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Loading, construction and tests of the interlocks applicable to the ship deck crane with unsupported jib

SUMMARY

The jib luffing and slewing interlocks applicable to the ship deck crane are presented. The interlocks were provided to fix the crane jib without supporting it during sea voyage.

The paper contains description of design and operation mode of the interlocks, assumptions to and results of their calculations and tests carried out at the manufacturer's test station.

The paper was presented at the IV Conference on Port Technology and Ship Deck Equipment held in Międzyzdroje on 3 to 5 June 1998.

INTRODUCTION

Aiming at maximization of cargo handling equipment capacity and amount of the loads transported on ships leads to newer and newer demands on cargo handling facilities. Such requirements formed the basis for designing a universal ship equipped with the deck cranes so situated on the ship's deck that providing a support of the crane jib without lowering the assumed number of containers was not possible. Such support fixing the crane jib at the sea-going position during sea voyage is necessary to secure the crane structure against action of large forces due to motion accelerations of the ship in waves. The structure of the so fixed crane is much less loaded than during cargo handling operations, therefore the operation conditions are decisive of the design stresses. The conditions of not supporting the crane jib forced TOWIMOR, the crane supplier, to elaborate a new design of the crane which would comply with the shipyard requirements for the safe manufacturing and simultaneously with all safety conditions during sea voyage, inclusive of the allowable levels of both immediate and fatigue stresses. The design assumptions, calculations and documentation were jointly elaborated by the Ship Design and Research Centre, Gdańsk, Technical University of Szczecin (Department of Shipboard Mechanisms and Cranes) and TOWIMOR Design Office within the R&D project financially supported by the State Committee for Scientific Research. On the basis of results of the project the first crane was completed in October 1997 and 14 cranes were manufactured till the end of 1998.

DESCRIPTION OF THE CRANE CONSTRUCTION

A standard cargo crane of the parameters ensuring the demanded load hoisting capacity, reach and cargo handling rate served as a basis for elaboration of the new design. The main operational parameters of the crane are presented in Tab.1.

***Tab.1.** Technical characteristics of the crane*

Item	Parameter	Magnitude
1	Load hoisting capacity	40/32 t for the hook 26.6 t for the grab
2	Maximum reach	30 m at 40 t 33 m at 32 t and 26.6 t
3	Minimum reach	4 m
4	Total hoisting height	41.5 m
5	Hoisting speed	20/40 m/min
6	Luffing change time	74 s
7	Rotation speed	0.8 rpm
8	Total mass	62 t
9	Drive type	Electro-hydraulic
10	Supply voltage	3 x 440 V, 60 Hz

The crane is intended for fitting on ships to be used in the ports without cargo handling facilities. A sketch of the crane fitted on the column is shown in Fig.1. A way of parking the crane at the sea-going position, which could replace the jib support was the basic problem to be solved. The main condition was that during parking operation the jib had to be freely suspended by means of the ropes, and the crane frame duly fixed and secured against acceleration loads. Moreover, the parking operation was assumed to be automatically effected and controlled from the crane operator post. In consequence, interlocking the crane frame onto the flange of the column was the only possible solution. After a comprehensive analysis the following way, deemed optimal, of crane parking was chosen :

- to interlock the crane frame by means of the hydraulically controlled bolts placed in the crane frame and lowered into the sockets fitted to the deck column flange
- the jib was assumed to be freely suspended by the luffing ropes, and additionally the luffing winch drum was assumed to be locked to secure the brake and toothed gear.

