REPORT No. 208 S
(SC/320)

September 1976

NEDERLANDS SCHEEPSSTUDIECENTRUM TNO

NETHERLANDS SHIP RESEARCH CENTRE TNO
SHIPBUILDING DEPARTMENT LEEGHWATERSTRAAT 5, DELFT

★

NUMERICAL VIBRATION ANALYSIS OF THE DECKHOUSE OF A FAR EAST CONTAINER SHIP

(NUMERIEK TRILLINGSONDERZOEK AAN EEN DEKHUIS VAN EEN DERDE GENERATIE CONTAINER SCHIP)

by

Ir. G. T. M. JANSSEN
(Institute TNO for Mechanical Constructions)

Issued by the Council

476

VOORWOORD

Een lastige complicatie bij het voorkomen van trillingshinder aan boord van schepen, is het eigen trillingsgedrag van grote subconstructies. Algemeen bekend zijn in dit verband ondermeer excessief trillingsgedrag van de dubbele bodem tussen twee waterdichte schotten, het achterschip, grote dekhuizen en dergelijke subconstructies.

Tengevolge van de toenemende afmetingen van zeeschepen zijn de bovenbedoelde constructies minder stijf en dus ook gevoeliger voor trillingen in het aan boord aangeboden frequentie gebied. Bovendien hebben de hogere geïnstalleerde vermogens tot gevolg dat meer excitatie energie aanwezig is.

In het geval van dekhuizen leveren de relatief inhomogene constructie en de variatie in de wijze van bevestiging van het dekhuis aan de scheepsconstructie extra problemen. Het is niet ongewoon dat een dekhuis een dertigtal eigen frequenties heeft die elk aanleiding kunnen geven tot een vorm van resonantie in een frequentie gebied van ca. 10 tot 60 Hz.

Sinds 1969 is op voorstel van Nederlandse reders en werven het NSS gevraagd veel aandacht te besteden aan oplossingen voor dit probleem. Aan boord van diverse typen schepen zijn met behulp van een excitator dekhuis trillingsmetingen verricht en de resultaten in samenhang met de constructie eigenschappen geanalyseerd. Daarnaast is een rekenmodel ontwikkeld dat zo goed mogelijk het gemeten trillingsgedrag kon reproduceren.

Met het aldus ontwikkelde rekenmodel is het trillingsgedrag van een dekhuis van een derde generatie containerschip berekend. Waar nodig is het ontwerp op grond van deze prognose gemodificeerd. De resultaten zijn samengevat in Report no. 208 S "Numerical vibration analysis of the deckhouse of a Far-East containership". Teneinde de juistheid van de betreffende rekenmethode finaal te toetsen zijn uitgebreide excitator geexciteerde trillingsmetingen aan boord van het betreffende schip ss. Nedlloyd-Delft, reeds bij de aanvang van het research project voorbereid. Deze metingen zijn inmiddels uitgevoerd. Een vergelijking van berekende en gemeten resultaten wordt binnenkort gepubliceerd.

Veel dank is verschuldigd aan de "Nederlandse Scheepvaart Unie BV" voor de medewerking die zij heeft willen verlenen bij het tot stand komen en bij de uitvoering van dit research project.

NEDERLANDS SCHEEPSSTUDIECENTRUM TNO

PREFACE

The prevention of "vibration annoyance" in ships is a complicated problem due to the behaviour of large mass sub-structures.

It is generally known that excessive vibrations occur in the double bottoms between two bulkheads, the aft ship structure, large deckhouses and comparable sub-structures.

In view of the increasing size of vessels, sub-structures are relatively less stiff and thus more sensitive to vibrations in the excited frequency range.

In the case of deckhouses, extra problems are created with the heterogeneous structure and the influence of the support on the ships structure. It is not uncommon for a deckhouse to have up to thirty natural *frequencies* each of which can give cause to resonance in a frequency range of approximately 10 to 60 Hz.

In 1969 Dutch shipowners and shipyards made a joint proposal to the Netherlands Ship Research Centre to find a solution of the subject problem. Consequently experiments were carried out on ships with different types of deckhouses. Excitator induced vibrations were measured and the results were exhaustively analysed in relation with the structural characteristics.

In addition a mathematical model was developed which could best describe the measured vibration behaviour.

The model was then used to calculate the deckhouse vibration characteristics of a third generation container ship and where necessary the true design was adjusted. The results are summarized in report no. 208 S "Numerical vibration analysis of the deckhouse of a far east container ship."

In order to verify the subject calculation model extensive excitator induced vibration measurements were put on board the ss "Nedlloyd-Delft" at the commencement of the research project. These results will be published in the very near future.

The co-operation of the "Nederlandse Scheepvaart Unie BV" in this research project, is gratefully acknowledged.

THE NETHERLANDS SHIP RESEARCH CENTRE TNO

NUMERICAL VIBRATION ANALYSIS OF THE DECKHOUSE OF A FAR EAST CONTAINER SHIP

by

Ir. G. T. M. JANSSEN

Summary

The lower natural frequencies and the corresponding vibration modes of the deckhouse of a Far East containership have been determined by numerical methods. The deckhouse was assumed to be fixed to a rigid support. In order to investigate the usefulness of a comprehensive analysis coupling the deckhouse with the main ship structure (for which all data were available), an additional testcase with modified deckhouse support conditions has been evaluated.

1 Introduction

The stiffness distribution of the deckhouse of a Far East containership (Fig. 1) has been determined by a finite element analysis. To cut down computational expenses without sacrificing much accuracy for the lower natural frequencies, the complex deckhouse structure has been replaced by a simple model composed of rectangular orthotropic membrane elements and edge stiffeners [1, 2]. The effects of plate stiffening and local structural discontinuities have been taken into account by adjustments of the elastic constants of the plates which actually have isotropic properties.

For the determination of the mass matrix nearly all the distributed masses are represented by concentrated masses directly coupled with the chosen degrees of freedom. Only for two parts on top of the deckhouse, viz. the funnel and the radar antenna mast, the mass concentrated at the centre of gravity and also the moments of inertia have been taken into account. After determination of the complete stiffness and the mass matrices, the system has been condensed to an equivalent smaller one. The lower natural frequencies and vibration modes were computed by standard eigenvalue programmes.

Initially the analysis has been carried out under the assumption that the shiphull is very stiff with respect to the deckhouse so that a good approximation of the local frequencies of the deckhouse, rigidly supported, could be obtained.

Since the appropriate stiffness and mass matrices for the ship were available from Van Beek's stress analysis [3] and from Meijers' hull vibration analysis [4], a comprehensive vibration analysis coupling the deckhouse with the main shipstructure would be possible.

To get an impression of the usefulness carrying out such a coupled deckhouse-ship analysis a trial case with modified support conditions has been analyzed. For this latter case some of the maindeck degrees of freedom used in the hull vibration analysis, have been suppressed, and the vertical displacements at intermediate base points remained free.

The deckhouse structure [7] itself has been represented by the model given in Fig. 2.

2 Analysis

The containership general arrangement showing the position of the superstructure is given in Fig. 1. The geometry of the deckhouse according to [7] has been simplified (Fig. 2). Ignoring some small unsignificant details, the structure is symmetric about the x–y-plane giving the opportunity of reducing the required data input and computational effort considerably to obtain natural frequencies and their symmetric and anti-symmetric vibration modes.

The thicknesses of the orthotropic plates in the simplified structure have been kept the same as those of the actual structure. Local stiffening and weakening have been distributed and incorporated in the effective elastic constants. These constants are defined by the stress-strain relations:

$$\begin{vmatrix} \varepsilon_x \\ \\ \varepsilon_y \\ \\ \gamma_{xy} \end{vmatrix} = \begin{vmatrix} \dfrac{1}{E_1} & -\dfrac{\nu_{21}}{E_2} & 0 \\ -\dfrac{\nu_{12}}{E_1} & \dfrac{1}{E_2} & 0 \\ 0 & 0 & \dfrac{1}{G_{12}} \end{vmatrix} \begin{vmatrix} \sigma_x \\ \\ \sigma_y \\ \\ \tau_{xy} \end{vmatrix}$$

where

$$\frac{\nu_{12}}{E_1} = \frac{\nu_{21}}{E_2}$$

Stiffening may be caused by the presence of stiffeners which are regularly spaced in the plate field. Often the stiffeners are placed in one-direction. For the relation between the cross-section of such stiffeners and the resulting corresponding elastic constants refer to [1, 2].

