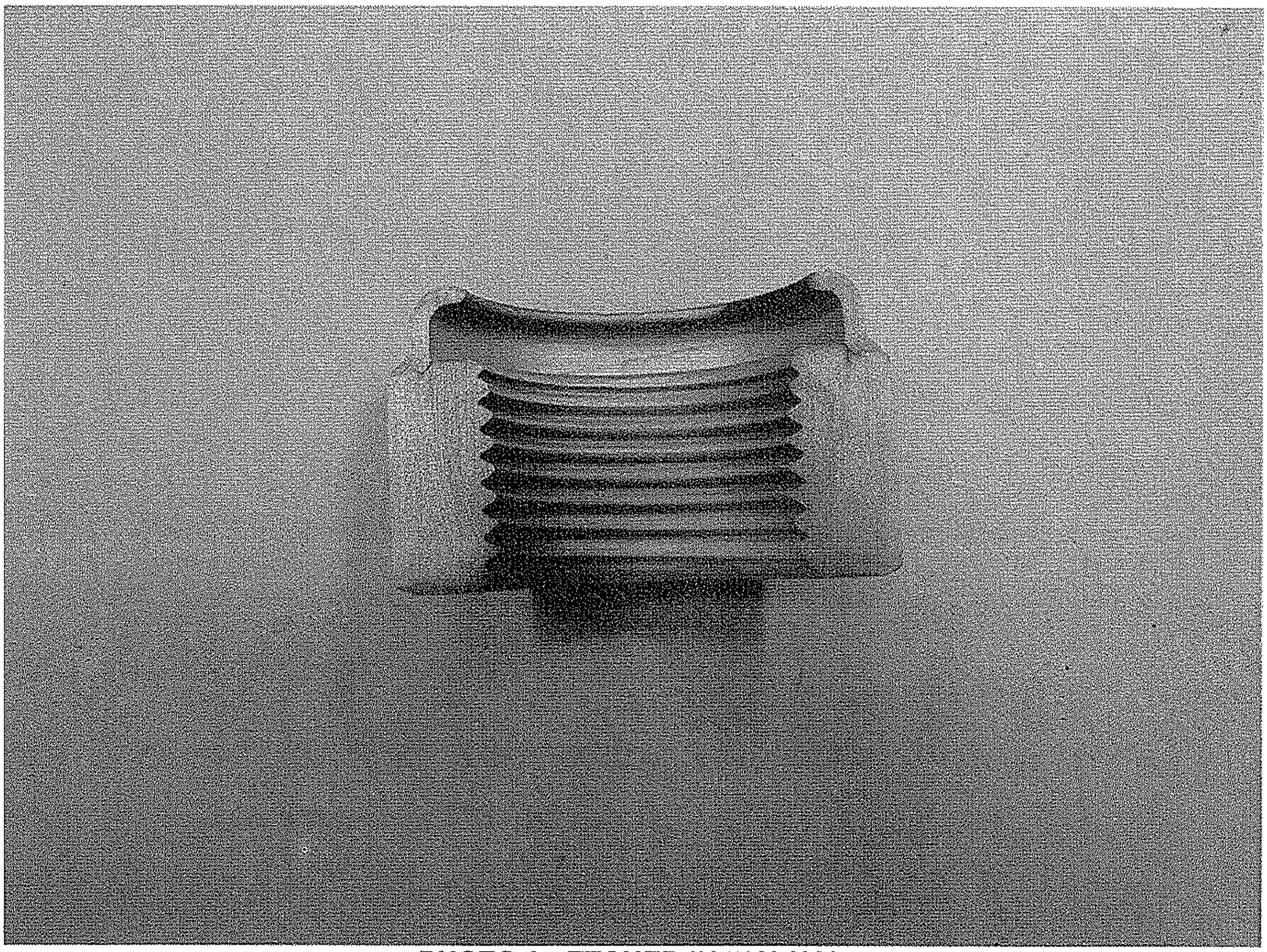

PHOTO 3 - ETCHED IN 50% HCL

PHOTO 4 - FERRITE/PEARLITE STRUCTURE X100 MAGNIFICATION

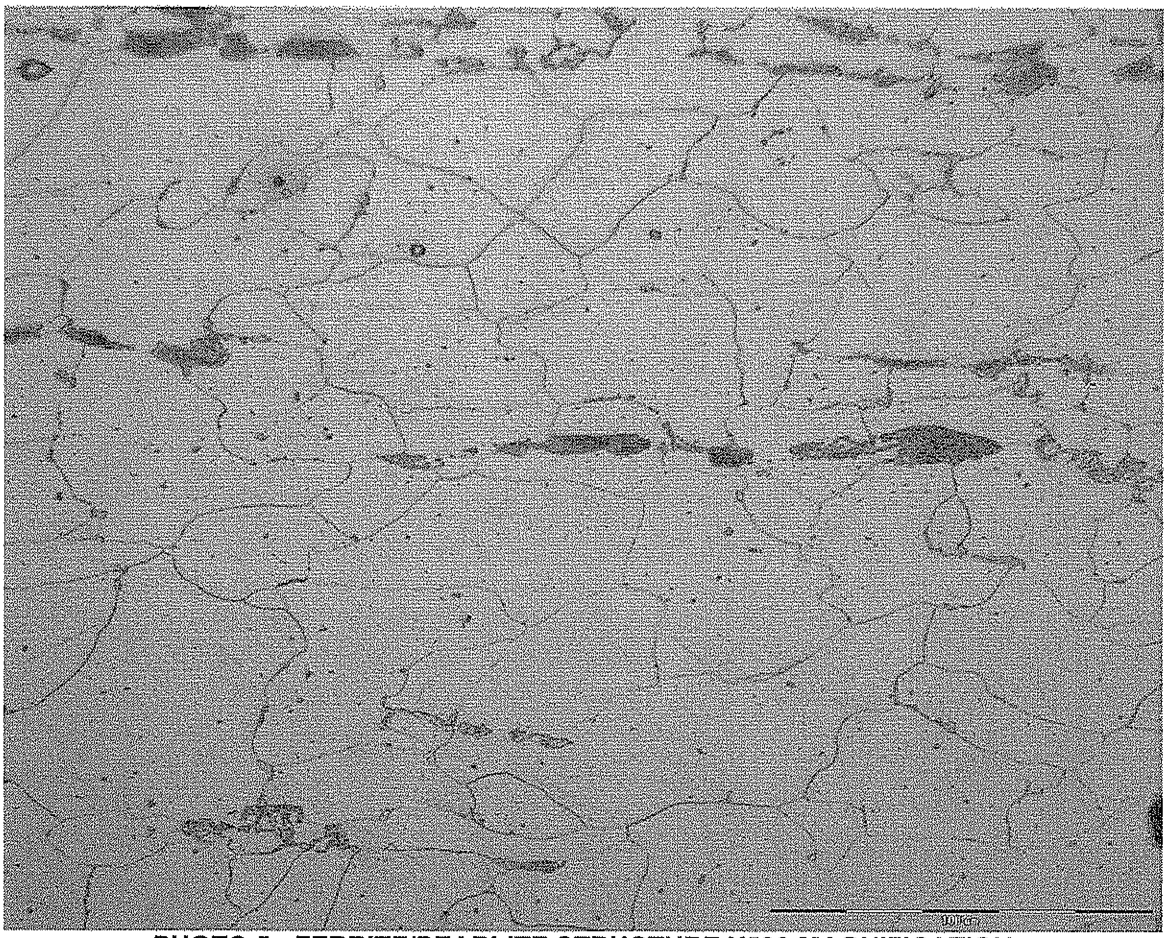

PHOTO 5 - FERRITE/PEARLITE STRUCTURE X500 MAGNIFICATION

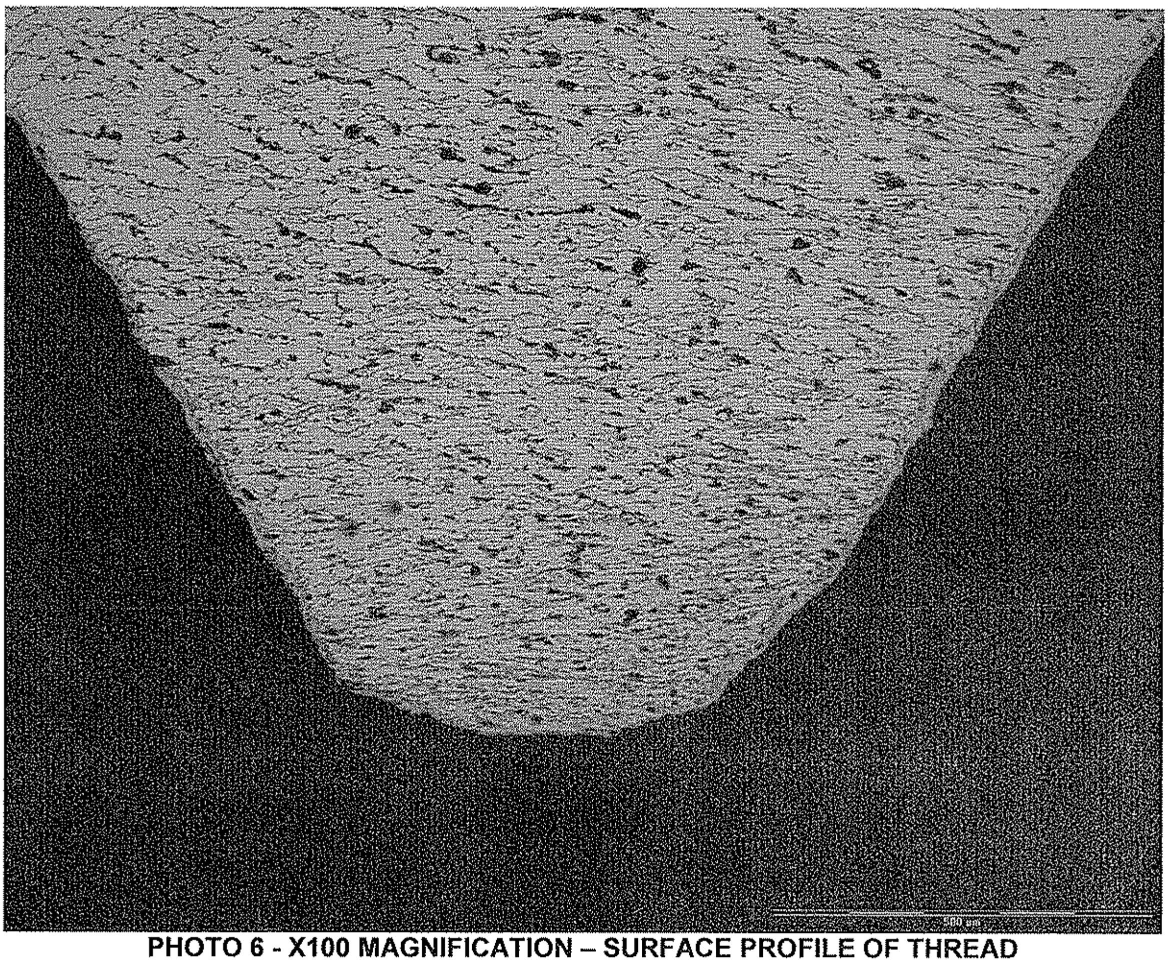

PHOTO 6 - X100 MAGNIFICATION – SURFACE PROFILE OF THREAD

Annex C

Report from Bolt Science on the failure of the Holding Down Fasteners

Report on the Failure

of the

Holding Down Fasteners on a Crane

Report completed for
Marine Accident Investigation Branch

Report completed by:
[redacted] *CEng BSc PhD MIMechE*
Bolt Science Limited
(www.boltscience.com)
6 October 2015

Contents

1. Abstract and Introduction

My understanding is that a number of fasteners failed that were holding down a crane to its foundation. The holding down fasteners consisted of two rows of M24 property class 10.9 studs, each row having 4 sets of fasteners. The studs had M24 x 2 (fine pitch series) threads. On each stud two nuts were located under the foundation with a single, thin, non-metallic prevailing torque type ('Nyloc') nut used to tighten the stud.