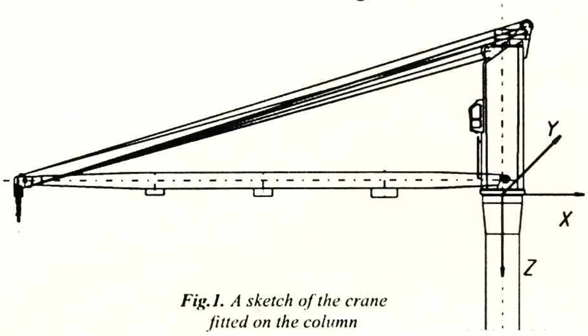

Fig.1. A sketch of the crane fitted on the column

LOADING OF THE CRANE WITH UNSUPPORTED JIB DURING SEA VOYAGE

Analyzing the influence of the ship's motion in rough weather on the crane structure was necessary to determine loads acting on the interlocks. To be on the safe side the loads were analyzed and determined on the basis of two calculation methods [1,2] as follows :

- The ship's operation conditions in waves was analyzed by means of the Design Sea Method (DSM). In this way the long – term response prediction of the ship in irregular waves with accounting for captain's reaction was obtained. The following conditions were chosen as being decisive of magnitude of the structural loadings :
  - applicable to transverse accelerations : the course angle of 120°, ship speed of 5 knots, wave height of 6 m, wave period of 8.37 s
  - applicable to vertical accelerations : the course angle of 150°, ship speed of 5 knots, wave height of 16 m, wave period of 9.93 s.

  The data were used for computer calculations of component acceleration values in the x,y,z coordinate system shown in Fig.1, and on their basis, for determination of the loads acting on the crane at its selected points.
- The conditions and values of the loads acting on the crane were checked by using results of the analytical calculations carried out in accordance with the classification societies' requirements : of Germanischer Lloyd [9] and Lloyds' Register [10].

Thus, the load values for immediate strength assessment could be determined. For fatigue strength assessment the equivalent loads were determined in accordance with the linear cumulation hypothesis of the damage due to the random wave loads acting on the ship during entire ship service life [2]. The loads for immediate strength assessment of the crane, wind loads and loads for fatigue strength assessment are presented in Tab.2,3 and 4, respectively [1]. Directions of the component loads are indicated in Fig.1.

Tab.2. The loads for immediate strength assessment of the crane, the wind loads excluded

		Force [kN]			Moment [kNm]		
Item	Loading case	P_z	P_y	P_x	M_z	M_y	M_x
1	Rolling state acc. GL	579	650	-	-5200	1820	4632
2	Pitching state acc. GL	942	-	335	-	-	6600
3	Complex state acc. LR	814	600	-204	-4800	1680	7083
4	Sea state 1 acc. DSM	855	550	-210	-4500	1550	7470
5	Sea state 2 acc. DSM	970	380	-340	-3100	1080	8750

Tab.3. The maximum wind loads acting on the crane acc. GL [9]

		Force [kN]			Moment [kNm]		
Item	Loading case	P_{zw}	P_{yw}	P_{xw}	M_{zw}	M_{yw}	M_{xw}
1	Beam wind	-	158	-	-1820	295	4632
2	Head wind	-	-	-30	-	-	113

Tab.4. The equivalent loads for fatigue strength assessment of the crane

		Force [kN]			Moment [kNm]		
Item	Load magnitude	P_{ze}	P_{ye}	P_{xe}	M_{ze}	M_{ye}	M_{xe}
1	Average	557	0	0	0	0	4480
2	Maximum	641	124	69	1000	350	5350
3	Minimum	473	-124	-69	-1000	-350	3610

INTERLOCKS

Slewing interlock

In the accepted design solution the slewing interlocks are loaded by the moment about z - axis (see Tab.2,3 and 4). Selecting the moment value of 4500 kNm obtained from the DSM (Sea state 1 in Tab.2) to be the input to the strength calculations was deemed most justified as based on the exact method. To be on the safe side the stress values due to 5200 kNm moment (Rolling state acc. GL in Tab.2) were also checked. For the immediate strength assessment the moment values were enlarged by 1820 kNm wind moment (Tab.3) with the course angle taken into account. For each of the load values the stresses in the bolts were calculated and compared with the permissible levels given by safety standards.