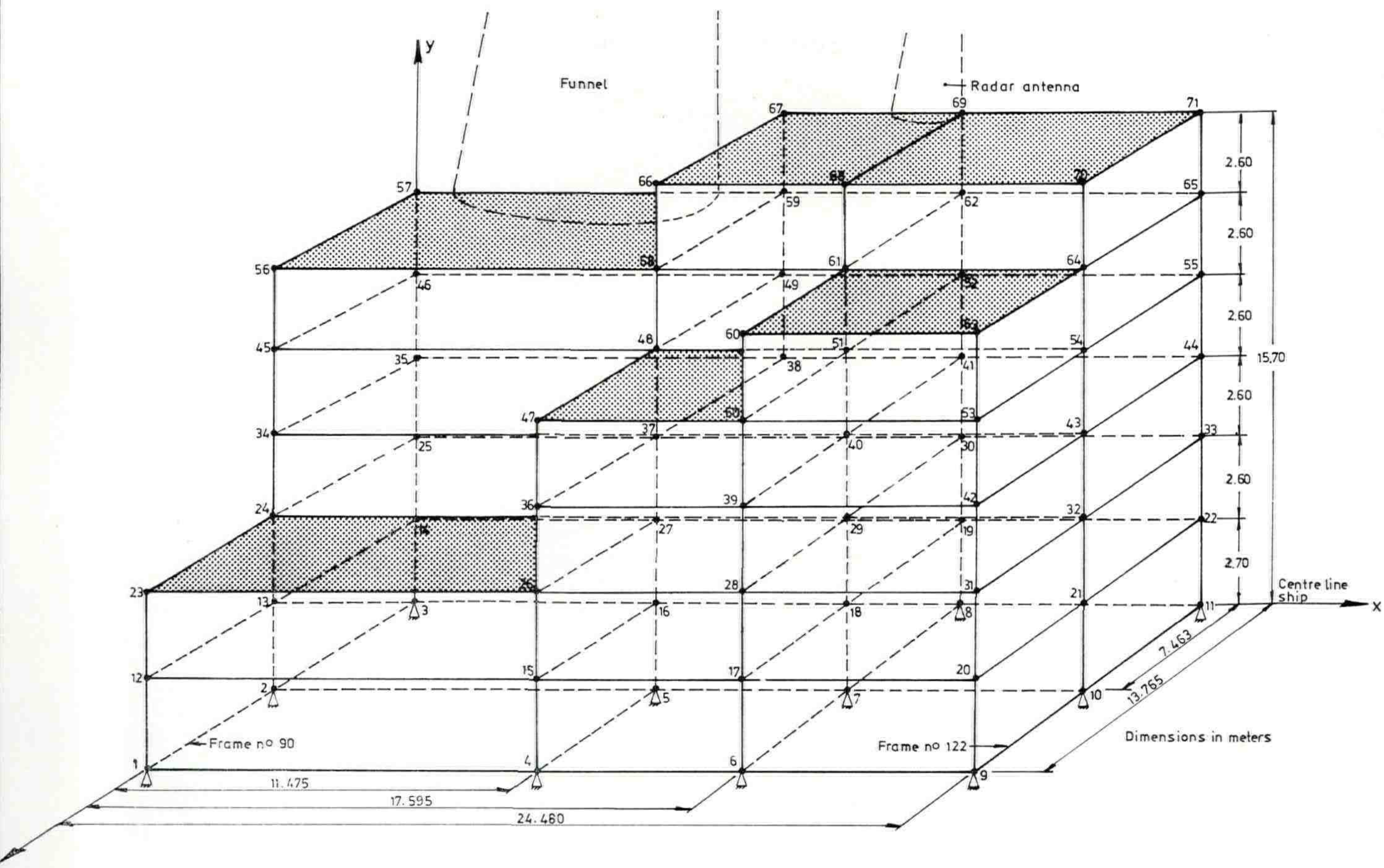


Fig. 2. Simplified deckhouse.

The stiffness of intermediate bulkheads which do not coincide with plate elements in the simplified structure has been added to the stiffness of the nearest membrane elements.

At some locations in the deckhouse an important stiffness reduction in vertical direction is due to the fact that the transverse bulkheads on the different levels of the deckhouse are not placed in line with each other. This reduction has been estimated as accurately as possible and has been accounted for by reducing the effective Young's modulus in vertical direction.

The fictitious elastic constants and the plate thicknesses of the orthotropic plate-elements are given in Tables I–III (Appendix). The cross-section of the stiffener elements are presented in Table IV (Appendix).

Table V (Appendix) shows the concentrated masses at the nodal points. These masses are statically equivalent with the distributed mass of the structure. The funnel and the radar antenna are two heavy items on top of the deckhouse. Their masses concentrated at their centres of gravity and their moments of inertia (Table VI Appendix) have been coupled kinematically with the neighbouring degrees of freedom.

The structural stiffness matrix has been generated using the ASKA software package for structural analysis. Subsequently the large stiffness and mass matrices representing the system of the concentrated masses have been condensed to smaller matrices related to so-called master degrees of freedom which are suitably chosen.

A condensation procedure is justified because it is known that if only the lower natural frequencies are required the system may be represented by a system with far less degrees of freedom than is required to determine the stiffness distribution with sufficient accuracy, and the condensation combined with an eigenvalue analysis for a smaller system is from the point of computer cost more attractive than a direct eigenvalue analysis for the large system (Table 1).

The lower natural frequencies and the corresponding

Table 1. Model characteristics

vibration mode	symmetric vibrations	anti-symmetric vibrations	symmetric vibrations
support condition	rigidly clamped	rigidly clamped	restricted clamping
number of nodal points	71	71	71
elements	123	123	130
degrees of freedom	157	138	175
master degrees	59	57	50

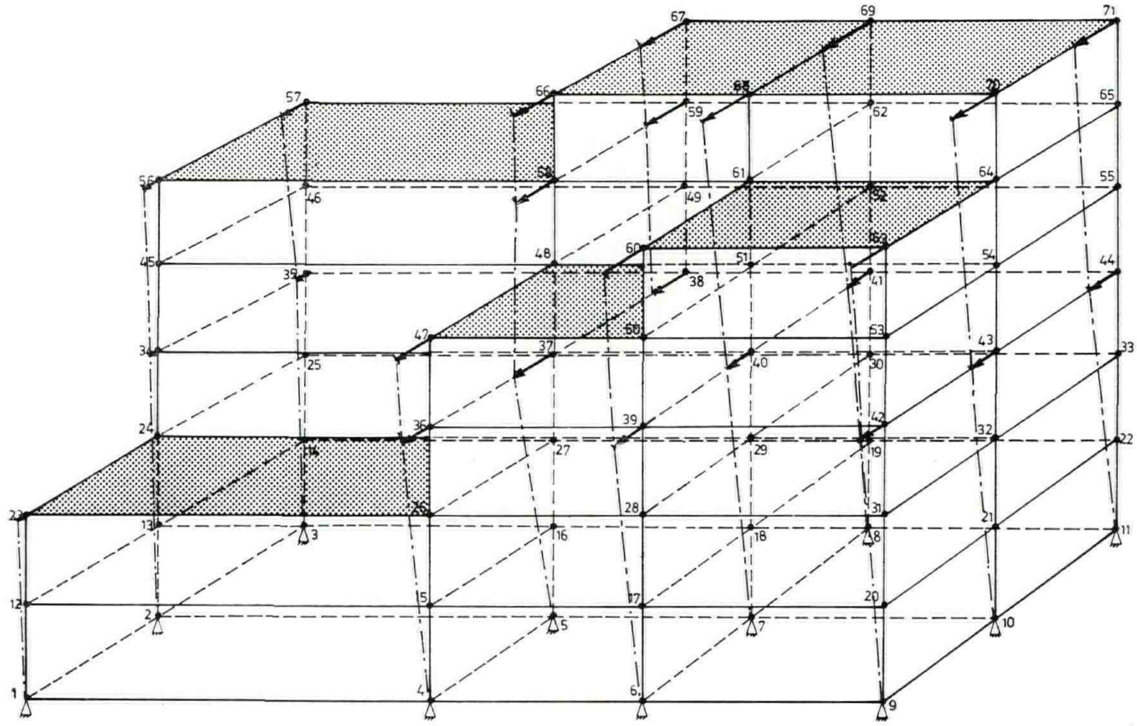


Fig. 5. First anti-symmetric vibration mode ($f_1 = 15.3$ Hz).