A summary of the results of the calculations completed in this report are:

Calculated thread stripping load for the nut:	254 kN
Anticipated preload range with a tightening torque of 600 Nm	139 to 231 kN
Anticipated preload range with a tightening torque of 350 Nm	48 to 133 kN
Load acting on a single bolt due to lifting the load at the point of failure	90.6 kN

Since the applied force from lifting is significantly lower than the anticipated thread stripping load, then either one or more of the points below likely applies:

1. One or more of the nut threads had been partially stripped (sheared) by the tightening process, the applied loading subsequently sustained being sufficient to completely shear the threads.
2. The thread tolerances are not as assumed in the analysis, that is, the thread dimensions were outside normal practice.
3. The deformation of the washers resulted in the force needed to strip the threads being reduced.
4. The fasteners were not evenly loaded due to the orientation of the applied load or due to some fasteners being only partially tight.

The thread tolerances that have been used in this report are based upon industry standard practices but it is unknown at the time of writing what are the thread dimensions of the nuts and bolts involved in this accident. Thread dimensions have a significant influence on the thread stripping characteristics.

It is uncertain whether the collapse of the washer occurred during tightening or was damaged during the collapse of the crane. If the deformation of the washer occurred when the nuts were initially tightened, which is a distinct possibility, it would point towards the preload in the fastener being high. The large gap under the washer would also result in bending stresses being incurred by the washer in addition to the direct bearing stresses. If the washer distorted during tightening, an uneven loading would occur in the threads resulting in thread stripping load being lowered. Such loading is not represented in a standard proof load test or in the calculations presented in this report. The author has previously observed such a reduction on an offset pull test on a fine threaded fastener.

It is good practice for the nuts to be as strong, or stronger, than the bolts to avoid the risk of thread stripping if the nuts are over-tightened. That is, considering that property class 10.9 bolts had used in this application, full height property class 10 nuts should have been used.

2. Details of the Installation/Joint

My understanding is that a number of fasteners failed that were holding down a 170.2 VCS crane to its foundation. The holding down fasteners consisted of two rows of M24 property class 10.9 studs, each row having 4 sets of fasteners. The studs had M24 x 2 (fine pitch series) threads. The arrangement of the assembly and the joint is shown in figure 1. On each stud two nuts were located under the foundation with a single thin non-metallic prevailing torque type ('Nyloc') nut used to tighten the stud. My understanding is that the nut was to the standard DIN 985[1]. The key dimensions of the nut, taken from the DIN standard are shown in figure 2.