For fatigue strength assessment the moment range of ± 1000 kNm (Tab.4) was assumed without accounting for the wind loads. The strength of the interlocks was checked with the use of two methods : a preliminary analytical method and the more exact Finite Element Method (MSC NASTRAN software). The prelimiarily calculated stress values which the conceptual design was based on, are given below. The maximum total moment M_0 taken for the immediate strength assessment was determined as follows [3] :

$$M_O = M_Z + M_{ZW} \cdot \cos 30^o$$

A sketch of the slewing interlock is presented in Fig.2. The interlock consists of two identical bolt units fitted inside the lower part of the crane frame and activated by means of the hydraulic cylinders [4,5]. They are electro-hydraulically controlled from the luffing mechanism installation.

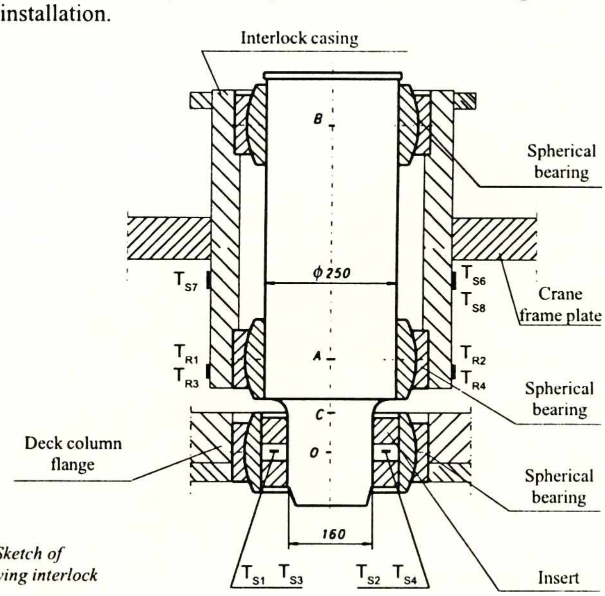

Fig.2. Sketch of the slewing interlock

As soon as the crane is slewed into the sea-going position that is indicated by a rotation sensor, the hydraulic cylinders are activated from the operator post and the bolts are lowered into the sockets fixed to the deck column flange. The bolts are the most loaded elements. Both pressures and bending stresses in their cross-sections (points) A, B, C and O were checked [3]. The preliminary calculations revealed that if the typical bronze sliding sleeves are selected for bearing the bolt then large pressures exceeding the allowable values appear at the sleeve edges. Therefore self-aligning, spherical sliding bearings were applied in the guiding parts of the bolts. The inner ring of the bearing was made of BA1032 bronze. The solution was registered in the Polish Patent Office.

The calculations of the design solution performed with the assumed uniform distribution of pressure confirmed the magnitude of the compressive stresses between 107 and 118 MPa appearing at the point A (Fig.2) of the most loaded bearing, dependent on assumed loading. The stress values are on the safe side ensuring the correct work of sleeves. The largest calculated stress values appeared at the cross-section A and C of the bolts. The reduced stress calculated from the formula :

$$\sigma_{zr} = \sqrt{\sigma^2 + 3\tau^2}$$

were : at A : 338 to 372 MPa, and at C : 237 to 261 MPa dependent on assumed loading. The solution complied with safety criteria as 34HNM surface-quenched steel was applied with the assumed allowable stress value of 486 MPa = 0.9 R_e.

The fatigue strength of the bolts was assessed in accordance with DIN 743 standard [12]. The cross-section C of the bolt was checked in the region where the sharpest notch is placed (Fig.3).

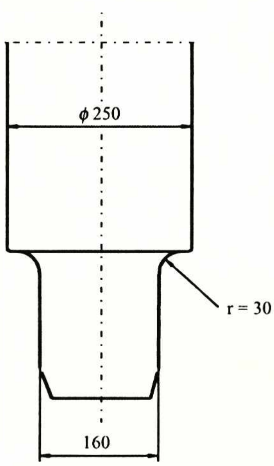

Fig.3. Lower part of the slewing interlock bolt

The calculations confirmed that the infinite fatigue life of the bolt is ensured.