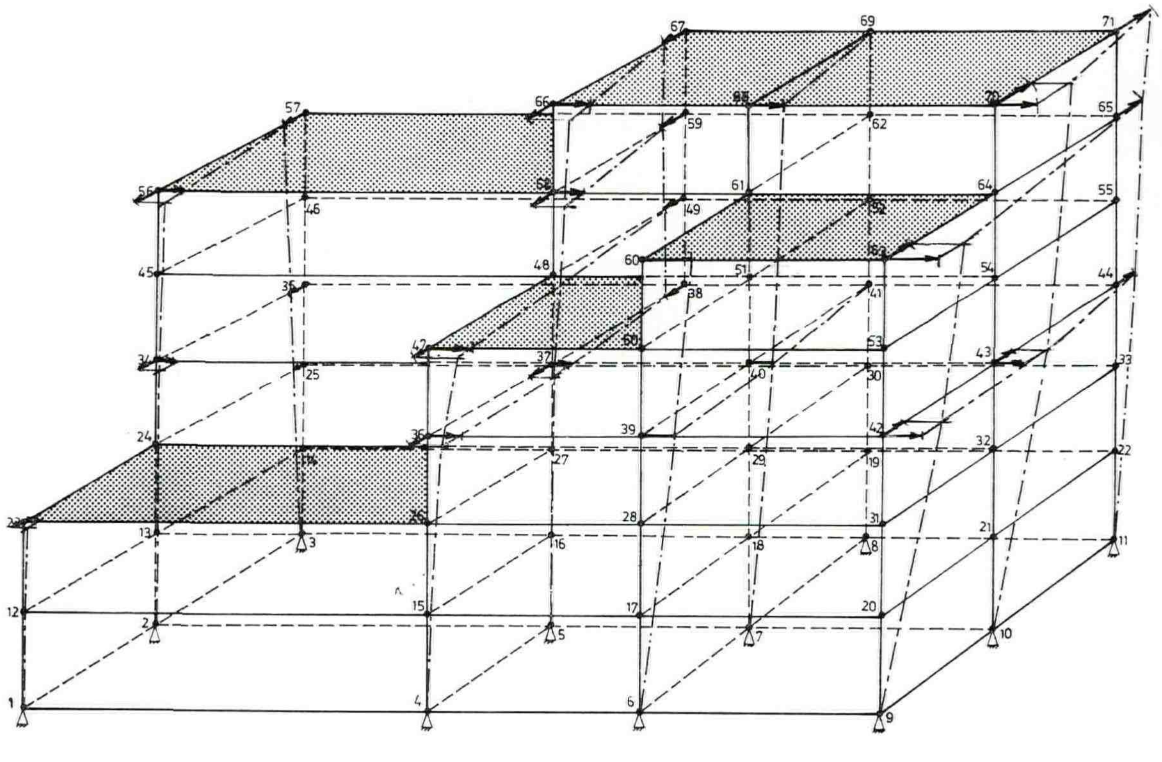


Fig. 6. Second anti-symmetric vibration mode ($f_2 = 21.0$ Hz).

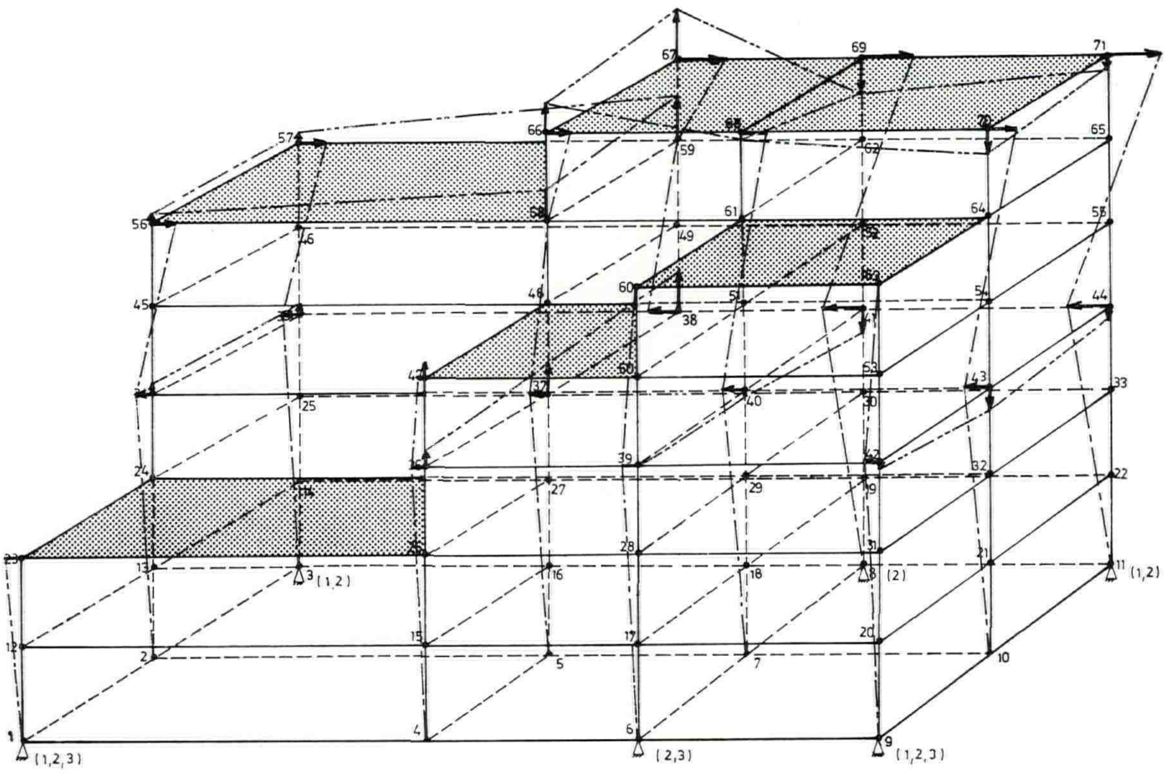


Fig. 9. Third symmetric vibration mode ($f_3 = 25.7$ Hz). Testcase for coupling.

vibration modes for both symmetric and anti-symmetric vibrations have been calculated with the Dynan [6] software package for dynamic analysis.

As mentioned in the introduction the analysis has been carried out for two different support conditions, viz. a. a fully fixed condition and b. a condition with only the degrees of freedom from the hull vibration analysis suppressed. Keeping remaining vertical connections detached. The inplane stiffness of the maindeck was simulated by a plate of thickness equal to the thickness of the maindeck. For the vertical degrees of freedom which are only significant since the inplane stiffness of the maindeck will always be very high, this is the best possible coupling with the grid of freedoms of the hull.

Should it turn out that the reduction in the frequencies for this case compared with the case of the fully fixed support is significant, a comprehensive coupled analysis of deckhouse and entire ship structure utilizing a locally refined grid system of the original hull model, from which the mass- and stiffness-matrix are available, might be necessary.

The analysis for the case with partial attachment of the deckhouse to the maindeck has only been carried out for symmetric vibrations.