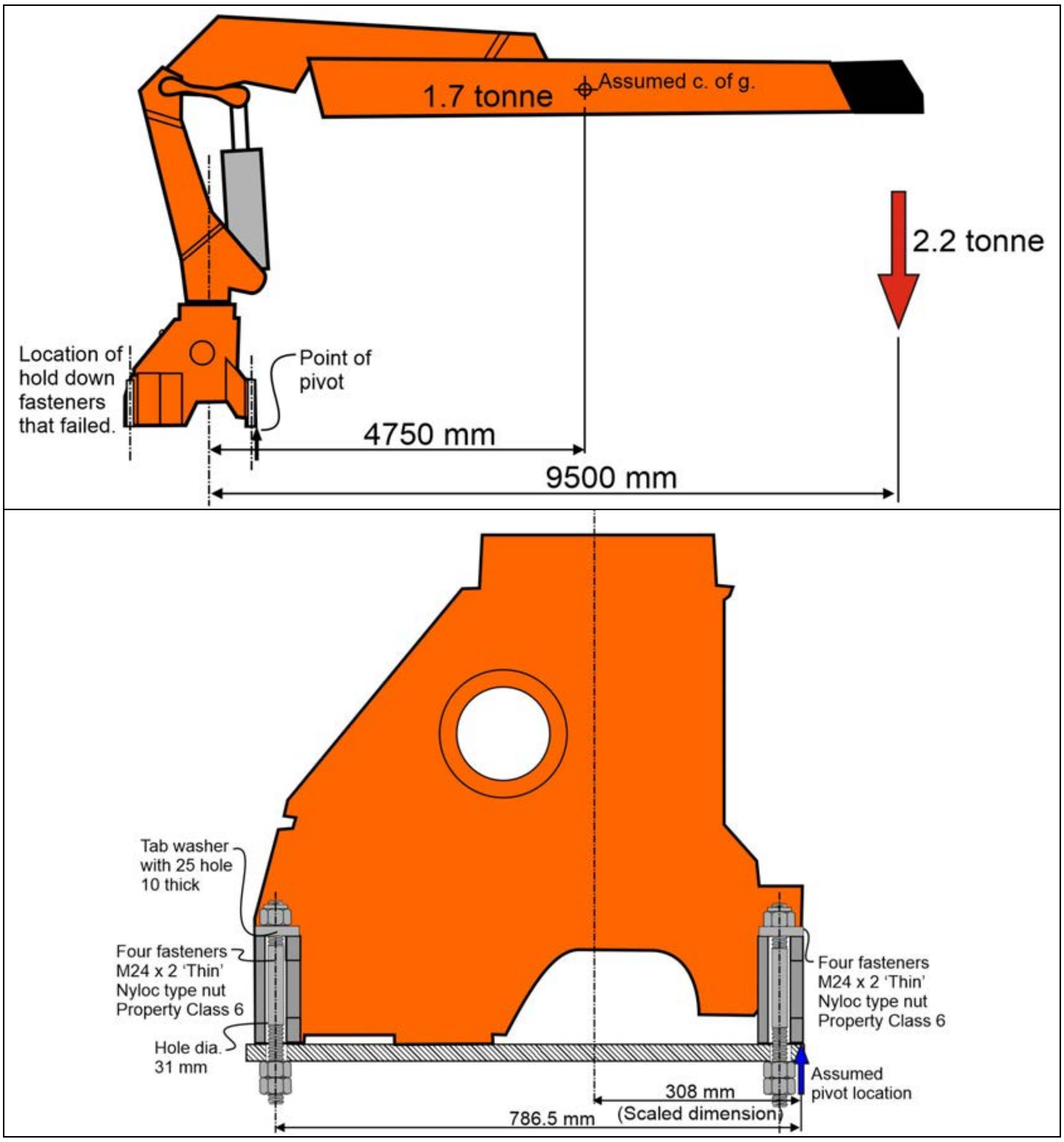

Figure 1 Arrangement of the installation/joint

[1] DIN 985 Prevailing torque type thin nuts with nonmetallic insert.

My understanding is that when the crane was lifting a weight of 2.2 tonnes, the nuts on one side failed by thread stripping allowing the crane to topple.

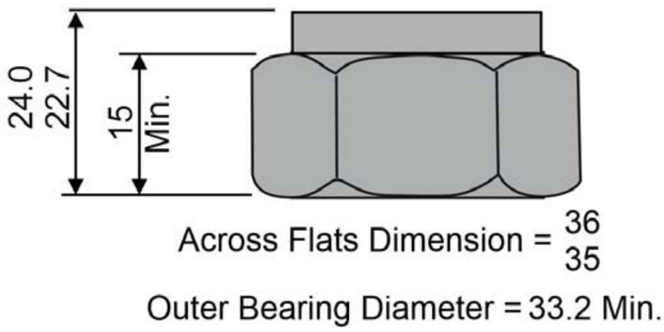

Figure 2 Key dimensions of the M24 x 2 Prevailing Torque Nut based upon DIN 985

Based upon a metallurgical investigation report[2], a hardness survey revealed that the hardness was 225 HV10. From BS EN ISO 18265 this hardness corresponds to a tensile strength of 720 MPa.

As can be seen in figure 3, one or more of the washers had been severely distorted, before or as a result of the accident.

Figure 3 Photo of one of the washers following the accident

[2] The report has been redacted - no reference available at the time of writing.

3. Loading acting on the fasteners at the time of the accident.

Just prior to the failure, the fasteners would sustain the load/moment from the crane as direct axial forces. On the side which the fasteners failed the loading would be tending to push the nut off the stud. The assumed dimensions, to allow an estimate of the axial force acting on an individual fastener are shown in figure 4.

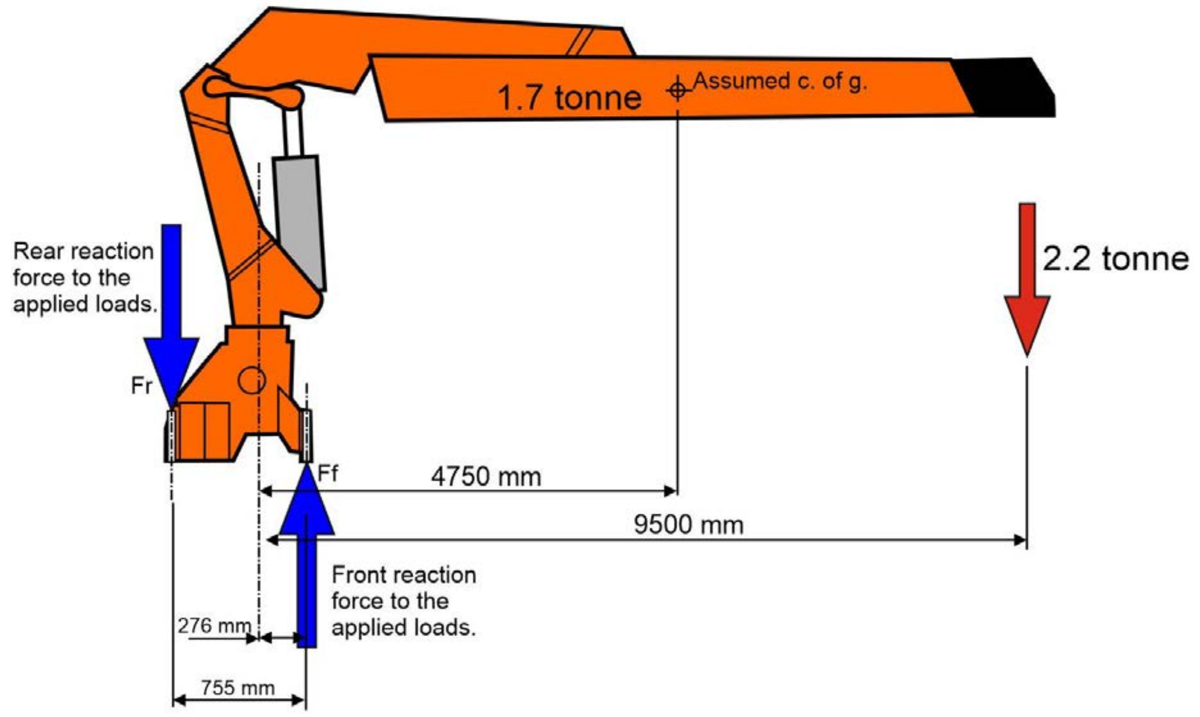

Figure 3 Fastener reaction forces

Taking moments about the front reaction point Ff gives:

$$2.2(9500-276)+1.7(4750-276)=755\ x\ Fr$$
$$Fr=36.95\ tonne=362497\ N$$

The load per fastener would be 362497 / 4 = 90624 N

4. Thread Stripping Calculations

Thread stripping is a failure mode that is best avoided since it may go unnoticed at the time of assembly. It starts at the first engaged thread, due to thread deformations causing it to carry the highest load and successively shears off subsequent threads. This may take several hours to complete and so may appear fine at the time of assembly. The risk is therefore present that a defective product may enter service. In this application, because of the short length of engagement and the material used for the internal thread being substantially weaker than the external thread, overtightening the bolt could cause thread stripping to occur.