Luffing interlock

The luffing interlock is loaded by the force in the rope due to M_y and P_z vertical loadings. The force S loading the rope system in the most unfavourable case amounts to S = 650 kN [2]. Therefore the force P in the rope amounts to P = S/n = 108.3 kN , as the compound pulley reduction ratio n = 6 was assumed for the applied rope system. The strength of the luffing drum interlock was assessed for the so determined value of the rope force. The interlock latch is loaded by the force :

$$F = M_p/b \text{ [kN]}$$

where :

$M_p = P \times R_b$ [kNm] - luffing drum moment due to the force P
R_b - luffing drum radius
b - distance between the point of application of the force F and drum axis

A scheme of the interlock is presented in Fig.4.

The luffing interlock is fitted to the luffing winch [4,5]. This is a locking spring latch controlled by means of a hydraulic cylinder supplied from the crane control circuit. When the interlock is activated the latch is shifted and wedged at the rope drum flange, thus rotating the drum is not possible any longer and no failure of the winch brake can ocuur. The total load is transferred by the latch which works as a wedge. The stress calculations [3] shows that the pressure values are rather low (64 MPa) so the adequate strength of the device is ensured.

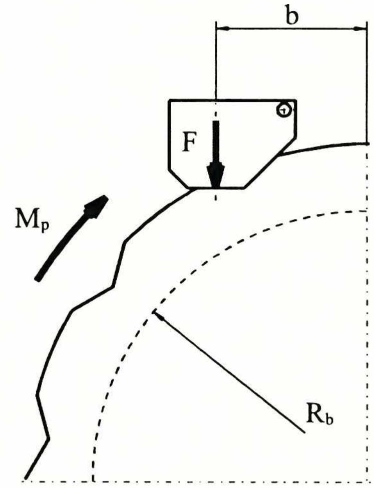

Fig.4. Scheme of the luffing drum interlock

TESTS AND EXPERIMENTAL INVESTIGATIONS OF THE INTERLOCKS

The interlock tests contained a series of investigations under the working and sea-going conditions [6]. The test station had to be adapted to the high loads to cope with these demanding conditions [7].

Investigations of the luffing interlock

The investigations which consisted in testing the functioning and strength of the interlock were performed on the complete crane assembled at the test station.

The functioning test was carried out on the crane with no weight on the hook and consisted in multiple activation of the latch from the operator's post. The crane jib was then put into a position close to sea-going one.

The strength test was performed on the crane with 40 t weight suspended on the hook and the jib placed at 60° angle to the horizontal line (giving 17.1 m reach). In this condition the maximum value of the luffing rope loading (of about 108 kN) could be expected during sea voyage. The tests confirmed that the crane strength was appropriate and that after small corrections of the latch shape the interlock operated adequately.

Investigations of the slewing interlock

The investigations which consisted in testing the functioning and strength of the interlock were carried out in several stages on the crane frame itself fixed to the test stand before the jib has been assembled.

The functioning test was performed on the not loaded crane frame. It consisted in multiple running onto the slewing gauge and activating the interlock bolts from the operator's post.

The strength test was performed after fitting the test hydraulic cylinders to the appropriately adapted test stand. By the cylinders the test loads were exerted on the interlock bolts through the jib pins. The loads were step-by-step increased by increasing the pressure in the hydraulic cylinders and simultaneously stress values were measured with the use of the strain gauges glued on the bolts, and recorded [6,7]. The stress measurement points, marked by T_S and T_R (the arrangement of the strain gauges) are shown in Fig.2 according to [8].

External loading values exerted by the test hydraulic cylinders are shown in Tab.5.