3 Numerical results

The characteristics of the three cases analyzed are indicated in Table 1. For all cases the 15 lower natural frequencies (Fig. 10) and the corresponding vibration modes have been determined. The symmetric and anti-symmetric vibration modes corresponding to the lower two frequencies and for the test case corresponding to the lower three frequencies are shown in Figs. 3–9.

The detaching of the intermediate base points in vertical direction causes a 17% reduction of the primary natural frequency. The second vibration mode is now mainly a vertical vibration mode which will in case of the fully fixed support condition occur at a much higher frequency.

4 Conclusions

The vibration mode of the lowest natural frequency (14.4 Hz) is a symmetric vibration in the longitudinal direction of the ship.

The vibration mode of the second natural frequency (15.3 Hz) is anti-symmetric and is mainly a vibration in the transverse direction of the ship.

The third frequency (21.0 Hz) belongs to an anti-symmetric vibration mode which is mainly a torsional vibration.

The second symmetric vibration mode (27.6 Hz) shows translational motions in longitudinal as well as in vertical direction.

The test case, introduced to examine the usefulness of a coupled analysis, shows that the natural frequencies decreased significantly by the uncoupling of the vertical displacements at the deleted intermediate connections. It is concluded that a coupled analysis applying the grid at the deckhouse-ship main structure interconnection as was previously used for the hull vibrations study [4] will not give realistic results.

In case a comprehensive coupled analysis has to be carried out, it appears to be necessary to condens the stiffness and mass matrices of the ship main structure in such a way that a larger number of degrees of freedom is chosen at the intersection of the deckhouse and the main ship structure.

Since the stiffness of the deckhouse supporting structure is high in comparison with that of the deckhouse itself, it is expected that the results using the fixed support will be reasonably accurate for the lower frequencies. However, one must be carefull applying such a conclusion to situations on other ships.

References

1. CATE, W. TEN and P. MEIJERS, Berekening van eigenfrequenties en trillingsvormen van de bovenbouw van een schip. (Calculation of the natural frequencies and vibration modes of the superstructure of a ship). TNO-IWECO, 4671/1, April 1970.
2. MEIJERS, P., W. TEN CATE, L. J. WEVERS and J. H. VINK, Numerical and experimental vibration analysis of a deckhouse. Netherlands Ship Research Centre TNO, Report No. 184 S, December 1973.
3. BEEK, A. W. VAN, Finite element analysis of a third generation containership. Netherlands Ship Research Centre TNO, Report No. 182 S, December 1973.
4. MEIJERS, P., Numerical hull vibration analysis of a Far East container ship. Netherlands Ship Research Centre TNO, Report No. 195 S, July 1974.
5. ASKA User's Reference Manual, ISD-Report No. 73. Stuttgart, 1971.
6. DYNAN User's Reference Manual. ISD-Report No. 97. Stuttgart 1971.
7. BREMER VULCAN. Shipyard and Engineering Works. Drawings No. SFZ 305-315 and SFZ 701-707, Bremen, 1972.

APPENDIX

Table I. Elastic constants and plate thickness of rectangular elements*

				elements parallel to x–y plane						
no.	nodal point numbers				t (mm)	E_1/E	E_2/E	G_{12}/G	ν_{12}/ν	ν_{21}/ν
1	2	13	16	5	10	0.800	1.218	1.106	1.000	
2	8	19	22	11	4	0	0.655	0.575	0	0.655
3	5	16	18	7	10	0.712	1.105	1.262	0.890	
4	7	18	21	10	10	0.315	0.957	0.910	0.393	
5	4	1	12	15	10	1.380	1.212	0.929	0.910	
6	6	4	15	17	10	1.361	0.995	0.962	0.952	
7	9	16	17	20	10	1.185	0.995	0.950	0.939	
8	22	19	30	33	3.5	0	0.795	0.667	0	1.000
9	16	13	24	27	10	0.892	0.100	1.262	1.000	
10	18	16	27	29	10	0.601	0.920	1.426	0.752	
11	15	12	23	26	10	0.506	0.685	0.547	0.695	
12	21	18	29	32	7	0.402	0.862	0.803	0.402	
13	17	15	26	28	10	0.907	0.928	0.870	0.977	
14	20	17	28	31	10	0.914	0.933	0.883	0.980	
15	27	24	34	37	10	1.005	1.398	1.076	0.715	
16	33	30	41	44	10	0	0.239	0.212	0	0.239
17	29	27	37	40	10	0.725	0.823	0.788	0.763	
18	32	29	40	43	10	0	0.434	0.340	0	0.434
19	28	26	36	39	10	0.685	0.728	0.689	0.941	
20	31	28	39	42	10	0.899	0.895	0.829	1.004	
21	38	35	46	49	10	0.511	0.684	0.488	0.511	
22	41	38	49	52	10	0.423	0.576	0.504	0.792	
23	37	34	45	48	10	0.932	1.186	0.879	0.932	
24	40	37	48	51	10	0.792	1.096	0.907	0.792	
25	39	36	47	50	9	0.936	0.663	0.629	0.887	
26	43	40	51	54	6	0.636	1.214	0.536	0.636	
27	42	39	50	53	9	0.941	0.947	0.879	0.992	
28	49	46	57	59	10	0.361	0.371	0.221	0.361	
29	48	45	56	58	10	0.530	0.595	0.266	0.530	
30	52	49	59	62	9	0.167	0.722	0.670	0.722	
31	51	48	58	61	9	0.542	1.033	0.946	0.542	
32	54	51	61	64	6	0.381	0.573	0.444	0.540	
33	53	50	60	63	9	0.939	0.947	0.879	0.992	
34	62	59	67	70	8.5	0.380	0.445	0.357	0.380	
35	61	58	66	68	8.5	0.577	0.675	0.543	0.577	
36	64	61	68	70	8.5	0.675	0.861	0.584	0.675	

* Index 1 corresponds with x-direction and index 2 with y-direction.

Table IV. Cross-section of bar elements (stiffeners)

no.	nodal points bar elements		A (10^2 mm)	no.	nodal points bar elements		A (10^2 mm)
1	12	15	191.3	15	46	49	23.8
2	23	24	50.0	16	49	52	33.8
3	24	25	115.6	17	52	55	33.8
4	24	27	138.8	18	45	48	36.2
5	27	29	132.0	19	47	50	107.3
6	29	30	123.0	20	45	46	56.0
7	34	35	162.3	21	62	65	28.0
8	37	38	141.7	22	68	70	120.0
9	40	41	14.2	23	70	71	21.6
10	35	38	31.3	24	3	14	49.8
11	38	41	30.0	25	14	25	49.8
12	41	44	28.0	26	7	18	632.0
13	34	37	50.4	27	25	35	28.0
14	36	37	162.3				

Table V. Concentrated mass at nodal points* (half deckhouse)

nodal point	mass (kg)	nodal point	mass (kg)	nodal point	mass (kg)	nodal point	mass (kg)
1	3408	23	6190	45	4247	67	3563
2	7848	24	7962	46	2357	68	4445
3	2159	25	1489	47	1955	69	5519
4	5262	26	11158	48	5498	70	5275
5	10167	27	17130	49	2909	71	3973
6	5349	28	9351	50	8037		
7	5073	29	15973	51	16363		
8	2264	30	6823	52	7970		
9	2955	31	5498	53	4542		
10	5850	32	10294	54	10255		
11	3056	33	5592	55	5438		
12	7711	34	5028	56	4049		
13	7370	35	1640	57	2567		
14	2315	36	3034	58	5358		
15	13927	37	12058	59	6024		
16	14299	38	833	60	3493		
17	10547	39	9293	61	10374		
18	16496	40	17682	62	9275		
19	6080	41	8128	63	3156		
20	5767	42	5472	64	9390		
21	11908	43	10151	65	5352		
22	6345	44	5468	66	405		

* Total mass is 494,292 kg (half deckhouse).