Thread stripping strength depends upon the area of the thread being stripped and the shear strength of the thread material. The calculations will be based upon the work of Alexander[3], his work is also used in VDI 2230. The stripping strength of the external thread is, in general, different to the stripping strength of the internal thread. The thread stripping strengths are:

$$F_{\text{int}} = AS_s \,.\, \tau_{\text{int}} \,.\, C1 \,.\, C3$$

$$F_{ext} = AS_n \,.\, \tau_{ext} \,.\, C1 \,.\, C2$$

Where
F_{int} = Stripping strength of the internal thread
F_{ext} = Stripping strength of the external thread
τ_{int} = Shear strength of the internal thread material
τ_{ext} = Shear strength of the external thread material
UTS = Tensile strength of the thread
ASs = Shear area of the external thread
ASn = Shear area of the internal thread
Shear Ratio = Ratio of the shear to tensile strengths.
C1 = Nut dilation factor
C2 = Thread bending factor for external threads
C3 = Thread bending factor of internal threads

The shear areas can be calculated using the equations shown below. For the external thread:

$$AS_s = \frac{\pi}{p} \,.\, LE \,.\, D_{1\max} \left(\frac{p}{2} + 0.57735 \left(d_{2\min} - D_{1\max} \right) \right)$$

and for the internal thread

$$AS_n = \frac{\pi}{p} \,.\, LE \,.\, d_{\min} \left(\frac{p}{2} + 0.57735 \left(d_{\min} - D_{2\max} \right) \right)$$

Where:
p = Thread Pitch mm
LE = Length of Thread Engagement mm
D_{1max} = Maximum minor diameter of the internal thread mm
D_{2max} = Maximum pitch diameter of the internal thread mm
d_{2min} = Minimum pitch diameter of the external thread mm
d_{min} = Minimum major diameter of the external thread mm

No information has been provided as to the tolerance classes of the male or female thread. Standard practice is to use a 6g tolerance class for the bolt thread and a 6H tolerance class for the nut thread. My

[3] Alexander, E.M., *Analysis and design of threaded assemblies*. 1977 SAE Transactions, Paper No . 770420

understanding is that the major diameter of the bolt thread is within the tolerance band for a 6g tolerance class. Based upon a 6g/6H tolerance class combination, the thread details are:

Parameter	**Symbol**	**Formula**	**Value**
Basic major diameter	d	Nominal size	24
Thread Pitch	p	pitch	2
Basic pitch diameter	d_2	d - 0.6495p	22.701
Basic minor diameter	d_1	d - 1.0825p	21.835

External Thread M24 x 2 6g	
Maximum Major Dia.	23.962 mm
Minimum Major Dia.	23.682 mm
Maximum Pitch Dia.	22.663 mm
Minimum Pitch Dia.	22.493 mm
Maximum Minor Dia.	21.835 mm
Minimum Minor Dia.	21.261 mm

Internal Thread M24 x 2 6H	
Minimum Major Dia.	24.000 mm
Maximum Pitch Dia.	22.925 mm
Minimum Pitch Dia.	22.701 mm
Maximum Minor Dia.	22.210 mm
Minimum Minor Dia.	21.835 mm

The ISO 898-1 standard specifies the mechanical property requirements for a bolt of property class 10.9:

Minimum Tensile Strength	1040 N/mm²
Maximum Hardness	39 HRC
Maximum Tensile Strength	1230 N/mm² (note this is the conversion 39 HRC[4])
Ratio of the shear to tensile strength	0.62[5]
Minimum Bolt Shear Strength	645 N/mm²

Based upon a laboratory report[6], the nut hardness was measured as 225 HV10. From BS EN ISO 18265, this hardness converts to a tensile strength of 720 N/mm². Taking the ratio of shear to tensile strength as being 0.65[7] gives a shear strength of 468 N/mm².

Figure 2 shows the overall dimensions of the nut taken from DIN 985[8]. This standard shows that the bearing face is countersunk. The countersink region will not be effective in carrying load when compared to the full thread. The diameter of the countersink has a maximum diameter of 25.9 mm. Alexander allowed 40% effectiveness for the height of the countersink. Assuming countersinking on the bearing face side only, the height of the countersink section will be:

$$LE = 15 - \left[\frac{25.9 - 21.835}{2}\right](1 - 0.4) = 13.78\ mm$$

Based upon research reported by Alexander, part of the nut thread can be bell-mouthed. Bell-mouthing is partial tapering of the tapped hole due to instability of the drill as it first creates the hole. Bell-mouthing affects the shear strength of the bolt thread. This can be accounted for by varying the diameter over

[4] The standard: *BS EN ISO 18265: 2003 - metallic materials - conversion of hardness values*, was used to convert the hardness value to an approximate tensile strength.
[5] Taken from table 7 for a property class 10.9 bolt in *VDI 2230 Part 1 (December 2014) Systematic calculation of highly stressed bolted joints. Joints with one cylindrical bolt*
[6] The report has been redacted - no reference available at the time of writing.
[7] Table 6 in *VDI 2230 Part 1 (December 2014) Systematic calculation of highly stressed bolted joints. Joints with one cylindrical bolt* quotes the shear ratio as being between 0.65 to 0.85 for heat treatable steel.
[8] The nuts are stated to be to DIN 985 by the laboratory report provided to the author.

discreet increments. The maximum degree of bell-mouthing is approximately 1.03 times the minor diameter and can be accounted for by employing the mean diameter over the length of bell-mouthing. The length of the bell-mouthed section LB is typically taken as being 0.5 d where d is the nominal thread diameter. The mean diameter of the bell mouthed section of the nut D_m is:

$$D_m = \frac{(D_1 + 1.03D_1)}{2} = \frac{(21.835 + 1.03x21.835)}{2} = 22.163mm$$