Tab.5. *Test loading on the crane frame*

		Hydraulic cylinder I	Hydraulic cylinder II
Test no.	Loading on the crane frame	Force [kN]	Force [kN]
1	M_t = 2670 kNm, P_t = 0 kN	1000	-1000
2	M_t = 4005 kNm, P_t = 0 kN	1500	-1500
3	M_t = 5186 kNm, P_t = 685 kN	2285	-1600

They correspond to the loading due to the rolling state given in Tab.2 according to GL requirements. The loading duration time was 60 s for each test. Values of the tensile (or compressive) and reduced stresses based on strain gauge measurements in T_S points (tensile or compressive stress from single strain gauges) and T_R points (reduced stress calculated from rosette strain gauges records), shown in Fig.2, are presented in Tab.6.

Tab.6. *Tensile (or compressive) and reduced stresses in the interlock bolts*

Strain gauge no.	Test 1	Test 2	Test 3
T_{R1}	88.4	150	194
T_{R2}	18.2	24.1	30.7
T_{R3}	20	26.3	28.6
T_{R4}	197	238	279
T_{S1}	-5.2	-8.9	-11
T_{S2}	96	188	274
T_{S3}	15.1	12.2	9.1
T_{S4}	5.4	3.7	1.4
T_{S5}	-102	-134	-165
T_{S6}	19	21	22.7
T_{S7}	24.7	30.1	35.8
T_{S8}	-99.9	-130	-165

The test results demonstrated that the loading applied to each of the two interlocking bolts was not uniformly distributed. The results made determining the loading assymetry possible and, in consequence, the bolts could be designed with accounting for that phenomenon. Also, during the tests the maximum loading exerted on the bolts fixing the jib to the frame was determined.

It was simultaneously revealed that in the case of the crane with unsupported jib the sea loading, and not operational one, was the design (dimensioning) load for the bolts in question.

FINAL REMARKS

- In the paper the way of designing the interlock device, one of the main structural units of the ship deck crane adopted to parking the jib without any support was presented.
- During elaboration of the design, its realization and stand testing, verifying some design assumptions and solutions appeared necessary.
- For instance, comparison of the sea-voyage and working loading of the jib luffing mechanism revealed that the strength of the toothed gear and brake at both loading cases was sufficient and therefore application of the luffing drum interlock was unnecessary.
- It was decided, however, to leave the interlock for convenience of the crane operator as it makes overhauling the brake in the case of the unsupported jib possible.
- It was also revealed that if two bolts were applied to the slewing interlock the loading was distributed between them in a highly non-uniform way (see Tab.6) and the non-uniformity decreased as the loading increased; this aspect had to be taken into account in designing.
- In general, the stress level obtained from the crane testing was lower than that calculated, but modelling all the load components acting on the crane in seaway was not possible at the TOWIMOR's test station. At the same time, however, the investigations and measurements showed that the main design assumptions were in compliance with the measurement results.
- The structural safety of the deck crane with unsupported jib during sea, voyage was additionally confirmed with the use of the method of computer random process simulation [11] during elaboration of the design and its verification stage. The calculated stresses in all examined structural elements maintained within the allowable ranges.

Appraised by Mieczysław Hann, Assoc.Prof.,D.Sc.,M.E.

NOMENCLATURE

DSM	- Design Sea Method	R_e	- yield point of the material
GL	- Germanischer Lloyd	S	- force in the luffing rope system
LR	- Lloyd's Register	T_R	- rosette strain gauge
M	- moment	T_S	- single strain gauge
M_0	- maximum total moment	x,y,z	- Cartesian coordinates (see Fig.1.)
M_x,M_y,M_z	- moments round the x,y,z axis	σ	- normal stress
n	- compound pulley reduction ratio	σ_{zr}	- reduced stress
P	- force in the rope	τ	- tangent stress
P_t,M_t	- crane loading during tests		
P_x,P_y,P_z	- forces along the x,y,z axis		
r	- notch radius		

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