Table VI. Mass and moment of inertia of the funnel and radar antenna (half structures)

	m (kg)	I_x (kgm^2)	I_y (kgm^2)	I_z (kgm^2)
funnel	7,500	97,100	62,400	116,150
radar antenna	2,500	9,710	435	10,250

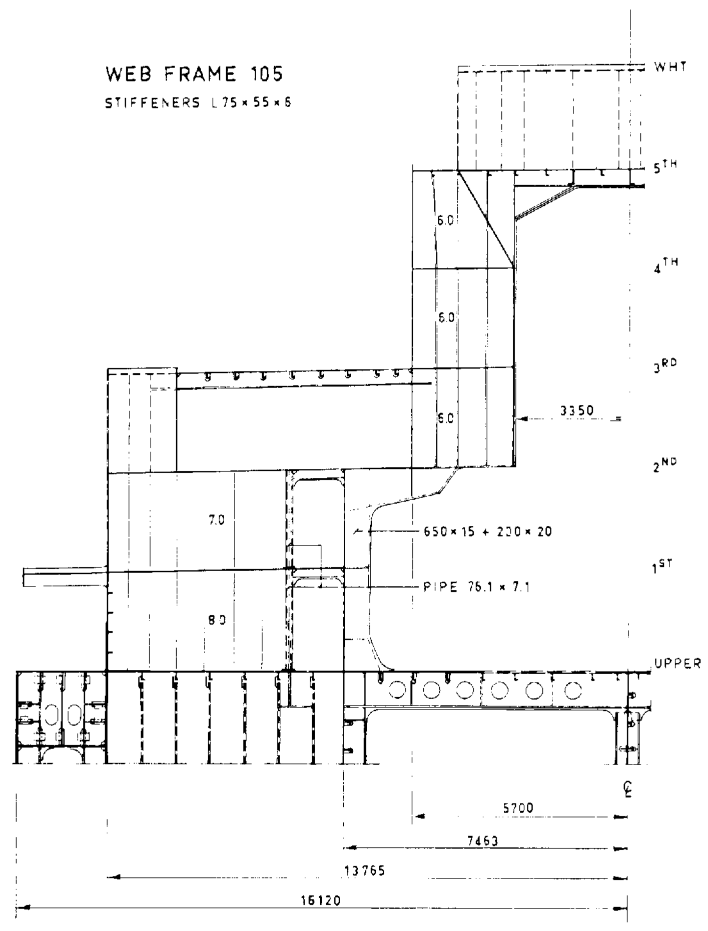
177 M Maritime transportation of containerized cargo. Part III. Fire tests in closed containers. H. J. Souer, 1973.
178 S Fracture mechanics and fracture control for ships. J. J. W. Nibbering, 1973.
179 S Effect of forward draught variation on performance of full ships. M. F. van Sluijs and C. Flokstra, 1973.
180 S Roll damping by free surface tanks with partially raised bottom. J. J. van den Bosch and A. P. de Zwaan, 1974.
182 S Finite element analysis of a third generation containership. A. W. van Beek, 1973.
183 M Marine diesel engine exhaust noise. Part VII. Calculation of the acoustical performance of diesel engine exhaust systems. J. Buiten, E. Gerretsen and J. C. Vellekoop, 1974.
184 S Numerical and experimental vibration analysis of a deckhouse. P. Meijers, W. ten Cate, L. J. Wevers and J. H. Vink, 1973.
185 S Full scale measurements and predicted seakeeping performance of the containership "Atlantic Crown". W. Beukelman and M. Buitenhek, 1973.
186 S Waves induced motions and drift forces on a floating structure. R. Wahab, 1973.
187 M Economical and technical aspects of shipboard reliquefaction of cargo "Boil-off" for LNG carriers. J. A. Knobbout, 1974.
188 S The behaviour of a ship in head waves at restricted water depths. J. P. Hooft, 1974
189 M Marine diesel engine exhaust noise. Part VIII. A revised mathematical model for calculating the acoustical source strength of the combination diesel engine – exhaust turbine. P. J. Brandenburg, 1974.
190 M Condition monitoring, trend analysis and maintenance prediction for ship's machinery (literature survey). W. de Jong, 1974.
191 S Further analysis of wave-induced vibratory ship hull bending moments. F. F. van Gunsteren, 1974.
192 S Hull resonance no explanation of excessive vibrations. S. Hylarides, 1974.
193 S Wave induced motions and loads on ships in oblique waves. R. Wahab and J. H. Vink, 1974.
194 M On the potentialities of polyphenylene oxide (PPO) as a wet-insulation material for cargo tanks of LNG-carriers. G. Opschoor, 1974.
195 S Numerical hull vibration analysis of a Far East container ship. P. Meijers, 1974.
196 S Comparative tests of four fast motor boat models – in calm water and in irregular head waves and an attempt to obtain full-scale confirmation. J. J. van den Bosch, 1974.
197 M Transverse vibrations of ship's propulsion systems. Part I. Theoretical analysis. S. Hylarides, 1974.
198 M Maritime transportation of containerized cargo. Part IV. Evaluation of the quality loss of tropical products due to moisture during seatransport. P. J. Verhoef, 1974.
199 S Acoustical effects of mechanical short-circuits between a floating floor and a steel deck. J. Buiten and J. W. Verheij, 1974.
200 M Corrosivity monitoring of crankcase lubricating oils for marine diesel engines. L. M. Rientsma and H. Zeilmaker, 1974.
201 S Progress and developments of ocean weather routeing. C. de Wit, 1974.
202 M Maritime transportation of containerized cargo. Part V. Fire tests in a closed aluminium container. H. J. Souer, 1974.
203 M Transverse vibrations of ship's propulsion systems. Part II. Experimental analysis. L. J. Wevers, 1974.
204 S Full-scale measurements of stresses in the bulkcarrier m.v. 'Ossendrecht'. Part II. The spectral analysis, the distribution of amplitudes, the distribution of heights, the χ-test. F. X. P. Soejadi, 1975.
206 S Synthesis of cooperative fatigue investigations with notched plates and welded ship structures of St 42 and St 52. J. J. W. Nibbering, H. G. Scholte and J. van Lint, 1974.
208 S Numerical vibration analysis of the deckhouse of a far east container ship. G. T. M. Janssen, 1976.
210 S Stress analysis of detail structures of a third generation containership. W. ten Cate and A. W. van Beek, 1975.
215 S Some problems related to the use of scale models in experiments on ship acoustics. J. W. Verheij, 1975.
219 M Marine refrigeration engineering. Part VIII. Investigations into the distribution of cargo temperatures in a loaded full-size model of a ship's reefer space. P. J. Verhoef, 1975.
220 M A comparative laboratory investigation of materials for marine diesel engine fuel nozzles. J. W. Tichler and A. W. J. de Gee, 1975.
221 M Experimental work on diesel engine exhaust piping as a noise source in ship's accommodation. J. W. Verheij, 1975.

Communications (Mededelingen)