The internal thread shear area will be:

$$AS_n = \frac{\pi}{p} . LE . d_{\min}\left(\frac{p}{2} + 0.57735(d_{\min} - D_{2\max})\right)$$

$$AS_n = \frac{\pi}{2} x\, 13.78\, x\, 23.682\left(\frac{2}{2} + 0.57735(23.682 - 22.925)\right) = 736.65mm^2$$

The external shear area will be:

$$AS_s = \frac{\pi}{p} . LE . D_{1\max}\left(\frac{p}{2} + 0.57735(d_{2\min} - D_{1\max})\right) = \frac{\pi}{2} x\, 13.78x\, 22.210\left(\frac{2}{2} + 0.57735(22.493 - 22.210)\right) = 559.3\, mm^2$$

Under load the bearing face of the nut expands radially. This nut dilation is due to the wedging action of the 60 degree threads which has the effect of increasing the minor diameter and reducing the effect shear areas of both the nut and bolt. Alexander accounted for this effect by the use of a nut dilation factor C1.

The nut dilation factor C1 is determined using the following equation:

$$C1 = \left[-\left(\frac{S}{D}\right)^2 + 3.8\left(\frac{S}{D}\right) - 2.61\right] \quad for\ 1.4 \le \frac{S}{D} < 1.9 \qquad \frac{S}{D}\ is\ the\ width\ across\ flats\ to\ nom.\ dia.$$

For this nut: S/D = 35/24 = 1.458

$$C1 = \left[-\left(\frac{S}{D}\right)^2 + 3.8\left(\frac{S}{D}\right) - 2.61\right] = \left[-(1.458)^2 + 3.8(1.458) - 2.61\right] = 0.805$$

The thread bending factor for internal threads C3 can be determined from the following equation:

$$C3 = 0.728 + 1.769R_s - 2.896R_s^2 + 1.296R_s^3 \quad for\ 0.4 < R_s < 1$$

$$C3 = 0.897\ for\ R_s \ge 1$$

The strength ratio Rs is defined as: $R_s = \frac{(\sigma_n AS_n)}{(\sigma_s AS_s)}$

where σ_n is the tensile strength of the nut and σ_s is the tensile strength of the bolt. Substituting values gives:

$$R_s = \frac{(\sigma_n AS_n)}{(\sigma_s AS_s)} = \frac{(720x736.65)}{(1040x559.3)} = 0.912$$

Using this value to determine the C3 factor:

$$C3 = 0.728 + 1.769x0.912 - 2.896x0.912^2 + 1.296x0.912^3 = 0.916$$

The force needed to strip the internal thread will be:

$$F_{\text{int}} = AS_s \,.\, \tau_{\text{int}} \,.\, C1 \,.\, C3 = 736.65 \; x \; 468 \; x \; 0.805 \; x \; 0.916 = 254213 \; N$$

Since it has been found that nut dilation more readily occurs when the threads are rotating, the thread stripping load achieved when a nut is being tightened can be less than that which would be achieved by a proof load type of test.

5. Torque-Preload Calculations

Introduction

The mathematical model relating the applied torque and tension in the bolt (preload) is generally given by:

$$T = \frac{F}{2}\left[\frac{p}{\pi} + \frac{\mu_t d_2}{\cos\beta} + D_e \mu_n\right]$$

This can be simplified for metric and Unified thread forms to:

$$T = F\left[0.15915p + 0.57735\mu_t d_2 + \frac{D_e \mu_n}{2}\right]$$

with $\mathrm{D_e} = \dfrac{d_o + d_i}{2}$

where

T	Total tightening torque
F	Bolt preload
μ_t	Coefficient of friction for the threads
d	Nominal major diameter of the thread i.e. for a M24 thread d=24mm
d_2	The basic pitch diameter of the thread. For metric threads d_2=d-0.6495p
β	The half included angle for the threads
p	Pitch of the thread.
μ_n	Coefficient of friction for the nut face or bolt head
D_e	The effective bearing diameter of the nut
d_o	The outer bearing diameter of the nut
d_i	The inner bearing diameter of the nut face

My understanding is that the nuts were tightened to 600 Nm. The torque specified by the manufacturer is 350 Nm. The major unknown in determining the bolt preload that would result from the tightening torque is the value of the coefficient of friction. This nut had a prevailing torque created by a non-metallic (polymer) insert. My understanding is that the nuts had a zinc coating. Such nuts can be provided with a light coating of lubricant, often wax, to prevent galling.

Based upon VDI 2230, galvanic coatings such as the zinc coating used in this application would fall into friction coefficient class B (friction coefficient range 0.08 to 0.16) if a lubricant had been used and into friction coefficient class C (friction coefficient range 0.14 to 0.24) without a lubricant.

The countersink diameter of the nut is 25.9 mm diameter which would be the inner bearing diameter (my understanding is that the tab washer has a hole diameter of 25 mm). The outer bearing diameter is taken as 33.2 mm.

The effective bearing diameter is: $\mathrm{D_e} = \dfrac{d_o + d_i}{2} = \dfrac{33.2 + 25.9}{2} = 29.55 \; mm$

The torque-tension equation can be adjusted to allow for the prevailing torque that the nut will possess and can be transposed to give:

$$F = \frac{T - T_p}{\left[0.15915p + 0.57735\mu_t d_2 + \frac{D_e \mu_n}{2}\right]}$$

Where T_p is the prevailing torque of the nut. My understanding is that there is no information on the actual prevailing torque of the nut. The standard BS EN ISO 2320[9] provides some details as to the likely value of the prevailing torque on first tightening. For a M24 property class 6 nut, the specified maximum first installation prevailing torque for a non-metallic insert type nut is 40 Nm with the minimum first removal torque being 11.5 Nm. From experience of testing this type of nut, in practice, the prevailing torque displayed by this type of nut tends to be closer to the first removal torque. Hence in these calculations a prevailing torque of 11.5 Nm will be assumed.