18 S An experimental simulator for the manoeuvring of surface ships. J. B. van den Brug and W. A. Wagenaar, 1969.
19 S The computer programmes system and the NALS language for numerical control for shipbuilding. H. le Grand, 1969.
20 S A case study on networkplanning in shipbuilding (Dutch). J. S. Folkers, H. J. de Ruiter, A. W. Ruys, 1970.
21 S The effect of a contracted time-scale on the learning ability for manoeuvring of large ships (Dutch). C. L. Truijens, W. A. Wagenaar, W. R. van Wijk, 1970.
22 M An improved stern gear arrangement. C. Kapsenberg, 1970.
23 M Marine refrigeration engineering. Part V (Dutch). A. H. van der Tak, 1970.
24 M Marine refrigeration engineering. Part VI (Dutch). P. J. G. Goris and A. H. van der Tak, 1970.
25 S A second case study on the application of networks for productionplanning in shipbuilding (Dutch). H. J. de Ruiter, H. Aartsen, W. G. Stapper and W. F. V. Vrisou van Eck, 1971.
26 S On optimum propellers with a duct of finite length. Part II. C. A. Slijper and J. A. Sparenberg, 1971.
27 S Finite element and experimental stress analysis of models of shipdecks, provided with large openings (Dutch). A. W. van Beek and J. Stapel, 1972.
28 S Auxiliary equipment as a compensation for the effect of course instability on the performance of helmsmen. W. A. Wagenaar, P. J. Paymans, G. M. A. Brummer, W. R. van Wijk and C. C. Glansdorp, 1972.
29 S The equilibrium drift and rudder angles of a hopper dredger with a single suction pipe. C. B. van de Voorde, 1972.
30 S A third case study on the application of networks for productionplanning in shipbuilding (Dutch). H. J. de Ruiter and C. F. Heijnen, 1973.
31 S Some experiments on one-side welding with various backing materials. Part I. Manual metal arc welding with coated electrodes and semi-automatic gas shielded arc welding (Dutch). J. M. Vink, 1973.
32 S The application of computers aboard ships. Review of the state of the art and possible future developments (Dutch). G. J. Hogewind and R. Wahab, 1973.
33 S FRODO, a computerprogram for resource allocation in networkplanning (Dutch). H. E. I. Bodewes, 1973.
34 S Bridge design on dutch merchant vessels; an ergonomic study. Part I: A summary of ergonomic points of view (Dutch). A. Lazet, H. Schuffel, J. Moraal, H. J. Leebeek and H. van Dam, 1973.
35 S Bridge design on dutch merchant vessels; an ergonomic study. Part II: First results of a questionnaire completed by captains, navigating officers and pilots. J. Moraal, H. Schuffel and A. Lazet, 1973.
36 S Bridge design on dutch merchant vessels; an ergonomic study. Part III: Observations and preliminary recommendations. A. Lazet, H. Schuffel, J. Moraal, H. J. Leebeek and H. van Dam, 1973.
37 S Application of finite element method for the detailed analysis of hatch corner stresses (Dutch), J. H. Vink, 1973.
38 S A computerprogram for displacement and stress analysis with membrane elements on constructions consisting of plates and trusses. User's manual (Dutch). G. Hommel and J. H. Vink, 1974.
39 S Some experiments on one-side welding with various backing materials. Part II. Mechanised gas-shielded arc welding in the flat and horizontal position (Dutch). J. M. Vink, 1974.
40 S Bridge design on dutch merchant vessels; an ergonomic study. Part IV: Evaluation of standards and recommendations by means of a static mock-up. J. Moraal, H. Schuffel, H. J. Leebeek and A. Lazet, 1974.
41 S Some experiments on one-side welding with various backing materials. Part III. Mechanised welding with flux cored wire and braided wire, horizontal position (Dutch). J. M. Vink, 1975.

WEB FRAME 105

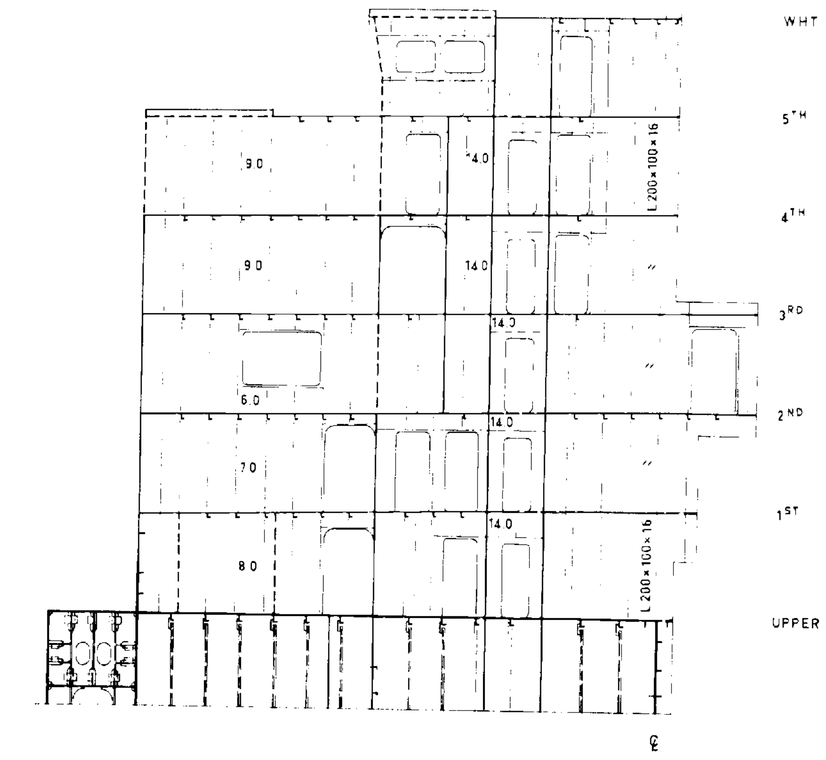
STIFFENERS L 75 × 55 × 6



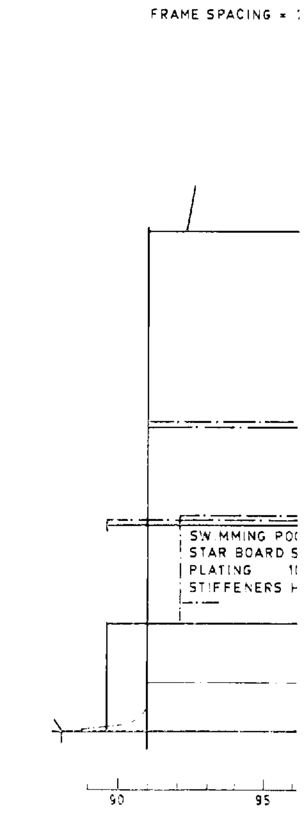
MACHINERY CASING			
	FORE END	SIDES	AFTER END
PLATING	6.0	9.0	9.0
STIFFENERS	L 100 × 65 × 7	L 100 × 65 × 7	L 100 × 65 × 7
PLATING	6.0	9.0	9.0
STIFFENERS	L 100 × 65 × 7	L 100 × 65 × 7	L 100 × 65 × 7
PLATING	6.0	9.5	9.5
STIFFENERS	L 100 × 65 × 7	L 100 × 75 × 8	L 100 × 75 × 8
PLATING	7.0	8.0	9.5
STIFFENERS	L 100 × 65 × 7	L 100 × 65 × 7	L 100 × 75 × 8
PLATING	8.0	8.0	9.5
STIFFENERS	L 100 × 65 × 7	L 100 × 65 × 7	L 100 × 75 × 8

WEB FRAME 113 (BHD)

STIFFENERS L 100 × 55 × 7

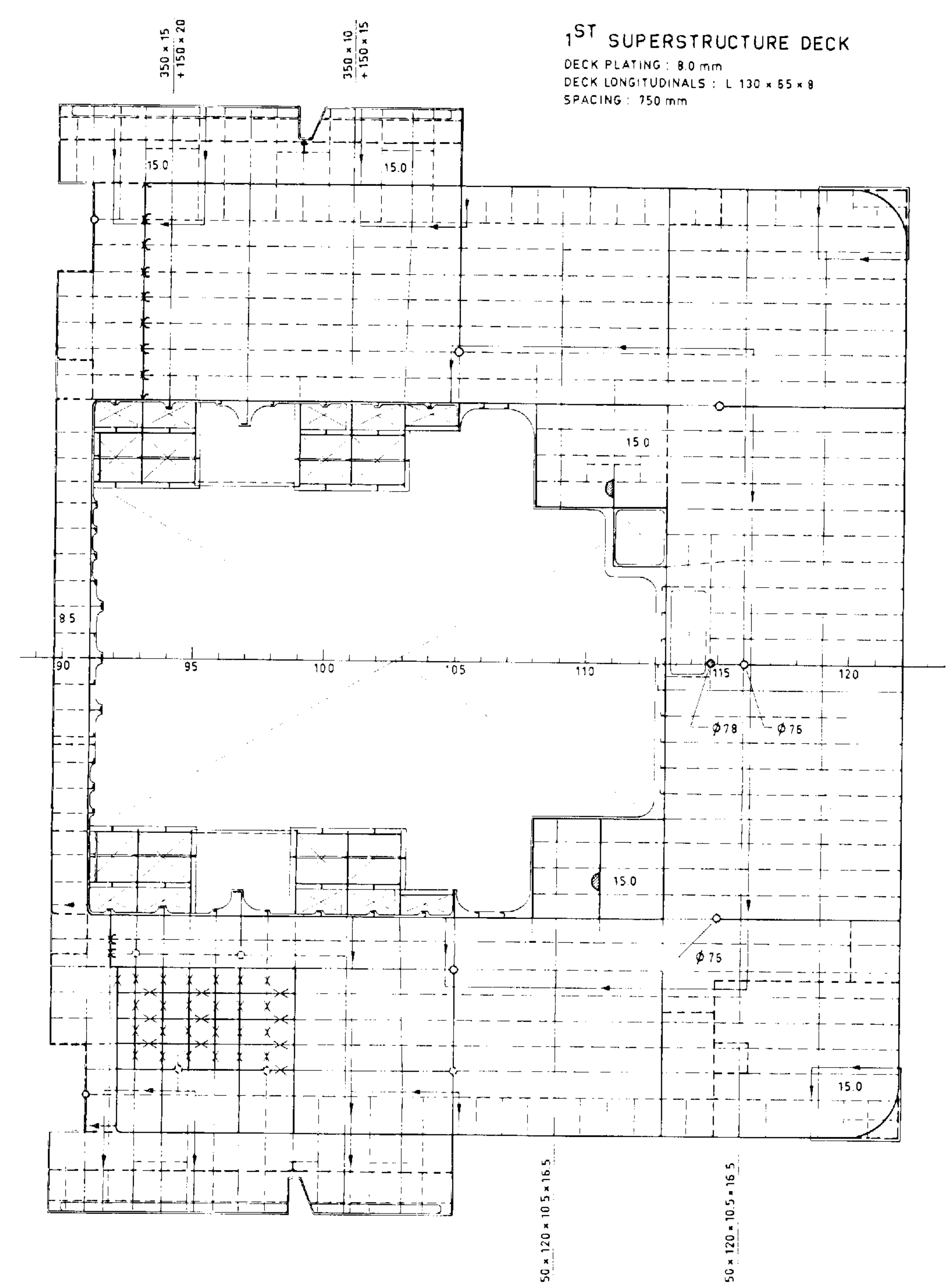


LONGITUDINAL



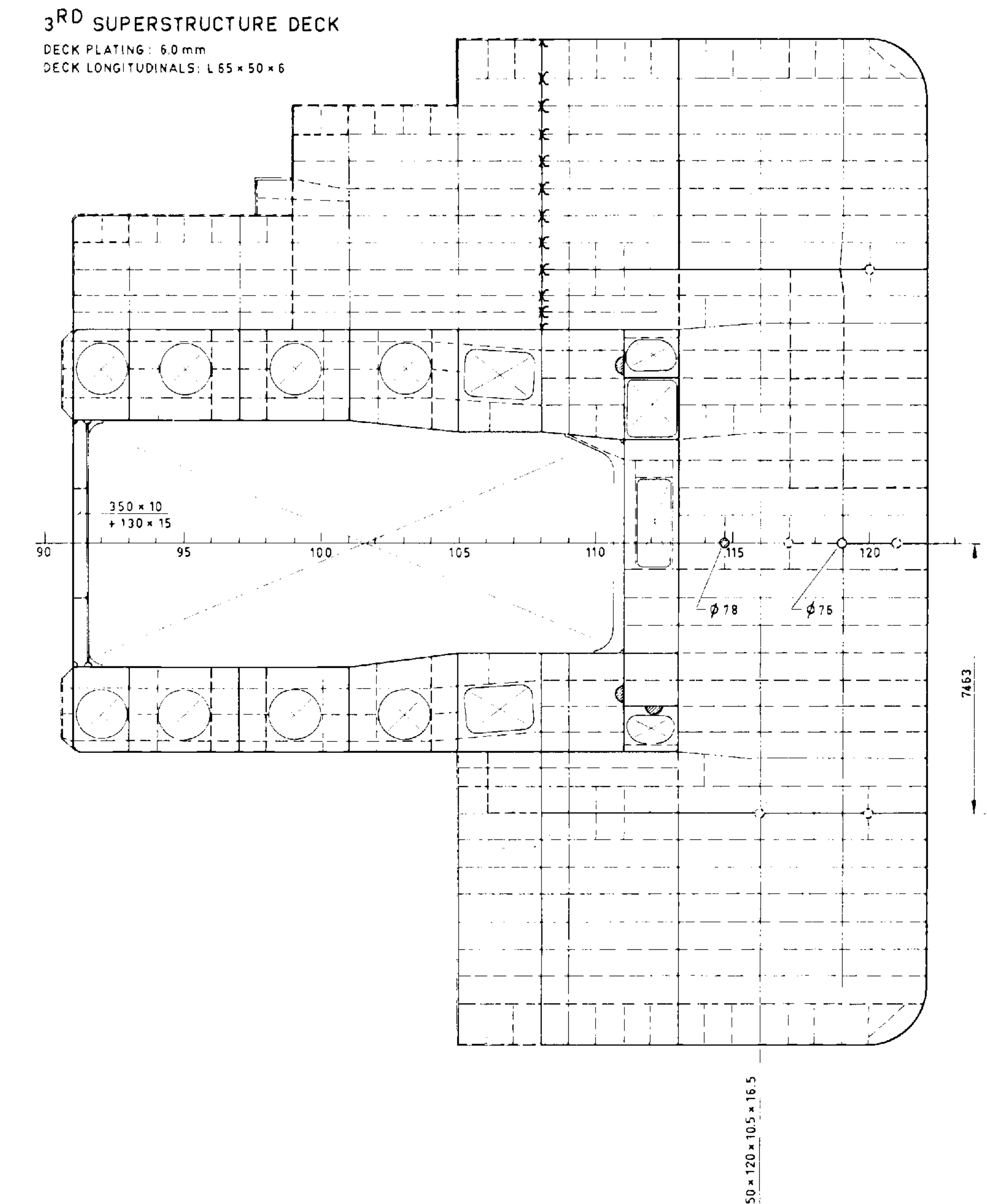
1ST SUPERSTRUCTURE DECK

DECK PLATING : 8.0 mm
DECK LONGITUDINALS : L 130 × 65 × 8
SPACING : 750 mm



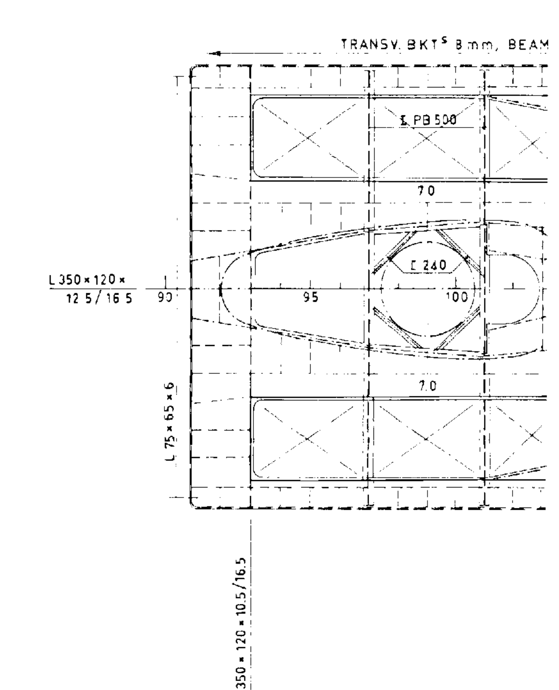
3RD SUPERSTRUCTURE DECK

DECK PLATING : 6.0 mm
DECK LONGITUDINALS : L 65 × 50 × 6