Whether a thin or normal height type of nut is used does not make a difference to the torque-tension relationship if it is assumed that the coefficient of friction is the same in both circumstances. Also, if it is assumed that the coefficient of friction is the same, whether a plain nut or one having a nut with a prevailing torque is used makes only a small difference in the preload. (This is due to the prevailing torque normally being only a small fraction of the overall tightening torque.)

Nuts tightened to 600 Nm

For a tightening torque of 600 Nm with a coefficient of friction of 0.08, the bolt preload F would be:

$$F = \frac{T - T_p}{\left[0.15915p + 0.57735\mu_t d_2 + \frac{D_e \mu_n}{2}\right]} = \frac{[600 - 11.5]\,x\,1000}{\left[0.15915\,x\,2 + 0.57735\,x\,0.08\,x\,22.701 + \frac{29.55\,x\,0.08}{2}\right]} N = 230919$$

For a tightening torque of 600 Nm with a coefficient of friction of 0.14, the bolt preload F would be:

$$F = \frac{T - T_p}{\left[0.15915p + 0.57735\mu_t d_2 + \frac{D_e \mu_n}{2}\right]} = \frac{[600 - 11.5]\,x\,1000}{\left[0.15915\,x\,2 + 0.57735\,x\,0.14\,x\,22.701 + \frac{29.55\,x\,0.14}{2}\right]} N = 139409$$

For a tightening torque of 600 Nm with a coefficient of friction of 0.24, the bolt preload F would be:

$$F = \frac{T - T_p}{\left[0.15915p + 0.57735\mu_t d_2 + \frac{D_e \mu_n}{2}\right]} = \frac{[600 - 11.5]\,x\,1000}{\left[0.15915\,x\,2 + 0.57735\,x\,0.24\,x\,22.701 + \frac{29.55\,x\,0.24}{2}\right]} N = 83957$$

The stress area (A_s) of the thread can be calculated from:

$$A_s = 0.7854(d - 0.9382p)^2 = 0.7854(24 - 0.9382x2)^2 = 384.4\,mm^2$$

Hence the above loadings represent a direct stress in the bolt threads of between:

$$\sigma = \frac{230919}{384.4} = 600.7\,N/mm^2 \quad to \quad \frac{83957}{384.4} = 218.4\,N/mm^2$$

These stresses represent a utilisation of the minimum yield strength (the 0.2% non-proportional limit) for a property class 10.9 bolt of between 23% to 64%.

[9] BS EN ISO 2320 - Prevailing torque type steel hexagon nuts - mechanical and performance requirements

The bearing stress would be: $\sigma = \dfrac{230919}{\frac{\pi}{4}\left(33.2^2 - 25.9^2\right)} = 681\ N/mm^2 \quad to \quad \dfrac{83957}{\frac{\pi}{4}\left(33.2^2 - 25.9^2\right)} = 248\ N/mm^2$

Nuts tightened to 350 Nm

Repeating the above calculations with a torque of 350 Nm gives:

For a tightening torque of 350 Nm with a coefficient of friction of 0.08, the bolt preload F would be:

$$F = \frac{T - T_p}{\left[0.15915p + 0.57735\mu_t d_2 + \frac{D_e \mu_n}{2}\right]} = \frac{\left[350 - 11.5\right] x\, 1000}{\left[0.15915\ x\ 2 + 0.57735\ x\ 0.08\ x\ 22.701 + \frac{29.55\ x\ 0.08}{2}\right]} = 132807\ N$$

For a tightening torque of 350 Nm with a coefficient of friction of 0.14, the bolt preload F would be:

$$F = \frac{T - T_p}{\left[0.15915p + 0.57735\mu_t d_2 + \frac{D_e \mu_n}{2}\right]} = \frac{\left[350 - 11.5\right] x\, 1000}{\left[0.15915\ x\ 2 + 0.57735\ x\ 0.14\ x\ 22.701 + \frac{29.55\ x\ 0.14}{2}\right]} = 80181\ N$$

For a tightening torque of 350 Nm with a coefficient of friction of 0.24, the bolt preload F would be:

$$F = \frac{T - T_p}{\left[0.15915p + 0.57735\mu_t d_2 + \frac{D_e \mu_n}{2}\right]} = \frac{\left[350 - 11.5\right] x\, 1000}{\left[0.15915\ x\ 2 + 0.57735\ x\ 0.24\ x\ 22.701 + \frac{29.55\ x\ 0.24}{2}\right]} = 48289\ N$$

Hence the above loadings represent a direct stress in the bolt threads of between:

$$\sigma = \frac{132807}{384.4} = 345\ N/mm^2 \quad to \quad \frac{48289}{384.4} = 126\ N/mm^2$$

These stresses represent a utilisation of the minimum yield strength (the 0.2% non-proportional limit) for a property class 10.9 bolt of between 13% to 37%.

The bearing stress would be: $\sigma = \dfrac{132807}{\frac{\pi}{4}\left(33.2^2 - 25.9^2\right)} = 392\ N/mm^2 \quad to \quad \dfrac{48289}{\frac{\pi}{4}\left(33.2^2 - 25.9^2\right)} = 143\ N/mm^2$

6. Discussion/Conclusions

A summary of the results of the calculations completed in this report are:

Calculated thread stripping load for the nut:	254 kN
Anticipated preload range with a tightening torque of 600 Nm	139 to 231 kN
Anticipated preload range with a tightening torque of 350 Nm	48 to 133 kN
Load acting on a single bolt due to lifting the load at the point of failure	90.6 kN

Since the applied force from lifting is significantly lower than the anticipated thread stripping load, then either one or more of the points below applies:

1. One or more of the nut threads had been partially stripped (sheared) by the tightening process, the applied loading subsequently sustained being sufficient to completely shear the threads.
2. The thread tolerances are not as assumed in the analysis, that is, the thread dimensions were outside normal practice.
3. The deformation of the washers resulted in the force needed to strip the threads being reduced.
4. The fasteners were not evenly loaded due to the orientation of the applied load or due to some fasteners being only partially tight.