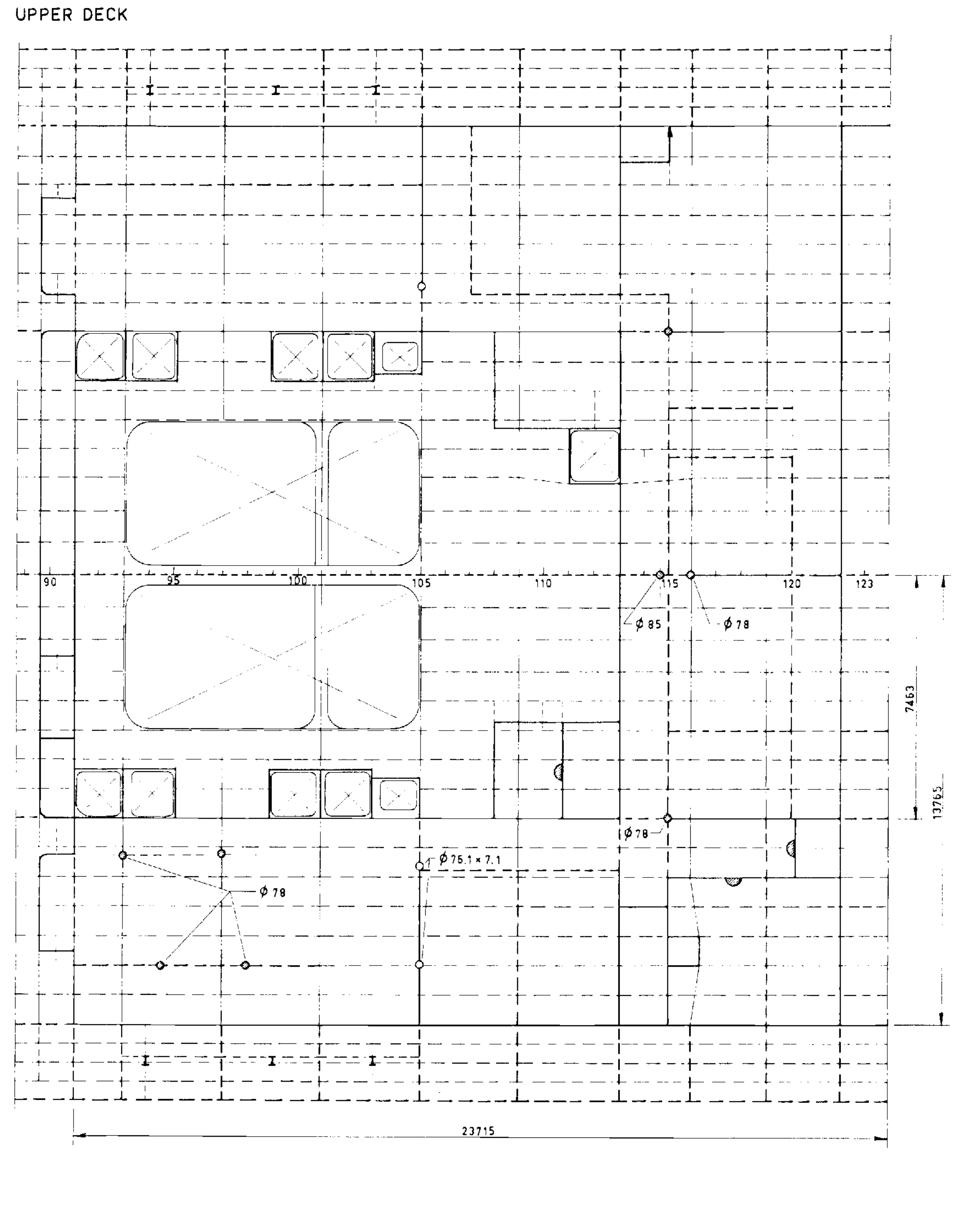


5TH SUPERSTRUCTURE DECK

THICKNESS OF DECK PLATING 6.0 mm
DECK LONGITUDINALS L 65 × 50 × 6

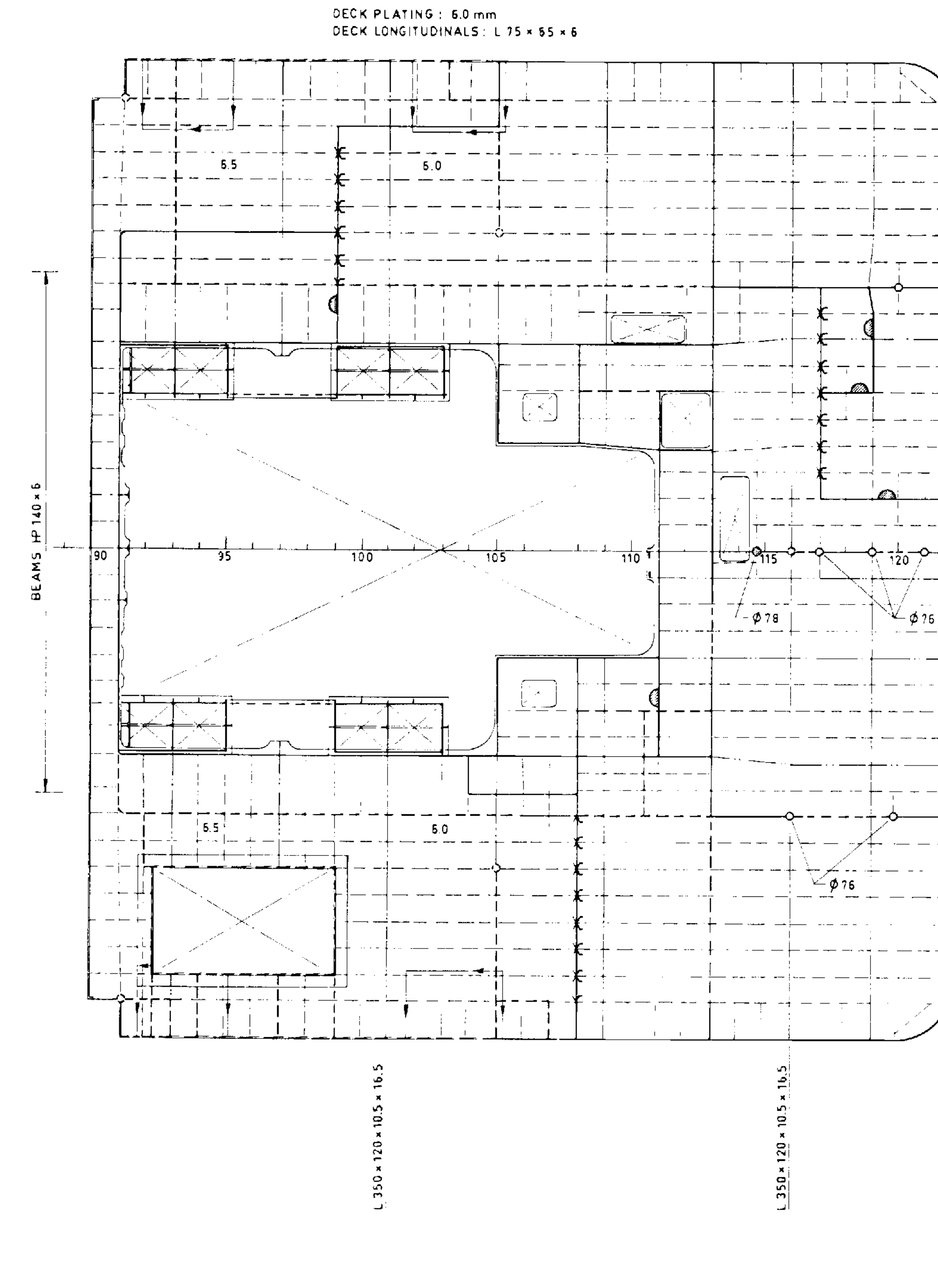


UPPER DECK



2ND SUPERSTRUCTURE DECK

DECK PLATING : 6.0 mm
DECK LONGITUDINALS : L 75 × 55 × 6



4TH SUPERSTRUCTURE DECK

DECK PLATING : 6.0 mm
DECK LONGITUDINALS L 65 × 50 × 6