The thread stripping calculations presented in this report are based upon Alexander's theory. To quote from BS EN ISO 898-2: 2014[10]: "*Extensive experimental tests proved Alexander's theory through practical results. Actual studies, including FEM-based calculations, confirmed Alexander's theory.*"

The thread tolerances that have been used in this report are based upon industry standard practices but it is unknown at the time of writing what are the thread dimensions of the nuts and bolts involved in this accident. Thread dimensions have a significant influence on the thread stripping characteristics.

It is uncertain whether the collapse of the washer occurred during tightening or was damaged during the collapse of the crane. If the deformation of the washer occurred when the nuts were initially tightened, which is a distinct possibility, it would point towards the preload in the fastener being high. My understanding is that the washer is made from mild steel. The limiting surface pressure for mild steel depends upon the specific steel used. For a low strength mild steel (tensile strength 340 N/mm^2), the limiting surface pressure[11] is 490 N/mm^2. This would point towards the preload value approaching or exceeding the thread stripping load.

The large gap under the washer would also result in bending stresses being incurred by the washer in addition to the direct bearing stresses. If the washer distorted during tightening an uneven loading would occur in the threads, resulting in the thread stripping load being lowered. Such loading is not represented in a standard proof load test or in the calculations presented in this report. The author has previously observed such a reduction on an offset pull test on a fine threaded fastener.

It is unknown whether all the nuts were tight, if one or more were loose this could result in the remaining fasteners sustaining a disproportionate share of the applied force from the lifting operation. This could result in the failure of the highest loaded fastener followed by load shedding, resulting in the failure of subsequent fasteners. Since thread shearing requires displacement of at least half the thread pitch, the load shedding hypothesis is a possibility but not probable in the author's view.

Fasteners of property class 10.9 and above are susceptible to hydrogen embrittlement and stress corrosion cracking. It is unknown to the author why property class 10.9 bolts are being used in an application where there is the potential for stress corrosion cracking to occur. At the 350 Nm specified torque level a property class 8.8 bolt would have been suitable, such bolts are not susceptible to hydrogen embrittlement.

[10] ISO 898-2 - 2012 Mechanical properties of fasteners made of carbon steel and alloy steel – Nuts with specified property classes – Coarse thread and fine pitch thread

[11] From table A9, *VDI 2230 Part 1 (December 2014) Systematic calculation of highly stressed bolted joints. Joints with one cylindrical bolt*

It is good practice for the nuts to be as strong, or stronger, than the bolts to avoid the risk of thread stripping if the nuts are over-tightened. That is, considering that property class 10.9 bolts had used in this application, full height property class 10 nuts should have been used.

[redacted]

[redacted] CEng FIMechE BSc PhD
Bolt Science Limited

Advisory Notice issued by the National Workboat Association

Gmail

[redacted]

Advice, Award and Reminders

1 message

National Workboat Association <secretary@workboatassociation.org> 20 November 2015 at 11:14

Reply-To: secretary@workboatassociation.org
To: secretary@workboatassociation.org

If you are having difficulties reading this email, click here

nwa

Tel +44 (0) 1430 470013

NWA Advisory Notice to Owners / Operators of Coded Workboats

As a result of independent investigations into several recent incidents concern has been raised that modifications and/or additions have been made to Code vessels without any reference to the Certifying Authority responsible for that vessel's coding. We have been asked by the MCA and MAIB to issue this advice;-

Owners and Operators are reminded of their responsibility under both MGN 280 and the revised Workboat Code, to "notify and seek approval from the Certifying Authority prior to implementing any change or modification to the vessel." – see Ch 27.11.5 of the revised code. Recent incidents have shown that even apparently minor alterations can have a serious effect on the stability and/or the safe operation of the vessel.

Whilst writing, we would also remind you that it is the Owner / Operator's responsibility to advise the Certifying Authority when a vessel is due for its next certification survey, to ensure the vessel is made available for survey and to ensure the certification is maintained in date.

Award in Maritime Studies: Use of Radar and Electronic Chart Systems in Code Vessels

This award has been created at the request of the NWA and MSA for those candidates who have

[redacted]

undertaken the following courses;-

"Use of Radar for Safe Navigation and Collision Avoidance on Domestic and Code Vessels"

"Operate non-ECDIS marine Electronic Chart Systems"

As you should be aware both these courses are strongly recommended under the revised Workboat Code and will become a requirement within 3 years. These courses are run separately as 3 days each, or can be run concurrently as a 5 day course. One of our Associate members, MARITAS, has been running these courses as their 'RADECS' course for some time and will now be the first training provider to gain approval to be able to offer this award as part of the course certification. Anyone interested in getting crew members onto these courses, or hosting a course at your premises should contact [redacted] at MARITAS – dc@maritas.co.uk

Hopefully, other Training Providers will be looking to offer this course going forward as there will be significant demand once the revised Workboat Code is published in the near future.

Reminder – Have you Booked you Room(s) for the Annual Dinner/AGM 21st/22nd January?

We had a good early flush of room bookings following the last Newsletter ref the Annual Dinner and AGM to be held at the Lakeside on 21st/22nd January, but we know there are quite a few regular attendees and new members who have not yet booked – I would encourage you to put your booking through to me –secretary@workboatassociation.org - sooner rather than later, as we do not want anyone to be disappointed.

– and if that is not sufficient warning for you , [redacted] says – if you haven't booked by the time I give him the list in a week or so, he'll be chasing you himself!! – you have been warned!

And Lastly, Don't Forget - Safety Forum – Weds 25th November at Hoylake Sailing School

Secretary

[redacted]